THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 37

BUFFALO MEETING
SAN FRANCISCO MEETING
NEW YORK MEETING
1915

NEW YORK
PUBLISHED BY THE SOCIETY
29 West 39th Street
1916

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California	199	New Mexico	4
Canal Zone	12	New York	1800
Colorado	34	North Carolina	19
Connecticut	310	North Dakota	2
Delaware	41	Ohio	459
District of Columbia	57	Oklahoma	9
Florida	11	Oregon	10
Georgia	24	Pennsylvania	793
Hawaii	12	Philippine Islands	8
Idaho	5	Porto Rico	6
Illinois	483	Rhode Island	94
Indiana	98	South Carolina	9
Iowa	27	South Dakota	2
Kansas	24	Tennessee	45
Kentucky	16	Texas	42
Louisiana	33	Utah	16
Maine	23	Vermont	19
Maryland	71	Virginia	44
Massachusetts	617	Washington	37
Michigan	247	West Virginia	14
Minnesota	81	Wisconsin	135
Mississippi	6	Wyoming	2
Missouri	131		
Montana	13	Total	6595

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Africa	14	Holland	1
Australia	7	India	7
Austria	2	Italy	2
Belgium	1	Japan	8
British West Indies	1	Mexico	6
Canada	107	Norway	3
Central America	1	Roumania	1
Channel Islands	1	Russia	9
China	3	Scotland	3
Cuba	18	South America	25
Denmark	1	Spain	3
Dutch East India	2	Sweden	4
England	63	Switzerland	2
Finland	2	Turkey	2
France	13	West Indies	1
Germany	19		
		Total	333

Summary by Residence

Membership in United States	6595
Membership in Foreign Countries	333
Present address unknown	3
Total Membership	6931

Summary by Grades

Honorary Members	14
Members	4500
Associates	407
Associate-Members	564
Juniors	1446
Total Membership December 31, 1915	6931

CONTENTS OF VOLUME XXXVII

BUFFALO, SAN FRANCISCO AND NEW YORK MEETINGS

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 37—1915

THE affairs of The American Society of Mechanical Engineers during the year 1915 are recorded in this volume. A general review of the work of the Society for this year is given in the Annual Report of the Council, here included. The volume contains a calendar of all the meetings of the Society—general meetings and local meetings—held during the year; and the papers, with discussion, presented at the Spring Meeting in Buffalo, the Annual Meeting in New York and the additional general meeting in San Francisco in connection with the Panama-Pacific International Exposition and the International Engineering Congress, 1915, are published practically in full. Papers read at local meetings and presented subsequently at a general meeting are also included; other local meeting papers will be found in The Journal for 1915, where a more detailed account of the local meetings is given. The reports of special committees received by the Council during 1915 and ordered published are here included; of special note among these is the report of the Power Test Committee. The reports of Standing Committees for the year are published in The Journal, December 1915.

JOHN A. BRASHEAR

John A. Brashear, President of The American Society of Mechanical Engineers for the year 1915, was born in Brownsville, Pa. in 1840, and was educated in the public schools of that town. He learned the machinist's trade, and from 1860 to 1880 worked in the rolling mills of Pittsburgh. The study of astronomy had been his hobby from boyhood, however, and all his spare time was devoted to it. In 1876 he attracted the attention of Langley, then head of the Allegheny Observatory, and after two years of work in a small shop on the South Side, Pittsburgh, where he was aided nobly by his wife, Brashear was enabled through the assistance of Mr. William Thaw, one of the Observatory patrons, to set up works in Allegheny for the manufacture of astronomical instruments. In a very short time he became widely known as an expert in the development and manufacture of astronomical instruments of precision. About this time his son-in-law, Mr. James B. McDowell, joined him in the work, and Brashear gives to him a large share of credit in the success of the undertaking. It is safe to say that a history of the work of Brashear's shop would be closely associated with the history of astronomy during the last thirty-five years.

Perhaps his most important achievement has been in connection with the design and development of the spectroscope for astronomical uses, particularly with reference to the accurate optical and mechanical features. In 1888 he completed the spectroscope for the 36-in. telescope of the Lick Observatory, furnishing the mechanical and optical parts. The excellence of the work which has been done by Professor Keeler at the Lick Observatory is freely attributed to Dr. Brashear's skill and genius, and many of the spectroscopes in the principal observatories of the world have been made in the Brashear workshop, as also many of the largest telescopes and objectives for astronomical research.

Dr. Brashear's more purely scientific work also brought recognition, and about the time of his removal to Allegheny he was given an appointment in the University of Western Pennsylvania of which the Allegheny Observatory was a department. From 1898 to 1900 he was acting director of the Allegheny Observatory, and has raised \$300,000 for the building and equipment of a new observatory in Riverview Park. He has always kept in touch with the development of this observatory, and through his efforts one department has been put in every possible way at the disposal of the public, as well as for astronomical and astrophysical research.

For twenty years past Dr. Brashear has been a trustee of the Carnegie Institute, for fifteen years of the Carnegie Institute of Technology, and for twenty years of the University of Pittsburgh, of which latter he has also served as Chancellor, and it is said that he has done more for the cause of education in Pittsburgh than any other three men. Several years ago a friend placed in his hands an endowment fund of $250,000, to be used for the advancement of teachers and teaching in the public schools, as a result of which to date over seven hundred teachers have been sent to different parts of the country for rest and study, bringing back with them new ideas and greater enthusiasm.

Dr. Brashear was elected to membership in The American Society of Mechanical Engineers in 1891, and was made an Honorary member in 1908. He was a manager of the Society from 1899 to 1902. In 1911, he was elected one of the Society's representatives on the John Fritz Medal Board, to serve for four years.

He is a fellow of the American Association for the Advancement of Science and the Royal Astronomical Society of Great Britain; is a past-president of the Engineers Society of Western Pennsylvania and the Pittsburgh Academy of Arts and Sciences; is a member of the British Astronomical Association, the Société Astronomique de France, the Société de Belgique, the American Philosophical Society, the Astrophysical Society of America, the Washington Academy of Sciences, the National Geographic Society, and an Honorary Member of the Royal Astronomical Society of Canada.

He has been honored with the degree of LL.D. by Washington and Jefferson College, by Wooster University and by the University of Pittsburgh, and with the degree of Sc.D. by Princeton University and the Western University of Pennsylvania; also with the degree of Doctor of Engineering by Stevens Institute of Technology.

ANNUAL REPORT OF THE COUNCIL

The Council herewith presents in brief the important phases of the Society activities during the year 1915, under the presidency of Dr. John A. Brashear. In reports published in The Journal, December 1915, the Standing Committees have given accounts of their work in detail.

This year the plan to have a member of the Council on every Standing Committee was carried out so far as possible in the appointments by the President. The new appointments on Standing Committees for the year were: *Finance*, A. E. Forstall; *Meetings*, L. P. Alford; *Publication*, Henry Hess; *Library*, Jesse M. Smith; *Membership*, George A. Orrok and later Charles E. Lucke; *House*, O. P. Cummings; *Research*, R. J. S. Pigott, A. M. Greene, Jr.; *Public Relations*, Spencer Miller; *Constitution and By-Laws*, James E. Sague.

MEMBERSHIP

Under the guidance of I. E. Moultrop, Chairman of the Increase of Membership Committee, coöperating with the Membership Committee in keeping to high ideals of membership, there has been a steady increase in enrollment of the most representative men of the profession. Fig. 1 gives comparative data on the membership of the four national engineering societies.

On account of the unusual conditions prevailing abroad and the inability to communicate with members there, foreign engineers who are members of the Society have been continued on the rolls of the Society until further action by the Council.

The Council elected to Honorary Membership, E. D. Leavitt, the second President of the Society, who served during one of the most difficult periods of its history. Mr. Leavitt is best known for his notable successes in the design of high-duty, compound pumping engines for city waterworks service and of engines and other machinery for some of the great mining operations of the country.

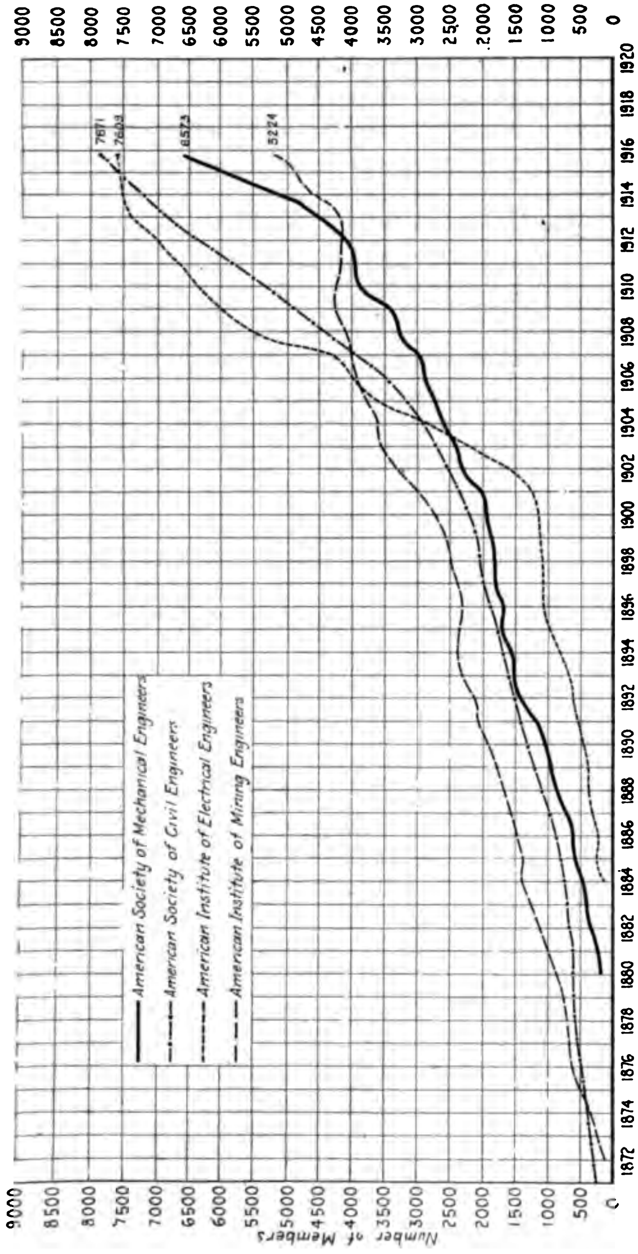

FIG. 1. COMPARATIVE GROWTHS OF MEMBERSHIP IN THE FOUR NATIONAL ENGINEERING SOCIETIES FROM THEIR ORGANIZATION TO OCTOBER 1, 1915

LOCAL SECTIONS

The Council at its January meeting appointed a Committee on Local Sections, the personnel of which is E. H. Whitlock, Cleveland, *Chairman*, W. F. M. Goss, Urbana, Ill., L. C. Marburg, New York, Walter Rautenstrauch, New York, and D. Robert Yarnall, Philadelphia. It hoped by this means to develop and put on an efficient basis this important phase of the Society's work.

One of the first plans carried out by the Committee was a conference at the Spring Meeting of Local Section representatives. Delegates attended from Atlanta, San Francisco, Milwaukee, Chicago, New York, Providence, Philadelphia, Cleveland, Worcester and Birmingham, and so much of interest developed that the Committee arranged for a second conference at the Annual Meeting, 1915. It is planned to make these conferences of Local Section chairmen a feature of either the Annual or Spring Meetings.

INCREASE IN MEMBERSHIP OF THE AM. SOC. M. E. DURING THE FISCAL YEAR 1914-1915

Grade	Oct. 1, 1913	Losses				Additions		Net Increase	Oct. 1, 1915
		Trans.	Resign	Lapsed	Death	Trans.	Elect.		
Hon. Mem...	14				1	1			14
Member	4077	1	29	36	38	33	328	257	4334
Assoc.	380	12	6	9	4	1	39	8	388
Assoc. Mem..	245				1	20	114	203	448
Junior	1274	42	25	24	2		212	119	1393
Total	5990	55	60	69	46	55	762	587	6577

Note—Affiliates, students' sections, 955, a decrease of 36 since 1914.

FINANCES

The annual report of the Finance Committee, together with the auditors' report, covers fully the financial status of the Society. In this connection the Council wishes to call attention to the fact that our life membership fund ($44,120.11), representing all that has been paid in for life memberships, is maintained as a separate fund and is invested. Similarly all the other trust funds, and for quite a number of years all the initiation fees, have been put into a fund and invested. From this latter fund, at a rate of $6,000 per year, we are retiring the certificates of indebtedness for the land on which the headquarters is built. We have thus accumulated $40,268.90 to offset the $54,100 of certificates of indebtedness. We have also $40,000 cash on hand, so, if we wished to, we could pay off all these certificates and be free and clear, but to do so, would leave little working capital. As we are doing a business of $150,000 a year, and discount all bills, it is economy to carry this extra cash, particularly as all our funds are invested at the same rate of interest as we are paying on the certificates.

Financially, we are one of the strongest engineering societies in the world, although we have secured far less by gift than many similar societies.

MEETINGS

The Society has held its general meetings during the year, the Annual Meeting in New York and the Spring Meeting in Buffalo, N. Y.

An additional general meeting was held in San Francisco, September 16 and 17. The headquarters of the Society was the Clift Hotel and the local arrangements were in the hands of the local committee of the Society, F. W. Gay, *Chairman*, F. H. Varney, *Vice-Chairman*, C. F. Braun, *Secretary*, H. L. Terwilliger and J. T. Whittlesey. Five professional papers were presented. This meeting preceded the general sessions of the International Engineering Congress, 1915.

LOCAL MEETINGS

Local meetings have been held under the auspices of the several local sections committees of the Society in Atlanta, Boston, Buffalo, Chicago, Cincinnati, Los Angeles, Milwaukee, Minnesota, New Haven, New York, Philadelphia, St. Louis, San Francisco, and the new section which has this year been established at Worcester.

Coöperation of local sections of the Society with existing engineering organizations in the same city has been enthusiastically supported by the Council in the endeavor to promote general concurrence of all engineering societies in technical meetings. St. Louis has formed such an affiliation with the Engineers' Club of that city and the Philadelphia Local Sections' Committee with the Engineers' Club of Philadelphia.

PUBLICATIONS

The history of the Society from the time of its formation in 1880 through the year 1915, has been completed and issued uniform with Transactions in size and binding, with the exception of color, dark green being employed to designate extra publications of the Society.

The Council has approved the plans of the Publication Committee as given in its report, which outlines a broad and progressive policy for the several publications of the Society.

COMMITTEE REPORTS

The Council has ruled that the signatures of members of the committee grouped as assenting or dissenting must accompany all reports. This ruling is covered in the following by-law:

> B-47 All written reports of all committees shall be presented to the Council. Each written report of every committee must be approved in writing by at least a majority of the members of that committee, before it is presented to the Council. A member of a committee who disagrees with the action of a majority of that committee may express his disagreement over his signature, either on the report of the committee or in a minority report. The minority report of any member of a committee if offered, shall be presented at the same time that the report of that committee is presented to the Council.
>
> All reports of committees must be first received by the Council who shall prescribe the manner in which they shall be presented to the membership of the Society and be made public and printed.

Boiler Code. On the completion of the report of the committee having in hand the preparation of a code for steam boilers, the original committee and its advisory committee were reappointed as one committee and asked to continue and extend its work to cover any structures and their operation that are connected with and serve purposes similar to boilers. The amount of time and work which the various members of this entire committee have given to the preparation of this report is so great that the Council wishes to record here its sincere appreciation.

Threads for Fixtures and Fittings. As the result of careful consideration of this subject and the coöperation by correspondence in detail with a selected list of thirty-eight of the largest manufacturers, the Committee on Threads for Fixtures and Fittings has completed its work on Rolled Threads. The report on Straight Threads will be presented at the Annual Meeting, 1915.

Tolerances for Screw Threads is a subject which is receiving the attention of a special committee, and with the ready coöperation of manufacturers of taps and dies, it is now at work on tolerances for taps and screw threads.

The report of the Power Test Committee is completed and presented for record in the Transactions for 1915. This is another example of years of self-sacrifice resulting in a complete standard for power tests.

EMPLOYMENT WORK

There has been marked activity in the employment work this year. Over 400 positions with firms who have asked our coöperation in securing men have been registered; this does not include recommendations of experts in consulting work of which we do not keep record but for whom we have many inquiries. The Society has record of 100 men who have been placed in positions through this work, and we are confident that this cannot represent the entire list as firms and individuals in many cases fail to advise the Secretary's office. It may be of interest to note that the other national engineering societies have now adopted this policy of assisting members and bringing firms and men together.

STUDENT BRANCHES

Student Branches are established in 38 universities, representing a total enrollment of nearly 1,000.

PRIZES

Prizes for Technical Papers. The Society has received by bequest from the late Rear-Admiral George W. Melville $1,000, the income of which is to provide a gold medal to be awarded annually to such competing member of the Society as presents the best original paper or thesis. As an adequate die for a medal cannot be purchased out of the above fund, upon application to the Court the Society has been authorized to allow the fund to accumulate until sufficient to defray the cost of a suitable set of dies.

Student and Junior Prizes. There was reported last year the receipt of $2,000, through the generosity of a member of the Council, from which fund there are to be awarded annually three prizes, one of $50.00 to a Junior and two of $25.00 each to members of student branches who contribute the best original papers. It is felt that such awards will influence young engineers to undertake original work. The committees in charge of these awards are: For Juniors, R. H. Fernald, *Chairman;* Fred E. Rogers, George B.

Brand. For Students, Frederick R. Hutton, *Chairman;* R. H. Fernald, D. S. Kimball.

One award was made this year for the paper, "Flow of Air Through Thin Plate Orifices," by Ernest O. Hickstein, Junior Member.

As the papers submitted by members of student branches did not meet the requirements as to standard, no awards were made this year.

RELATIONS WITH OTHER SOCIETIES

The American Institute of Electrical Engineers, the American Institute of Mining Engineers and The American Society of Mechanical Engineers, representing the Founder Societies of the United Engineering Society, have coöperated by combining the insurance of the three libraries and establishing a Library Search Bureau. The representative of the Society on the Board of Trustees is John R. Freeman, appointed to succeed Fred J. Miller, whose term of office expired and who, under the by-laws, was not eligible for reelection; Jesse M. Smith and Alex. C. Humphreys, Past-Presidents are the other appointees.

The Engineering Foundation. A noteworthy incident in the history of the profession of engineering was the inauguration in January 1915 of The Engineering Foundation, a fund to be administered "for the advancement of arts and sciences connected with engineering and the benefit of mankind." The basis of this fund is the gift of $200,000 by Ambrose Swasey of Cleveland, Ohio. Past-President of this Society. The American Society of Civil Engineers, American Institute of Mining Engineers, American Institute of Electrical Engineers and The American Society of Mechanical Engineers are equally represented in the administrative board of The Foundation. The United Engineering Society has been made custodian of the funds. The representatives of this Society on the Board are Jesse M. Smith and Alex. C. Humphreys; Frederick R. Hutton is Secretary of the Board.

Conference Committee. By the death of Alfred Noble a vacancy was caused in our representation on the Joint Conference Committee of the National Engineering Societies, and Arthur M. Greene, Jr., member of the Council, was appointed to fill the place. The Conference Committee is composed of two representatives from each of the five national engineering societies, the American Society of Civil Engineers, American Institute of Mining Engineers, American Institute of Electrical Engineers, The American Society of Mechanical Engineers and Society of Naval Architects and Marine Engineers.

Honorary Vice-Presidents. Frank B. Gilbreth represented the Society on the occasion of the opening of the new house of the Verein deutscher Ingenieure in Berlin. Major Wm. H. Wiley, Treasurer of the Society and member of the Council, was appointed the official representative of the Society at the convention of the Atlantic Deeper Waterways Association in Savannah, Ga., F. F. Gaines acting as alternate.

Greetings. As has been their practice for several years, the Austrian Society of Engineers on the occasion of their annual meeting sent a letter of greeting to the Council.

Graphics. As the result of invitations extended by the Society, a number of associations of national scope have appointed representatives on a committee to make a study of the methods used in the different fields of endeavor for presenting statistics and quantitative data in graphic form. A preliminary report has been published for the purpose of inviting suggestions.

Classification of literature of applied science. Delegates from about twenty national technical societies have conferred with a view to perfecting a permanent organization for the purpose of preparing a classification of the literature of applied science which may be generally acceptable and adopted.

The name of this organization is Committee on Classification of Technical Literature. This Society is represented on the committee by F. R. Low, who is chairman of the committee of the whole, L. P. Breckenridge, W. W. Bird, A. E. Forstall, E. J. Prindle. The Committee from this Society will also take under consideration a proposed digest of material that has appeared in the publications of the Society, as suggested by the New York Section Committee on the occasion of a paper by E. J. Prindle on "A Proposed System of Classifying and Digesting the Records of the Society to Render Immediately Available all Information on each Branch of Every Subject."

Expert Testimony. On request from and coöperating with the American Association for the Advancement of Science, the Council appointed F. H. Richards, W. H. Boehm, and H. deB. Parsons, with authority to add two to their number, on a committee having for its object better practice in the use of expert testimony of engineers.

The American Society for Testing Materials has invited the Society to a proposed conference to consider the desirability and feasibility of coöperation in important matters of mutual interest and advantage to the various societies.

Hydraulic Flanges. A committee composed of H. G. Stott, *Chairman,* A. R. Baylis, A. M. Houser, Julian Kennedy, E. A. Stillman and W. M. White has been appointed to coöperate with the standardization committee of the Manufacturers' Association, in the recommendation of Flange Standards for Hydraulic Work.

International Engineering Congress, 1915. The International Engineering Congress was held in connection with the Panama-Pacific International Exposition in San Francisco. It was conducted under the auspices of the five national engineering societies, the American Society of Civil Engineers, American Institute of Mining Engineers, American Institute of Electrical Engineers, The American Society of Mechanical Engineers, the Society of Naval Architects and Marine Engineers, and was placed in the hands of a Committee of Management consisting of the presidents and secretaries of these five societies, and eighteen other members representative of them and resident in or near San Francisco.

For the convenience of the members and their guests of the national engineering societies, an "Engineers' Special" train leaving New York September 9, and arriving in San Francisco September 15, was run to the Congress.

Two hundred and forty-one papers were presented at the fifty-two sessions, and although the existing international conditions made impossible a representative delegation from abroad, sixty-six of the papers contributed were by authors outside of the United States. It is interesting to note that forty-eight authors of papers coming from the United States were members of this Society.

The return trip of a large proportion of the engineers was made along the lines of the Southern Pacific and Canadian Pacific through Portland to Seattle, Victoria, Vancouver, Glacier, Lake Louise, Banff, Calgary and Moose Jaw, giving the party an opportunity to inspect the engineering works in this section of Canada, the last objective being the big dam at Bassano.

PUBLIC RELATIONS

The Council received an invitation from Worcester Polytechnic Institute to be represented at the fiftieth anniversary of the charter of Worcester Polytechnic Institute. Dr. Brashear, President, and Calvin W. Rice, Secretary, attended.

On invitation of the Governor of Massachusetts, E. F. Miller, I. E. Moultrop and John A. Stevens were appointed to represent the Society at the hearing on boiler legislation on December 17, 1915.

Park A. Dallis was appointed to represent the Society at the inauguration of Edward Kidder Graham as President of the University of North Carolina, and W. R. Dunn represented the Society at the inauguration of John Henry MacCracken as President of Lafayette College.

Sir William H. White Memorial. The movement recently started by the engineering societies of Great Britain to erect a memorial to the late Sir William H. White was responded to by the appointment of a memorial fund committee of this Society composed of Jesse M. Smith, *Chairman,* Alex. C. Humphreys and Frederick R. Hutton. The contribution from the members and friends of this Society was 63 guineas.

Constitutional Convention. The Committee on Engineers representing national and local engineering societies was organized to present recommendations to the New York Constitutional Convention, and was composed of representatives of the American Society of Civil Engineers, American Institute of Electrical Engineers, The American Society of Mechanical Engineers, American Institute of Consulting Engineers, New York Section of the American Institute of Mining Engineers, Municipal Engineers of the City of New York, and the Brooklyn Engineers' Club.

The Committee's recommendations were for continuity of control in the conduct of the State's public works and by persons appointed by virtue of special fitness for these offices. In the report made by the joint committee the opinion is expressed that considerable success was secured by the engineers' efforts.

Naval Consulting Board. The Society was honored by an invitation from the Secretary of the Navy, Hon. Josephus Daniels, to participate in the work of the Naval Consulting Board, by the appointment of two representatives; and by letter ballot W. L. R. Emmet and Spencer Miller were selected by the Council as the appointees of this Society. The Board is composed of two representatives each from eleven societies, including the Chairman, Thomas A. Edison, Honorary Member of this Society.

Engineer Reserve Corps. The suggestion was made in the spring of 1915 that the national engineering societies offer to assist the United States War Department in the formation of an Engineer Reserve Corps in the United States Army. Acting on this, the

American Society of Civil Engineers, American Institute of Electrical Engineers, American Institute of Mining Engineers, The American Society of Mechanical Engineers, and the American Institute of Consulting Engineers appointed committees authorized and directed to take such steps for the organization of such a reserve corps as might be advisable.

In order to simplify future conferences a single committee has been formed to represent jointly the five societies—or the engineering profession as a whole. This general committee consists of Wm. Barclay Parsons, *Chairman,* Committee American Society of Civil Engineers; Henry S. Drinker, *Chairman,* Committee American Institute of Mining Engineers; B .J. Arnold, *Chairman,* Committee American Institute of Electrical Engineers; Wm. H. Wiley, Chairman, Committee The American Society of Mechanical Engineers; R. D. Mershon, *Chairman,* Committee American Institute of Consulting Engineers. The special committee in this Society is composed of Wm. H. Wiley, *Chairman,* John A. Hill, *Vice-Chairman,* W. F. M. Goss, H. A. Gillis and James M. Dodge. The separate committee in each society will take care of the work of its society as soon as the decision of the War Department can be given and a general scheme of organization adopted.

Second Pan-American Scientific Congress. In response to an invitation from the Department of State a committee of this Society is coöperating in the plans for a second Pan-American Scientific Congress to be held in Washington, December 27, 1915 to January 8, 1916. The Committee of this Society consists of General W. H. Bixby, *Chairman,* also Chairman in charge of Section V. Engineering, Carl C. Thomas, Charles T. Plunkett, S. W. Stratton and Calvin W. Rice.

Upon invitation from the Department of State, Ambrose Swasey, Past-President, and W. H. Marshall have been appointed as representatives of the Society to the Congress. Dr. John A. Brashear, President, has been selected by the Secretary of the Department of State of the United States, as the representative of the engineering profession in America.

Fire Hose Specifications. The Society coöperated with the Chamber of Commerce in the joint appointment of H. de B. Parsons as the representative of both organizations on a special committee created by the Fire Department of the City of New York to revise the specifications for fire hose. The work has been performed and the report placed in the files of the Society.

No. 1473

MEETINGS JANUARY—JUNE

MEETINGS OF SECTIONS

BOSTON, JANUARY 6

Topic: Aviation. Albert A. Merrill discussed fore and aft stability; Greely S. Curtis, Mem. Am. Soc. M. E., presented a number of interesting slides showing aeroplanes in flight; Joseph C. Riley, Mem. Am. Soc. M. E., showed a number of slides of gasoline engines including four, six and eight cylinder models.

BUFFALO, JANUARY 7

Address: Recent Development in Steam Turbine Engineering, J. A. Moyer, Mem. Am. Soc. M. E.

CHICAGO, JANUARY 8

Subjects: Superheaters; Mechanical Stokers; Railway Economics. R. M. Ostermann and Robert Quayle, Members Am. Soc. M. E. discussed superheaters; Clement F. Street, Mem. Am. Soc. M. E. covered mechanical stokers, and Willard A. Smith, president of the *Railway Review,* spoke on railway economics. Mr. Ostermann's paper was published in The Journal, July, 1915.

ST. LOUIS, JANUARY 11

Annual Meeting and Dinner. Addresses by Edward Flad, Mem. Am. Soc. M. E., E. R. Kinsey, president of the Board of Public Service, Joseph A. Hook, director of Public Utilities, and E. R. Fish, Mem. Am. Soc. M. E.

PHILADELPHIA, JANUARY 14

Joint meeting with the Metallurgical Section of The Franklin Institute. Paper: Modern Steels and Their Heat Treatment by Robert R. Abbott, metallurgical engineer for the Peerless Motor Car Company. Published in The Journal, May, 1915.

LOS ANGELES, JANUARY 15

Paper: The Diesel Engine and its Applications in Southern California, by Walter H. Adams, Mem. Am. Soc. M. E. Published in this volume.

NEW YORK, JANUARY 15

Lecture: A Really Greater New York, by T. Kennard Thomson, Mem. Am. Soc. M. E.

CINCINNATI, JANUARY 21

Joint meeting with the Engineers' Club of Cincinnati. R. W. Rew, Department of Public Service of Cincinnati, spoke on the Engineering Features of the Proposed Rapid Transit System for Cincinnati.

BUFFALO, JANUARY 28

Subject: Manufacture of Portland Cement, by Prof. R. C. Carpenter, Mem. Am. Soc. M. E.

MINNESOTA, JANUARY 28

Joint meeting with the Western Association of Electrical Inspectors. Address: Principles Entering into Valuation of Public Utilities and Rate Making, C. L. Pillsbury, Mem. Am. Soc. M. E.

NEW YORK, FEBRUARY 9

Address: A Proposed System of Classifying and Digesting the Records of the Society to Render Immediately Available all Information on Each Branch of Every Subject, by Edwin J. Prindle, Mem. Am. Soc. M. E. Published in THE JOURNAL, May, 1915.

MILWAUKEE, FEBRUARY 10

Lecture: New Linwood Avenue Intake Tunnel, by L. G. Warren, resident engineer.

BUFFALO, FEBRUARY 11

Address: Bearings and Their Lubrication, by C. H. Bierbaum, Mem. Am. Soc. M. E.

ATLANTA, FEBRUARY 12

Luncheon. General discussion relative to the welfare and prospects of the Section.

MINNESOTA, FEBRUARY 12

Address: Triangulation, by Professor Stewart, of the Minnesota Agricultural College.

BOSTON, FEBRUARY 15

Sixth annual joint Engineers' Dinner, under the auspices of the Boston Society of Civil Engineers, the Boston Section of the American Institute of Electrical Engineers, and the Boston Section of the Society. Speakers: Hon. David I. Walsh, Governor of Massachusetts, Dr. Allan McLaughlin, Charles H. Eglee, of the Aberthaw Construction Co., Capt. Robert W. Bartlett, Harrison P. Eddy, and Charles Whiting Baker, Mem. Am. Soc. M. E.

ST. LOUIS, FEBRUARY 16

Committee appointed to prepare a tribute to the late Col. E. D. Meier, Past-President Am. Soc. M. E.

BUFFALO, FEBRUARY 25

Paper: Waste in Hiring and Discharging Employees, by M. W. Alexander, Mem. Am. Soc. M. E.

CINCINNATI, FEBRUARY 25

Dinner in honor of Calvin W. Rice, Secretary Am. Soc. M. E. Address by Mr. Rice describing trip of the Society to Germany in the summer of 1913.

BOSTON, FEBRUARY 26

Joint meeting with the American Institute of Electrical Engineers and the Boston Society of Civil Engineers. Subject: Training and Education of Employees and the Relation of the Employer to His Men and Their Education. Papers: The Responsibility of the Manufacturer for Training of Foremen and Skilled Workmen, by Walter C. Fish, general manager Lynn Works, General Electric Company; The Employer's Side of the Problems of Irregular Employment, Henry S. Dennison, treasurer, Dennison Manufacturing Company; Coöperation Between Employers and the Schools, William B. Hunter, director of Fitchburg Industrial School; The Economic Relation Between the Supply of Skilled and Intelligent Workmen and Unemployment of the Masses, Thomas N. Carver, Harvard University.

CHICAGO, FEBRUARY 26

Subject: Refrigeration; Ice-Making as a By-Product of Central Stations.

NEW YORK, MARCH 9

Address: Application of Engineering Methods to the Problems of the Executive, Director and Trustee, by Hollis Godfrey, Mem. Am. Soc. M. E. Published in this volume.

BUFFALO, MARCH 11

Paper: Interesting Features Involved in the Design of the Connors Creek Plant of the Edison Illuminating Company of Detroit, by C. F. Hirshfeld, Jun. Am. Soc. M. E. Published in this volume.

CINCINNATI, MARCH 18

Joint meeting with Engineers' Club of Cincinnati. Address: Electric Commercial Vehicle, T. H. Schoepf, Mem. Am. Soc. M. E.

CHICAGO, MARCH 19

Subject: Refrigeration with Special Reference to Ice-Making as a By-Product of Central Stations, by Heywood Cochrane, Western manager, Carbondale Machine Company. Published in THE JOURNAL, July, 1915.

BUFFALO, MARCH 25

Illustrated Address: Our Navy and What it Means, by Edward Breck, Field Secretary of the Navy League, Washington, D. C.

LOS ANGELES, MARCH 25

Paper: Gas Volume and Dust Concentration Determination in connection with the Cottrell Process, Wm. N. Drew, Mem. Am. Soc. M. E. Published in THE JOURNAL, December, 1915.

BOSTON, MARCH 31

Illustrated lecture on the Engineering Equipment of the New Technology Buildings, by Harry Gay, Mem. Am. Soc. M. E.; Geo. E. Libbey of the firm of Hollis French and Allen Hubbard, described the Heating, Ventilating and Sanitary Features of the Work; and Prof. A. L. Williston, Mem. Am. Soc. M. E., gave an illustrated talk on the Layout of Educational Institutions.

CHICAGO, APRIL 2

Subject: Power Plant Apparatus and General Equipment,

ST. LOUIS, APRIL 7

Joint meeting of the St. Louis Engineering Societies. Paper: City Water Supply, by Edward E. Wall.

BUFFALO, APRIL 8

Inspection trip. Mr. Hershey of Chicago presented a detailed illustrated explanation of the construction and operation of the automatic telephone apparatus. After the lecture an inspection trip was made through the Buffalo plant of the Federal Telephone and Telegraph Company.

WORCESTER, APRIL 8

Appointment of Committee: Paul B. Morgan, *Chairman*, E. H. Reed, *Secretary*, Carl F. Dietz, H. P. Fairfield, and F. W. Parks.

PHILADELPHIA, APRIL 12

Joint meeting with Philadelphia Section of the American Institute of Electrical Engineers. Paper: Turbine Driven vs. Engine Driven Units in Small Capacities, J. S. Barstow. Published in this volume.

NEW YORK, APRIL 13

Paper: Modern Electric Elevator and Elevator Problems, David Lindquist, chief engineer of Otis Elevator Company. Published in this volume.

SAN FRANCISCO, APRIL 16

Spring meeting of the Section. Paper: Design and Test of a Large Reclamation Pumping Plant, G. C. Noble. Illustrated.

NEW HAVEN, APRIL 21

Spring meeting of the Section. General subject: The Development of Machine Tools. Papers: The Early History of Machine Tools, Joseph W. Roe, Mem. Am. Soc. M. E.; Modern Developments in Milling Machines, Luther D. Burlingame, Mem. Am. Soc. M. E.; Milling Cutters and Cutting Tools, A. L. DeLeeuw, Mem. Am. Soc. M. E.; Modern Development in Vertical Boring and Turning Machines, E. P. Bullard, Jr., Mem. Am. Soc. M. E.; Special Forms of Presses for Working Sheet Metal, Darragh deLancey, Mem. Am. Soc. M. E.; Grinding as a Manufacturing Process, H. W. Dunbar, Norton Grinding Company, Worcester, Mass.

CINCINNATI, APRIL 22

Joint meeting with Engineers' Club of Cincinnati. Address: Forecasting the Weather, W. C. Devereaux. Illustrated.

MINNESOTA, APRIL 22

Paper: The Manufacture of Illuminating Gas, D. W. Flowers, Assoc-Mem. Am. Soc. M. E.

BOSTON, APRIL 23

Joint meeting with the American Institute of Electrical Engineers. Paper: Electrical Equipment Used in the Commonwealth Pier Development for the Port of Boston, Frank W. Hodgdon, chief engineer for the Directors of the Port of Boston. Illustrated.

BUFFALO, APRIL 27

Papers: Patents, Charles W. Parker; Conservation, H. B. Alverson, Mem. Am. Soc. M. E.

BUFFALO, MAY 6

Paper: Concrete, and Machine for Making Cement, Rolla C. Carpenter, Mem. Am. Soc. M. E.

MINNESOTA, MAY 10

Joint meeting with the American Institute of Electrical Engineers. Papers: Lake Nokomis Electric Dredge, Mr. Brillhart of the Minneapolis General Electric Company; Application of Electric Drive to Paper Mills and Data concerning the Paper Industry, H. F. Teetsell, Assoc. Am. Soc. M. E.

NEW YORK, MAY 11

Paper: Metal Spray Processes in Engineering and Art, John Calder, Mem. Am. Soc. M. E. Published in THE JOURNAL, July, 1915.

CHICAGO, MAY 14

Paper: The Electric Locomotive, by A. H. Armstrong, of the General Electric Co. Published in THE JOURNAL, July 1915.

ST. LOUIS, MAY 19

General discussion on requirements and length of an engineering course of study, by A. S. Langsdorf, J. L. Van Ornum, E. L. Ohle, Mem. Am. Soc. M. E., G. O. James and E. L. McCausland.

CINCINNATI, MAY 20

Joint meeting with Engineers' Club of Cincinnati. Illustrated Lecture: Egypt, Light of the World, A. O. Zwick.

ST. LOUIS, JUNE 9

Joint meeting with St. Louis Section of the American Society of Engineering Contractors. Paper: Needed Improvements in Specifications, J. B. Emerson, of Robert W. Hunt and Co. This was followed by an outline of tests by the U. S. Bureau of Standards on full sized columns, by R. G. Olhausen.

WORCESTER, JUNE 9

Meeting in connection with the celebration of the 50th Anniversary of Worcester Polytechnic Institute. Paper: The Washburn Shop of the Worcester Polytechnic Institute, Geo. I. Alden, Mem. Am. Soc. M. E. Published in THE JOURNAL, July, 1915.

MINNESOTA, JUNE 11 AND 12

Joint meeting with Minnesota Section of the American Institute of Electrical Engineers, held in Duluth. Paper, illustrated by lantern slides: The Great Northern's Development and Business, William N. Ryerson, Mem. Am. Soc. M. E. This was followed by a moving picture talk on the Iron Ore Industry of Minnesota, by John Hearding, of the Oliver Iron Mining Co.

The second day was devoted to excursions.

LOS ANGELES, JUNE 15

Joint meeting of the technical societies in Los Angeles. Subject: The Service of the Technical Man to the Community. Speakers: William Mulholland, chief engineer of the Los Angeles Water Board, Samuel Storrow and James A. B. Sherer, president of the Throop College of Technology.

ST. LOUIS, JUNE 16

Joint meeting under the auspices of the Am. Soc. M. E. Paper: Boiler Explosions, and What the A. S. M. E. is Doing to Prevent Them, E. R. Fish, Mem. Am. Soc. M. E. Published in THE JOURNAL, September, 1915.

CINCINNATI, JUNE 24

Joint meeting with Engineers' Club of Cincinnati. Discussion on The Relations between the Valuation of Public Utilities and the Determination of Rates. Discussed by J. A. Lilly, O. F. Shepard, and F. R. Healey.

THE SPRING MEETING

The Spring Meeting of the Society at Buffalo, N. Y., the first general meeting of the Society in this city, was held from Tuesday, June 22 to Friday, June 25, with headquarters at the Hotel Statler. A total of 424 registered, 223 members and 201 guests.

The headquarters were opened for registration at 2 p. m. on Tuesday, and the registration on the first day exceeded 150. A meeting of the Research Committee was held at 4 p. m., and at 6 o'clock the officers and representatives of Local Sections met the Local Sections Committee at a conference and dinner.

On Tuesday evening, an informal reception was held in the ball room of the Hotel Statler. Chairman David Bell of the Buffalo Local Section introduced Frank B. Baird, of Buffalo, who delivered an address of welcome. The President responded for the Society.

The first professional session was held at Niagara Falls on Wednesday, in the auditorium of the Shredded Wheat Company's manufactory. At this session four papers were presented. After the session the visitors were conducted on a tour of inspection of the company's plant. In the afternoon the power plants at the Falls were visited.

An entertaining lecture was delivered on Wednesday evening by Dr. F. H. Newell, Mem. Am. Soc. M. E., of the University of Illinois, and formerly chief of the U. S. Reclamation Service. The subject was The Engineer as a Citizen, and the lecture was illustrated with beautifully colored slides showing striking views of the reclamation work.

On Thursday morning seven papers were presented at two simultaneous professional sessions, and on Friday morning the concluding session was held, when three papers were presented. All the papers are listed in the program below.

A reception and dance, which proved to be a most delightful gathering, was held at the Hotel on Thursday evening. Another pleasurable event was a tea given by the members of the Twentieth Century Club of Buffalo for the entertainment of the ladies and members attending the convention. A great deal of interest was shown in the technical excursions to industrial plants in Buffalo, many of which opened their doors freely to the visitors.

The local committees had made most complete, and even elaborate preparations for the reception of the guests. During the time of the meeting many of the committee members were in constant attendance, and nothing was left undone that would in any way contribute to the pleasure of those present. The chairmen of the several local committees were the following: General Committee, David Bell; Finance Committee, D. W. Sowers; Reception Committee, H. P. Parrock; Entertainment Committee, David C. Howard; Women's Committee, Mrs. William Henry Barr; Hotel Committee, W. H. Carrier; Printing and Publicity Committee, John Younger.

PROGRAM

BUFFALO

Tuesday Afternoon, June 22

Registration of members and guests at headquarters.

Tuesday Evening

Informal reception by the members of the Engineering Society of Buffalo and local members of our Society. Introductory remarks by David Bell, chairman of the Buffalo Local Section. Address of welcome by Frank B. Baird. Response by President John A. Brashear.

NIAGARA FALLS

Wednesday Morning, June 23

BUSINESS MEETING

Report of Tellers on Amendment to C 45 of Constitution. Proposed amendments to C 48 and C 54 of the Constitution. Report of Committee on Special Threads for Fixtures and Fittings.

PROFESSIONAL SESSION

A STUDY OF AN AXLE SHAFT FOR A MOTOR TRUCK, John Younger.

Discussed by Radclyffe Furness, Cornelius T. Myers, H. Wade Hibbard, The Author.

A COMPARISON OF THE PROPERTIES OF NICKEL, CARBON AND MANGANESE STEEL BEFORE AND AFTER HEAT TREATMENT, Robert R. Abbott.

Discussed by Henry M Howe.

THE USE OF CORRUGATED FURNACES FOR VERTICAL FIRE-TUBE BOILERS, F. W. Dean.

Discussed by W. F. MacGregor, H. Wade Hibbard, A. M. Greene, Jr., Forrest E. Cardullo, Chas. H. Manning, The Author.

ON MEASURING GAS WEIGHTS, Thos. E. Butterfield.

Discussed by Arthur West, Sanford A. Moss, The Author.

Wednesday Afternoon

Technical excursions.

BUFFALO

Wednesday Evening

Lecture by Dr. F. H. Newell, Mem. Am. Soc. M. E., of the University of Illinois and formerly Chief of the U. S. Reclamation Service, on The Engineer as a Citizen. Illustrated by colored lantern slides.

Thursday Morning, June 24

PROFESSIONAL SESSION

A Basis for Rational Design of Heat Transfer Apparatus, E. E. Wilson.

Discussed by R. C. H. Heck, Leo Loeb, Edgar Buckingham, A. M. Greene, Jr., C. F. Braun, The Author.

Influence of Disk Friction on Turbine Pump Design, F. zur Nedden.

Discussed by C. G. de Laval, M. D. Hersey, The Author.

The Surface Condenser, C. F. Braun.

Discussed by F. W. Reynolds, H. Wade Hibbard, The Author.

SIMULTANEOUS SESSION

Some Mechanical Features of the Hydration of Portland Cement and the Making of Concrete as Revealed by Microscopic Study, Nathan C. Johnson.

Discussed by John R. Freeman, H. F. Porter, The Author.

Design of Rectangular Concrete Beams, Howard Harding.

Model Experiments and the Forms of Empirical Equations, Edgar Buckingham.

Discussed by M. D. Hersey, M. I. Nusim, A. R. Dodge, John R. Freeman, L. W. Wallace, The Author.

The Effect of Relative Humidity on an Oak Tanned Leather Belt, W. W. Bird and F. W. Roys.

Discussed by G. N. Van Derhoef, Carl G. Barth, F. B. Gilbreth, Wm. S. Aldrich, A. F. Nagle, W. W. Bird.

Thursday Afternoon

Technical Excursions. Tea at Twentieth Century Club.

Thursday Evening

Reception and Dance.

Friday Morning, June 25

PROFESSIONAL SESSION

On the Laws of Lubrication of Journal Bearings, M. D. Hersey.

Discussed by H. F. Moore, W. H. Herschel, F. zur Nedden, The Author.

The Relation Between Production and Costs, H. L. Gantt.

Discussed by D. B. Rushmore, Forrest E. Cardullo, W. N. Polakov, W. W. Bird, J. A. White, Chas. Piez, F. H. Neely, S. H. Bunnell, Keppele Hall, Carl G. Barth, Ralph E. Flanders, D. C. Fenner, C. B. Thompson, William Kent, The Author.

Laps and Lapping, W. A. Knight and A. A. Case.

Discussed by C. E. Gillett, W. A. Knight.

No. 1474

REPORT OF THE COMMITTEE ON STANDARDIZATION OF SPECIAL THREADS FOR FIXTURES AND FITTINGS

ROLLED THREADS FOR SCREW SHELLS OF ELECTRIC SOCKETS AND LAMP BASES

To The Council of The American Society of Mechanical Engineers:

Your Committee appointed on May 1st, 1914 to take up the subject of Standardization of Special Threads begs to report on rolled threads on sheet metal shells as follows:

2 During June 1912 some of the manufacturers of electrical wiring supplies and lamps held a meeting in the attempt to standardize these threads and arrived at a practical compromise on those then in use.

3 It being thought advisable to modify certain features of the standard agreed upon in 1912, another meeting of manufacturers was held March 18, 1914 and at this meeting The American Society of Mechanical Engineers was asked to take up the subject and make recommendations. This resulted in the appointment of your Committee.

4 Your Committee has held two meetings for the consideration of this subject and has conducted numerous conferences with manufacturers and corresponded in detail with a selected list of thirty-eight of the largest manufacturers. Only minor changes from the generally recognized standard have been made and twenty-seven manufacturers approve the recommendations and the balance of the thirty-eight, who have not replied to our letters, are—to the best of our knowledge—already using the recommended standards. No objections have been given.

5 An investigation of the German Standard on the medium size shells as given by J. E. Reinecker shows that the same number of threads per inch is used as on the American Standard but that the depth of thread is greater and differing diameter dimensions

Received by the Council, February 13, 1915, and ordered printed. Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

are in use. It has not seemed wise, therefore, to consider this standard.

6 The standards recommended are, therefore, as given below for the four sizes in both male and female shells. The male shells are used on lamp bases, fuse plugs, attachment plugs, etc., and the female shells for electric sockets, receptacles and similar devices.

The male shells are usually known as "lamp base screw shells" and the female as "socket screw shells."

7 Miniature size:

DIAMETER DIMENSIONS OF STANDARDS

	Socket Screw Shell	Lamp Base Screw Shell
"Go" gauge, top of thread	0.3775 in.	0.3750 in.
"Not Go" gauge, top of thread	0.3835 in.	0.3700 in.
"Go" gauge, bottom of thread	0.3375 in.	0.3350 in.
"Not Go" gauge, bottom of thread	0.3435 in.	0.3300 in.
Threads per inch	14	14
Depth of thread	0.020 in.	0.020 in.

Fig 1 shows the form of thread and for convenience repeats the above dimensions.

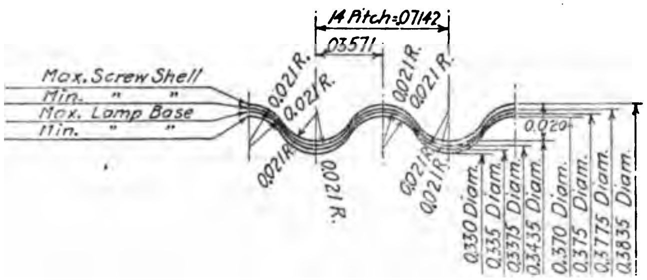

FIG. 1 FORM AND DIMENSIONS OF THREADS FOR LAMP BASE AND SOCKET SCREW SHELLS, MINIATURE SIZE

8 Candelabra size:

DIAMETER DIMENSIONS OF STANDARDS

	Socket Screw Shell	Lamp Base Screw Shell
"Go" gauge, top of thread	0.470 in.	0.465 in.
"Not Go" gauge, top of thread	0.476 in.	0.460 in.
"Go" gauge, bottom of thread	0.420 in.	0.415 in.
"Not Go" gauge, bottom of thread	0.426 in.	0.410 in.
Threads per inch	10	10
Depth of thread	0.025 in.	0.025 in.

Fig. 2 shows form of thread and for convenience repeats the above dimensions.

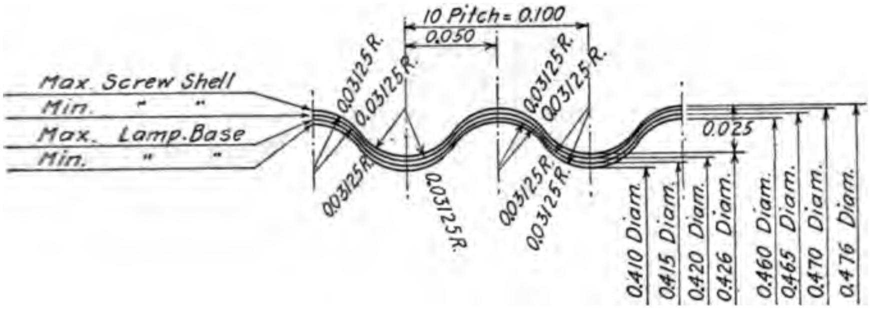

FIG. 2 FORM AND DIMENSIONS OF THREADS FOR LAMP BASE AND SOCKET SCREW SHELLS, CANDELABRA SIZE

9 Medium size:

DIAMETER DIMENSIONS OF STANDARDS

	Socket Screw Shell	Lamp Base Screw Shell
"Go" gauge, top of thread	1.045 in.	1.037 in.
"Not Go" gauge, top of thread	1.053 in.	1.031 in.
"Go" gauge, bottom of thread	0.979 in.	0.971 in.
"Not Go" gauge, bottom of thread	0.987 in.	0.965 in.
Threads per inch	7	7
Depth of thread	0.033 in.	0.033 in.

Fig. 3 shows form of thread and for convenience repeats the above dimensions.

10 Mogul size:

DIAMETER DIMENSIONS OF STANDARDS

	Socket Screw Shell	Lamp Base Screw Shell
"Go" gauge, top of thread	1.565 in.	1.555 in.
"Not Go" gauge, top of thread	1.577 in.	1.545 in.
"Go" gauge, bottom of thread	1.465 in.	1.455 in.
"Not Go" gauge, bottom of thread	1.477 in.	1.445 in.
Threads per inch	4	4
Depth of thread	0.050 in.	0.050 in.

Fig. 4 shows form of thread and for convenience repeats the above dimensions.

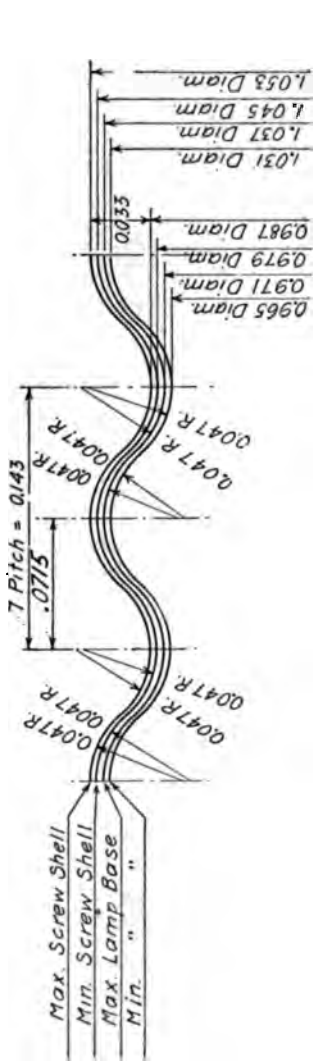

Fig. 3 Form and Dimensions of Threads for Lamp Base and Socket Screw Shells, Medium Size

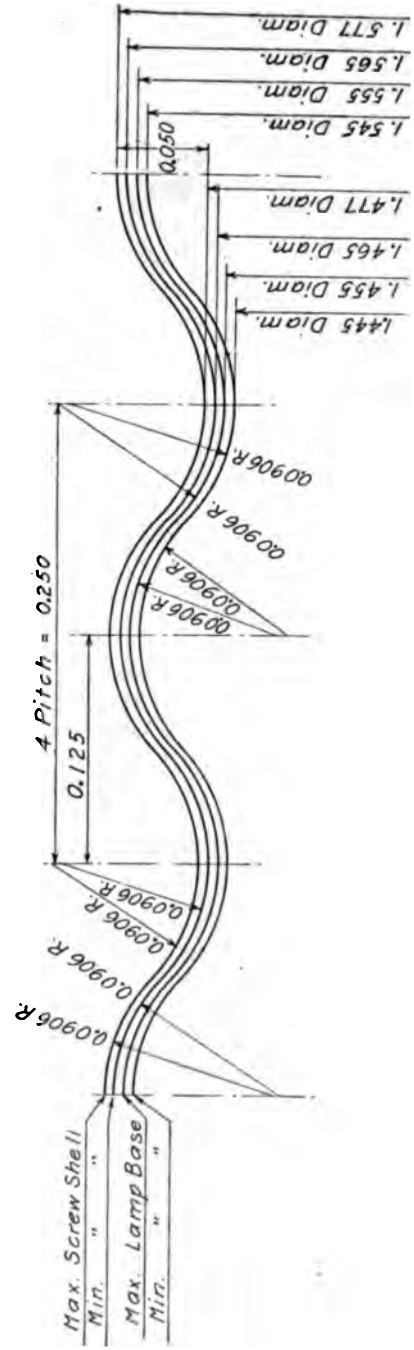

Fig. 4 Form and Dimensions of Threads for Lamp Base and Socket Screw Shells, Mogul Size

11 For each size of lamp base screw shells there should be two threaded ring gauges to govern the diameter of the bottom of the thread outside and the form of the thread. Also two plain ring gauges to govern the diameter of the top of the thread outside.

12 For each size of socket screw shells there should be two threaded plug gauges to govern the diameter of the top of the thread inside and the form of the thread. Also two plain plug gauges to govern the diameter of the bottom of the thread inside.

13 These gauges should be marked "Go" and "Not Go" respectively according to the dimensions above given.

14 Your Committee, therefore, recommends the use of the standards described above and further requests that there be deposited with the Bureau of Standards, Washington, D. C., master gauges for each size shell, the expense of such gauge to be borne by the manufacturers.

And it further recommends that these standards be known as "The American Standards" for the purpose in question.

Respectfully submitted,

Edward S. Sanderson, *Chairman.*
Wm. J. Baldwin
Stanley G. Flagg, Jr.
C. R. Hare
H. E. Harris
A. H. Moore
W. R. Webster
George B. Thomas, *Secretary*

No. 1475

THE USE OF CORRUGATED FURNACES FOR VERTICAL FIRE TUBE BOILERS

By F. W. Dean, Boston, Mass.

Member of the Society

I have been impressed for many years with the value of corrugated furnaces for vertical boilers, but only recently have actually used them.

2 By the use of such furnaces staybolts are done away with, and as there appear to be no disadvantages in the furnace this is a most important feature. As many hundreds of staybolts are avoided in each boiler there are just so many less opportunities for breakage and needed repairs. In the staybolted firebox it is necessary for safety to drill holes in the ends of the staybolts in order to know when they are broken.

3 The simplicity of vertical boilers with corrugated fireboxes must commend them to owners and makers. In the boiler shop the operations of building are of the simplest and most rapid kind.

4 This type of firebox provides for expansion and contraction of the tubes in a safe manner, but on account of its somewhat flexible character it should be assumed that it is advisable to support the lower tube plate as near the edge as practicable. The ordinary firebox is rigid vertically and supports the edge of the lower tube plate, but as the corrugated firebox has slight elasticity it is best to hold up as much of the tube plate as practicable by the tubes and provide little or no elasticity in the tube plates. The flat and unstayed portions of the upper and lower tube plates should be made equal in diameter in order to balance.

5 The flanging of the fire door presents no difficulty, but it should be done so that the corrugations coalesce with the conical part.

6 The behavior of the firebox end of the boiler when under pressure led to some speculation, for the area of the fire door opening

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers, 29 West 39th St., New York.

theoretically unbalances it. When under hydrostatic pressure, various gauges were used for showing distortion, but none could be discovered.

7 In regard to sizes of such furnaces the catalogue of the American maker gives 60 in. as the maximum inside diameter, but in fact this company can make them up to 72 in., and almost 1 in. thick. They have been made slightly larger in Germany and the furnace of the larger boiler illustrated was obtained in that country. If the inside diameter is 72 in., the grate will be 3 in. larger or 75 in. and the grate area 30.68 sq. ft. It is easy enough to generate 200 h.p. on a grate of this size with considerable capacity for forcing beyond this, and there is no difficulty in providing the heating surface for this horsepower.

8 In regard to pressure, a furnace 72 in. in diameter and 0.95 in. thick will carry 200 lb. If there were sufficient demand for larger furnaces they would probably be forthcoming. The theory of heat transmission through plates, and experience, show that thick furnaces, especially if without riveted joints, are unobjectionable.

9 The introduction of corrugated furnaces for the fireboxes of the vertical type of boiler is, I think, a real improvement in steam boilers. The type possesses the important qualities of giving maximum and permanent economy, superheating the steam from 20 deg. to 40 deg., being free from brickwork and requiring small floor space per horsepower.

10 Fig. 1 shows a boiler of the simplest possible design, two of which have been in use for a year and a half, and Fig. 2 shows one with the interior accessible for inspection and cleaning which has been in use a few months only.

DISCUSSION

W. F. MacGregor (written). In contemplating a change in any well known type of construction, it is natural to consider first its effect on that portion which experience has shown to have given the most trouble. This, in vertical fire tube boilers, is tube leakage at the crown sheet. The first question is, then, will a flexible furnace, granting that the corrugated furnace is more flexible, tend to increase or diminish tube leakage?

When the first Manning boilers were built, it was thought necessary to provide for the differential expansion between tubes and shell, and an attempt was made to do so in the OG ring, but found impracticable. The amount of differential expansion cannot be great, and it is questionable if the tubes acting through the medium of the flexible crown sheet can produce a change in length of the corrugated furnace. On the other hand, it is quite possible that

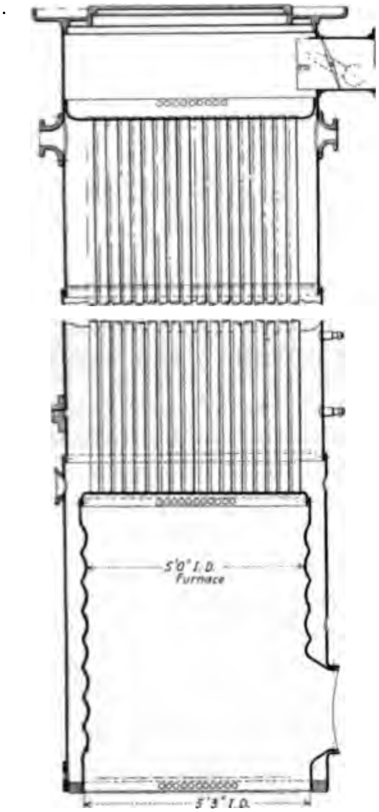

Fig. 1

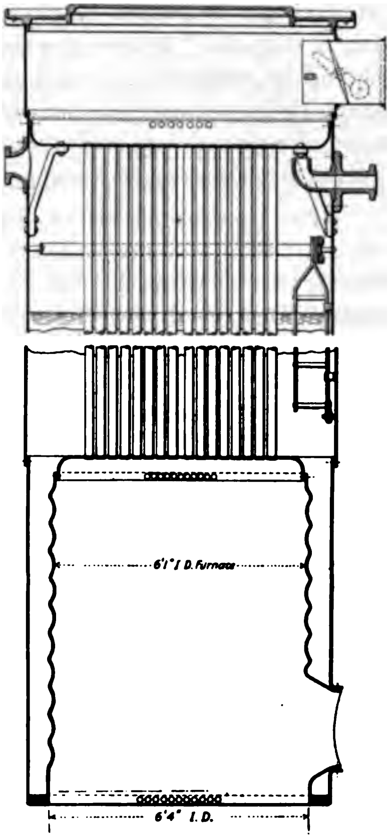

Fig. 2

Fig. 1 Simple Design of Vertical Boiler with Corrugated Furnace

Fig. 2 Larger Boiler of this Design with Interior Accessible for Inspection

the corrugated furnace may gradually change in length and tend to produce tube leakage. As to whether the tube leakage will be greater or less with a corrugated furnace can only be shown by experience.

However, the principal point in considering any new boiler construction is safety. Before abandoning the staybolted construction for other types, we should bear in mind its good points also. The staybolted surface is increased in strength by a slight deformation or bulging of the plates, while the corrugated furnace is weakened by any deformation, especially if local in character. Leakage at staybolts, or a slight bulging of the plates, is sometimes a "blessing in disguise," in giving warning of conditions in the boiler that should be corrected to avoid serious damage or disaster. Broken staybolts in stationary boilers are not so common as to condemn this type of construction, and when provided with tell-tale holes do not constitute a source of danger.

After considering safety, durability, efficiency and convenience, we must then take into account the cost of construction; and while the figures showing the cost of the corrugated furnace are not at hand, I believe it is more expensive than other types. The most apparent advantages of the corrugated furnace in the vertical fire tube boiler are decreased resistance to circulation and increased facilities for cleaning.

H. Wade Hibbard objected to the staying of the top tube sheet of the boiler shown in Fig. 2. He referred to the statements of the author that this particular boiler will carry 300 lb. pressure and that the steam is superheated to as high as 40 deg. superheat. The temperature at the top of the diagonal stay for the top tube sheet must be very much higher than the temperature of the superheated steam, which means that the top foot of the diagonal stay must certainly be at the temperature of blue heat. It is a well known fact, established by tests and experience, that in the blue heat region iron and steel are brittle, and if bent within the blue heat zone are far more liable to crack than if at a higher temperature or at a lower temperature. As the pressure in the boiler changes, the diagonal stay will be constantly subjected to bending action and there will be danger of the diagonal stay breaking at its bend.

Arthur M. Greene, Jr. Looking sideways at the illustration of this boiler, we have practically a locomotive type boiler, and I would like to ask the author to include in his closure some facts regarding the action of the Vanderbilt firebox in such boilers, because the experience of the railroads with that type of firebox would be of great value to members considering the adoption of the author's form of boiler.

FORREST E. CARDULLO. I agree with Mr. MacGregor with regard to the safety of the general form of design, and also with Professor Hibbard regarding the diagonal stay, but the latter might be very easily taken care of by carrying the boiler, in a conical form, from the joint flush with the crown sheet of the firebox up to the top tube sheet, so as to eliminate entirely the necessity of staying that part of the boiler.

I see no reason why anyone should build a 200 h.p. vertical tubular boiler. Such a boiler, in my experience, invariably causes stack temperatures of 800 or 1000 deg., which means that it is inherently a wasteful form of apparatus, unless provision is made for utilizing the waste heat.

CHAS. H. MANNING (written). Mr. Dean's paper sets forth the latest improvement in vertical boilers, which is certainly a great step forward and one which I have desired to see taken for a long time, as by eliminating staybolts it overcomes the greatest cost of repair on this type of boiler.

Mr. Cardullo loses sight of the fact that the temperature of the gases leaving a tube depends not on the position (vertical or horizontal) of the tube, but on the ratio of diameter to length of tubes, which seldom exceeds 1 to 60 with horizontal tubes but can readily reach 1 to 90 or over with vertical tubes and at which ratio the gases are reduced to 600 deg. fahr. or less.

Long horizontal tubes necessitate increased floor space, an expensive item in a city plant. Vertical tubes do not need large floor space.

THE AUTHOR. There should be no skepticism as to the safety and proper action of corrugated furnaces in the vertical position after their use by thousands at sea for nearly forty years in the horizontal position.

Mr. MacGregor mentions tube leakage in vertical boilers, but this cannot be influenced by corrugated furnaces, and in general it is not a troublesome matter with such boilers. The lower ends of tubes are rather sensitive to dirt on the crown sheet, but all types of boilers are somewhere sensitive to dirt, and the lesson to be drawn is that dirt should be kept out of boilers.

Professor Hibbard is mistaken in saying that the boilers shown would carry 300 lb. pressure, 200 lb. being the pressure given. Ordinarily the superheat is not over 25 deg., and no boiler of my design has superheated the steam 40 deg. This superheat has, however, been reached in another type of vertical boiler in everyday use, in which, however, there are no braces at the top. The boilers have been in use for several years without any trouble being apparent.

Answering Professor Greene, I never have been familiar with the working of the corrugated furnaces of Vanderbilt boilers, but it is generally understood that horizontal boilers with such furnaces have given trouble for want of circulation. The bottoms of such boilers are likely to be cold unless circulation is at least artificially started.

Mr. Cardullo considers wrongly that vertical boilers send off their gases hotter than a good boiler should. There is no reason why a vertical tube should not absorb heat furnished to it; and if coal is burnt to carbonic acid there is no better way to absorb the heat generated than by a collection of vertical tubes. They keep cleaner than other tubes and this is an advantage. That the gases pass off somewhat hotter than those from horizontal boilers is true, but the reason for this is that there is no way for cold air to cool them off, as through cracks in brickwork, and they escape at a normal temperature. This is beneficial to economizers where used. If gases are very much too hot from a properly designed vertical fire tube boiler it is due to bad firing, the formation of carbonic oxide and its combustion in the smoke box or flue. In textile mills in New England it is well known that the most economical plants in coal consumption are those which are equipped with vertical fire tube boilers.

In regard to size of boilers of this type there are many that are rated at 500 h.p., and some of those are habitually operated at 100 per cent or more above their rating.

Table 1, summarizing the results of a number of tests of vertical firetube boilers, gives the temperature of the escaping gases.

Table 1. Tests on Vertical Fire Tube Boilers Showing Temperature of Flue Gases

Year	Location	Grate Area, sq. ft.	Water heating surface and total including super. surface, sq. ft.	Steam pressure, lb. per sq. in.	Boiler h. p. developed	Kind of coal	Coal burned per sq. ft. of grate per hr., lb.	Evaporation per hr. per sq. ft. heating surface from and at 212° F., lb.	Evaporation from and at 212° F., lb. per lb. coal	Evaporation from and at 212° F., lb. per lb. combustible	Boiler h. p. per sq. ft. of grate surface	Sq. ft. of heating surface per h. p.	Temperature of escaping gases, deg. fahr.	Test by	How Fired
Unknown		28.3	1260 & 1700	46	243	Cumberland	13.1	3.2[1]	11.34	12.29			427	Barrus	Hand
1897	Grosvenordale Mills	33.2	1489 & 2154	153	123	Cumberland	11.7	2.86[1]	10.94	11.96	3.72	12[1]	520	Barrus	Hand
1894	Whitman Mills	28.27	1388 & 1869	123	194	Pocahontas	23.0	4.90[1]	10.07	12.00	6.85	7[1]	531	Denton	Hand
1894	Whitman Mills	28.27	1388 & 1869	123	126	Pocahontas	14.7	3.10[1]	10.38	12.32	4.47	11[1]	496	Denton	Hand
1894	Whitman Mills	28.27	1388 & 1869	156	95	Pocahontas	10.9	2.30[1]	10.62	12.35	3.38	15[1]	362	Denton	Hand
1894	Manchester W. W.	28.27	1388 & 1869	142	98	Cumberland	11.0	2.44[1]	10.89	11.92	3.47	14[1]	380[3]	Dean	Hand
1894	Manchester W. W.	28.27	1388 & 1869	140	103	Cumberland	12.64	2.56[1]	9.92	11.11	3.64	13.3[1]	397[3]	Dean	Hand
1894	Narragansett Mills	33.4	1384 & 2036	114	119	Pocahontas	11.30	2.90[1]	11.19	11.92	3.57	11.6[1]	395	Dean	Hand
1894	Narragansett Mills	33.4	1492 & 2036	115	162	Pocahontas	15.20	3.69[1]	10.02	11.83	4.85	9.2[1]	439	Dean	Hand
1894	Kunhardt's Mill	28.2	1745, total	67	115	Cumberland	12.79	2.28[2]	10.96	11.64	4.09	15.10[2]	491	Dean	Hand
1894	Kunhardt's Mill	28.2	1745, total	69	102	Cumberland	11.50	2.02[2]	10.82	11.48	3.62	17.1[2]	519	Dean	Hand
1895	Berkeley Mill	33.4	1384 & 2037	129	103	Pocahontas	10.37	2.57[1]	10.25	11.12	3.09	13.4[1]	437	Dean	Hand
1895	Fletcher Mfg. Co.	33.4	1897 & 2431	79	226	Pocahontas	21.56	4.11[1]	10.83	11.50	6.77	8.4[1]	422	Dean	Hand
1895	Fletcher Mfg. Co.	33.4	1897 & 2431	79	177	Cumberland	17.63	3.21[1]	10.35	11.11	5.30	10.7[1]	414	Dean	Hand
1909	Cumberland Mills	76.5	4900 & 6081	130	1005	New River	39.10	7.05	10.34	11.50	13.14	4.87[1]	609	Dean	Chain g'te stkr
1909	Cumberland Mills	76.5	4900 & 6081	127	788	New River	31.30	5.55	10.46	11.36	10.30	6.22[1]	559	Dean	Chain g'te stkr
1910	Cumberland Mills	76.5	4900 & 6081	123	757	New River	30.5	5.33[1]	11.20	12.77	9.90	6.46[1]	561	Chase	Chain g'te stkr
1910	Cumberland Mills	76.5	4900 & 6081	116	830	New River	33.4	5.85[1]	11.22	12.62	10.85	5.90[1]	573	Chase	Chain g'te stkr
1898		28.27	1347 & 1796	160	110	Pocahontas	11.2	2.42	12.09	12.76	3.88	12.24	418	Manning	Hand
1900		52.00	2498 & 3393	107	350	Cumberland	20.8	4.84	11.17	12.37	6.73	7.13	742	Manning	Roney stoker
1897		28.30	1260 & 1700	99	137	Pocahontas	13.6	3.10	11.89	12.74	4.66	9.54	522	Barrus	Hand
1901		38.5	2174 & 2852	115	192	Cumberland	15.89	3.04	10.57	12.47	4.98	11.[illegible]	368	Green	Hand
1894	Bristol Mfg. Corp.	169.62	8298 & 11124	125	747	Pocahontas	13.71	3.19	10.80	13.09	4.40	11.10	510	Denton	Hand
1894	Bristol Mfg. Corp.	226.16	11064 & 14832	124	797	Pocahontas	10.42	2.48	11.09	13.41	3.52	13.88	467	Denton	Hand
1894	Bristol Mfg. Corp.	197.89	9681 & 12978	125	715	Pocahontas	11.01	2.55	10.75	13.02	3.61	13.53	460	Denton	Hand

[1] Excluding superheating surface.
[2] Including superheating surface.
[3] Leaks in smokebox covers.

No. 1476

A COMPARISON OF THE PROPERTIES OF A NICKEL, CARBON AND MANGANESE STEEL BEFORE AND AFTER HEAT TREATMENT

By Robert R. Abbott,[1] Cleveland, O.

Non-Member

The effect of small quantities of manganese upon the physical properties of annealed steel is fairly well known. Up to about 2 per cent, each 0.01 per cent of manganese increases the tensile strength by about 160 lb. per sq. in. Its effect upon the reduction in area and elongation is very small, slightly increasing the reduction and lowering the elongation. From about 2½ per cent to 7 per cent manganese makes steel extremely brittle and above 7 per cent this effect disappears and we again have a useful alloy. An 11 per cent alloy has a wide range of use in the cast form for crossing frogs, rolls, gears, etc. It is usually finished by grinding, as it is nearly impossible to machine.

2 We can classify the average commercial steels made in this country with carbon less than 0.50 per cent into two groups; (*a*) Those with manganese contents approximately 0.40 to 0.50 per cent; this group includes the ordinary carbon steel and chrome-nickel steel. (*b*) Those with manganese contents approximately 0.60 to 0.70 per cent; this group includes nickel and chrome-vanadium steels.

3 I do not mean the above as a hard and fast classification, but it represents fairly well commercial practice. English steels fall readily into the same classification, but with German steels the grouping is not so well defined.

4 It is rare to find a steel in this country with manganese above

[1]Metallurgical Engineer, The Peerless Motor Car Co.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

1 per cent (and below 2 per cent). Abroad a considerable amount of steel is used, particularly for frames of automobiles, with manganese varying from 1.25 to 1.75 per cent.

5 Very little has been published regarding the heat treatment of these high manganese steels and I am therefore presenting a comparison of the effect produced upon the physical properties of three steels of about the same carbon contents. One of these is a plain carbon steel, another a nickel steel, and the third a manganese steel, containing 1.61 per cent manganese.

6 Their analysis is as follows:

	Carbon steel	Nickel steel	Manganese steel
Carbon	0.342	0.336	0.341
Phosphorous	0.014	0.019	0.047
Sulphur	0.029	0.019	0.025
Manganese	0.54	0.55	1.61
Silicon	0.030	0.188	0.009
Nickel	0.0	3.17	0.0
Chrome	0.0	0.0	0.0
Vanadium	0.0	0.0	0.0
Copper	0.0	0.05	0.02

7 In these three steels the upper critical temperatures were first determined. Test bars ¾ in. in diameter and 4½ in. long were next machined from each steel. One of each kind was annealed by heating in lead to a temperature 5 deg. fahr. above the upper critical temperature, holding there about ten minutes, and allowing to cool slowly until the lead solidified. The time of this cooling was about ten hours. The lead was then heated slowly to a temperature of 800 deg. fahr. and the test bars removed. They were then threaded and machined to a standard 2 in. test specimen and ground to a diameter of 0.505 in. (1/5 sq. in.). One end was left longer than the other to allow for hardness tests. They were then pulled in a tensile machine with the following results:

Steel	Elastic	Maximum	Reduction	Elongation	Brinell hardness
Carbon	36,600	67,250	51.0	32.0	120
Nickel	55,000	81,850	59.0	31.2	153
Manganese	61,150	87,850	58 5	29.9	150

8 From these figures we see that the manganese steel is slightly stronger than the nickel steel and has practically the same amount of "toughness" as shown by the reduction in area.

9 For the heat-treated specimens a test bar of the same size was used. The heating was all done in lead and was controlled by a Leeds & Northrup resistance pyrometer. The desired temperature was reached slowly and maintained as nearly constant as possible for ten minutes. All tests which were to be made at the same temperature were made simultaneously. The furnace contained half a ton of lead and therefore the temperature could be kept very uniform. The bars were quenched in water and were drawn to the desired temperature in a lead furnace holding about three tons of lead. All of the bars which were to be drawn at the same temperature were drawn simultaneously. The desired temperature was maintained constant for thirty minutes. The large mass of lead made this an easy matter. After treatment the bars were machined and then ground to a diameter of 0.505 in. and were pulled in a tensile machine with an autographic recording device. Hardness tests were made on the long end of the test bar after sawing off and grinding to a flat surface. The following determinations were made:

Elastic limit
Maximum strength
Reduction in area
Elongation
Brinell hardness
Scleroscope hardness
Rupture stress
Energy necessary to cause fracture.

10 The last three determinations are not considered in this article.

11 Test bars were treated from above the upper critical temperature and were then drawn to the following temperatures: 300 500, 700, 800, 900, 1000, 1200, 1300 deg. fahr.

12 In Fig. 1 are plotted the results of these tests for the three steels for the elastic limit and maximum strength. In Fig. 2 the results are given for the reduction in area and elongation, and in Fig. 3 are given the brinell hardness results.

13 From these charts it can be concluded that practically the same results can be obtained, as far as strength is concerned, by the heat treatment of a 1.6 per cent manganese steel as for a nickel steel of practically twice the per cent of nickel. However, since the nickel steel also contains 0.55 per cent manganese, we actually have an excess of only 1.06 per cent manganese which apparently has the same effect as about three times the same amount of nickel.

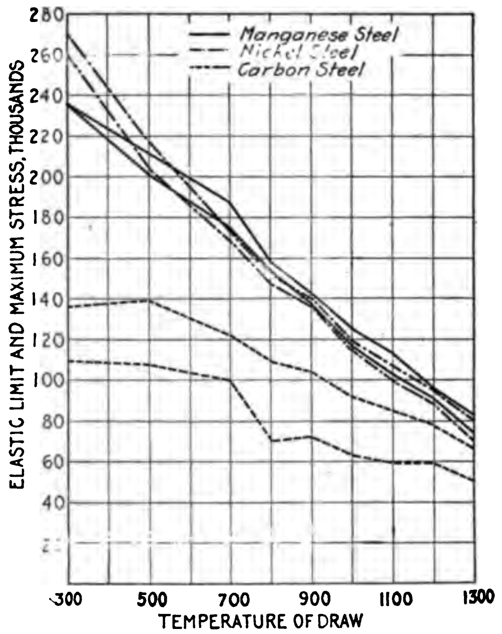

Fig. 1 Tests on Effect of Heat Treatment on Maximum Stress (Upper Curves) and Elastic Limit (Lower Curves)

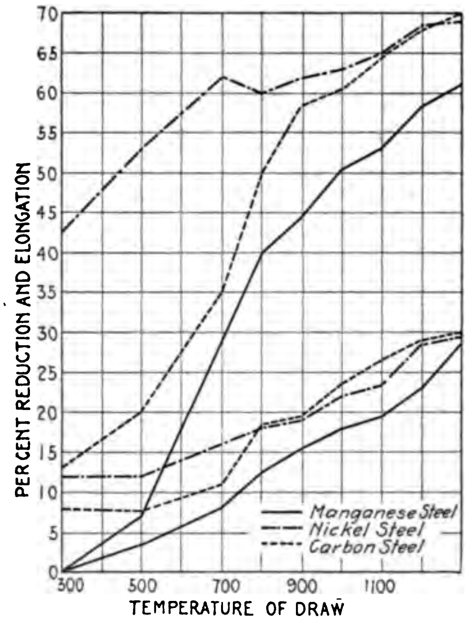

Fig. 2 Tests on Effect of Heat Treatment on Reduction in Area (Upper Curves) and Elongation (Lower Curves)

14 Now, regarding the "toughness" which for the sake of comparison can be considered as being measured by the reduction in area, we see that the manganese steel does not compare so favorably with the carbon or nickel steel for the same temperature of draw. Evidently also, while manganese increases the reduction in area in annealed steels, it has the opposite effect in heat-treated steels.

15 The following equations represent fairly well the average values of the elastic limits, maximum strengths, reduction in area

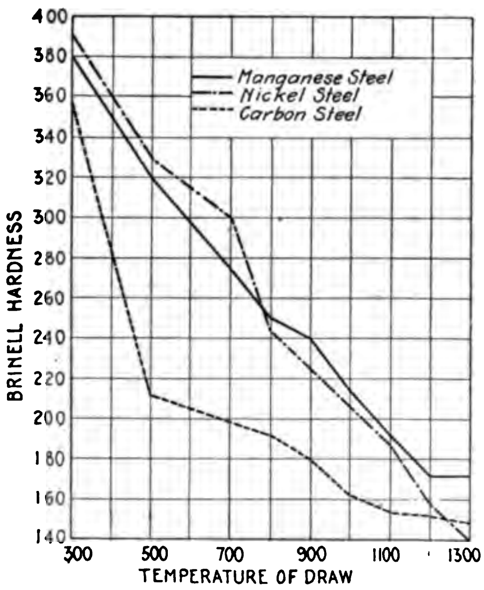

FIG. 3 TESTS ON EFFECT OF HEAT TREATMENT ON BRINELL HARDNESS

and elongation of the three steels.

E = elastic limit in pounds per square inch.
M = maximum stress in pounds per square inch.
r = reduction in area in per cent.
e = elongation in per cent.
T = temperature of draw in degrees fahrenheit.

For manganese steel

$$E = 284{,}000 - 163\,T$$
$$M = 288{,}000 - 159\,T$$

$$r = -19 + .068\,T$$
$$e = -10 + .028\,T$$

For nickel steel

$$E = 302{,}000 - 183\,T$$
$$M = 314{,}000 - 188\,T$$
$$r = 40 + .024\,T$$
$$e = 3.5 + .018\,T$$

For carbon steel

$$E = 134{,}000 - 66\,T$$
$$M = 170{,}000 - 77\,T$$
$$r = -5.8 + .063\,T$$
$$e = -3 + .026\,T$$

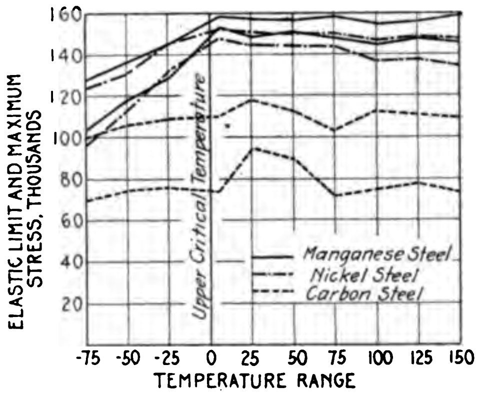

FIG. 4 TESTS ON EFFECT OF OVER- AND UNDER-HEATING ON MAXIMUM STRESS (UPPER CURVES) AND ELASTIC LIMIT (LOWER CURVES)

16 The next determination was upon the effect of over- and under-heating during the quenching process. For this purpose ten bars of each steel were quenched as follows: One 5 deg. above the upper critical point, six others at 25, 50, 75, 100, 125 and 150 deg. above and three at 25, 50 and 75 deg. below the critical point. These were all drawn to a temperature of 800 deg. fahr. and the regular tests conducted upon them. The results of these tests are plotted in Fig. 4 which gives the elastic limit and maximum strength, and in Fig. 5 which shows the reduction and elongation.

17 These tests show that there is very little difference between the nickel and manganese steels as far as over- and under-heating is concerned.

18 Briefly summarizing the above results: For a heat-treated 1½ per cent manganese steel the manganese in excess of that contained in a nickel steel of a corresponding carbon contents (about 0.34 per cent) exerts a strengthening effect equivalent to about three times the same amount of nickel.

19 While the manganese effect upon a steel which has not been heat-treated is to increase the toughness slightly, its effect upon a

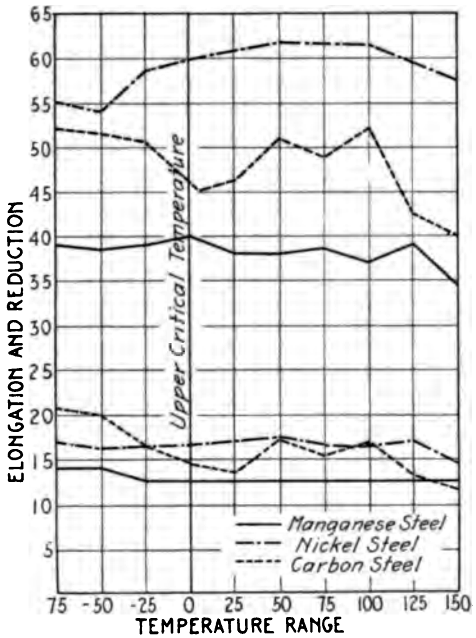

FIG. 5 TESTS ON EFFECT OF OVER- AND UNDER-HEATING ON REDUCTION IN AREA (UPPER CURVES) AND ELONGATION (LOWER CURVES)

heat-treated steel is decidedly the reverse. In the case of nickel the effect upon an untreated steel is practically zero, while in a heat-treated steel nickel increases the toughness decidedly.

20 An untreated steel containing about 1½ per cent manganese is fully as tough, and is stronger than a nickel steel of about 3¼ per cent nickel.

DISCUSSION

HENRY M. HOWE[1] (written). The manganese steel to which Mr. Abbott calls attention has come into use in this country more

[1]Columbia University, New York.

widely than might be inferred from his remarks. The late Maunsell White developed a steel of over 1 per cent manganese and somewhat lower in carbon than the author's, and this has gone into very wide use where high quality is needed.

Roughly speaking, 1 per cent of manganese is about equivalent to 2 per cent of nickel; at least, 1 per cent of manganese accomplishes some of the more important things which double the quantity of nickel accomplishes.

The reason why manganese and nickel are useful for such steels is a very simple one. In order to develop the properties of a given steel very highly it should be heated above the transformation range, to cause the usual rather coarse masses of ferrite to become reabsorbed. The steel should then be cooled rapidly, lest in slow cooling through the transformation range the ferrite should again form coarse masses.

If this cooling is done by quenching in water or oil the resultant steel is too brittle, that is to say, the chemical transformation is arrested and at the same time serious internal stresses are set up. In order to permit the transformation to complete itself, so that the chemical brittleness may be removed, and in order to relieve the stresses which are also a cause of brittleness, the steel must next be reheated, as in tempering or moderate annealing. The rise of temperature enables the chemical transformation to complete itself, so that the metal becomes transformed into ferrite and cementite. It also releases the stresses. But this is accompanied by an incidental damage, namely that the resultant ferrite coalesces more and more into larger and larger masses, and with the increase in the size of these masses the quality of the steel falls off progressively.

The advantage of manganese and nickel is that they cause this coalescence and coarsening to occur very slowly. As a consequence, when the steel is reheated so that the transformation occurs, removing the chemical brittleness, and when the stresses are removed, thus removing the second cause of brittleness, the coalescence of the ferrite is very much slower than in steel with less manganese. And in general this same sluggardizing effect of manganese and nickel under miscellaneous conditions gives rise to a finer structure than would otherwise form. Their effect in this respect is like that of vanadium, only less powerful.

No. 1477

A BASIS FOR RATIONAL DESIGN OF HEAT TRANSFER APPARATUS

By E. E. Wilson,[1] New York, N. Y.

Non-Member

An examination of the literature on the subject of heat transfer shows such a wide variation in the coefficients established that it is difficult to make a choice for use in practice. The fact that the results of careful experimenters do not agree leads to the conclusion that some variable or variables have been neglected, and the fact that for similar flow conditions, the rate of heat transfer in feed water heaters is consistently greater than that in the similar apparatus, the condenser, indicates that one such neglected variable is the temperature of the circulating water. It is the purpose of this paper to apply a correction for the water temperature to the results of reliable experiments, thereby reconciling not only the results of an individual experimenter but those of different experimenters as well, and to utilize the results as a basis for rational design.

2 The rate of heat transfer has often been related to the circulating water velocity and is generally expressed as some exponential function of the velocity. By correcting for all variables, this relation may be expressed in a different form as a straight line function. With this relation it is possible to evaluate the thickness of a water film capable of offering the same resistance to heat transfer as is encountered on the water side of a condenser or feed water heater. From the known internal conductivity of such a film it is possible to develop an expression for the area of heating surface required to transmit a given quantity of heat under all conditions of circulating water velocity and temperature, mean temperature difference, and tube diameters. With the use of suitable design factors this expression for area may form the basis of rational design, while a consideration of the

[1]Lieutenant, U. S. N.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

manner in which the variables appear may be of assistance to the operator.

3 The possibility of solution along the lines indicated was brought to the author's attention by Dr. C. E. Lucke of Columbia University in the course of instruction for officers of the Navy doing post-graduate work in engineering under his direction. Professor Lucke pointed out that the resistance to heat transfer in such apparatus is probably due to a film of gas or liquid on the tube walls; that the magnitude of each resistance depends upon the thickness of the film, and that the thickness of the film depends upon the amount of scrubbing action due to the velocity as well as the fluid temperature. The problem then is to locate the controlling resistance and by reducing the resistance increase the heat transfer.

4 In condensers and feed-water heaters the resistances to heat flow for clean tubes are:

a That due to the thickness of the tube walls.
b That due to the film of water on the steam side next the tubes.
c That due to air in the exhaust steam.
d That due to the water film on the water side.

5 Of these, (*a*) is ordinarily negligible, (*b*) may be quite large and (*c*) may be the controlling resistance if the quantity of air is large. If the velocity of the circulating water be zero and we consider the steam to be air free, then (*d*) becomes the controlling resistance because here the film thickness is the radius of the tube and is therefore much greater than (*b*). Under ordinary conditions of operation (*d*) will be the controlling resistance and any increase in heat transfer should come through reduction of the film thickness on the water side. This is evident when the general law of heat transfer is considered. Taking the analogy of the electric or magnetic circuit we may write,

$$U = t_m \div R$$

where U = B.t.u. transmitted per hour per sq. ft.
t_m = mean temperature difference between the hot and cold fluids
R = resistance to transfer

Putting in the individual resistances enumerated above this becomes

$$U = \frac{t_m}{R_w + R_a + R_s + R_t}$$

in which

R_w = resistance of the water film on the water side

R_a = resistance of the air film on the steam side
R_s = resistance of the water film on the steam side
R_t = resistance of the tube walls

6 If now we let ρ be the resistance per unit of film thickness and l. the thickness of the film the expression becomes

$$U = \frac{t_m}{\rho_w l_w + \rho_a l_a + \rho_s l_s + \rho_t l_t}$$

in which we have the heat transfer in terms of the film thicknesses involved.

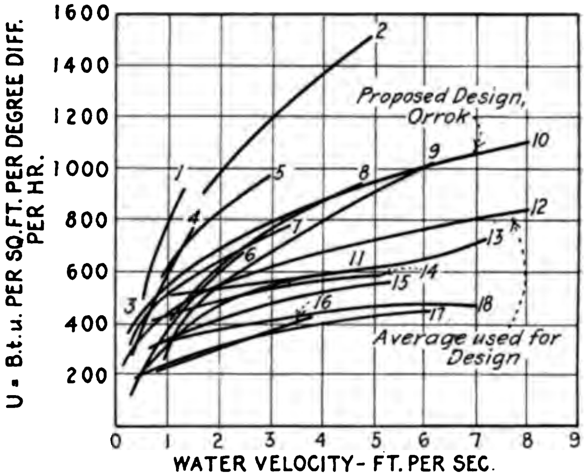

FIG. 1 COMPARISON OF VALUES OF U FOR SURFACE CONDENSERS

7 Of the resistances above enumerated, R_a, R_s and R_t are, for modern condenser conditions, quite constant. R_w, however, is a variable and a function of first the water velocity and second the mean temperature of the water. Any attempt to relate the rate of heat transfer to the water velocity alone is likely to give discordant results, and that this is the case is shown by Fig. 1, taken from a paper by George A. Orrok[1], summing up the investigations of many experimenters. Again, we see that the coefficient of heat transfer varies inversely as the sum of a variable and a constant and therefore cannot be expressed as a simple exponential function of one of the things upon which the variable depends,—water velocity. It so happens that such an exponential function of the mean tem-

[1] Trans. Am. Soc. M. E., Vol. 32, p. 1139.

perature difference or the water velocity may represent fairly enough the results of individual investigations, but it cannot be a rational general law. If we wish to use the circulating water velocity as a prime variable, we must first correct the velocity by reducing it to some standard temperature. This done, we can express the results in the form of resistances and obtain the general law desired.

THE CORRECTION FOR VISCOSITY

8 In his Scientific Papers Osborne Reynolds pointed out that if a fluid traversed a tube without turbulence, the resistance to the transfer of heat would be entirely independent of the velocity; while if the flow were turbulent, it would vary inversely as the velocity. In a later paper he showed that the transition from non-turbulent to turbulent flow was a discontinuous phenomenon. Up to a certain critical velocity of flow a fluid moves through a tube in parallel layers and without turbulence; but on this velocity being exceeded, turbulence sets in and does so abruptly. The law of loss of head in the tube changes abruptly from being proportional to the velocity to being proportional to the square of the velocity. Hence, when heat is being transferred from a hot tube to a fluid flowing in it, the law of heat transference changes abruptly once the critical velocity is reached.

9 The critical velocity according to Reynolds is given by the relation

$$V_{\text{critical}}=\frac{1}{B}\cdot\frac{P}{d}$$

where

P = Poiseuille's value for the ratio of viscosity to density
d = internal diameter of the tube
B = a constant

The value of P in centigrade units is given as

$$\frac{1}{1+0.0336T+0.000221T^2}$$

in which T is the temperature centigrade.

10 If now the value of P at some one temperature such as 60 deg., be settled upon as the standard, the ratio of P at any other temperature to that at the standard temperature may be computed as a specific viscosity and this has been done in Table 1. That portion of this table from 40 deg. to 103 deg. was taken from a paper in Engineering of Jan. 23, 1914, and the extension was made from the table of viscosi-

ties and densities in the Smithsonian Physical Tables. The whole is plotted in Fig. 3.

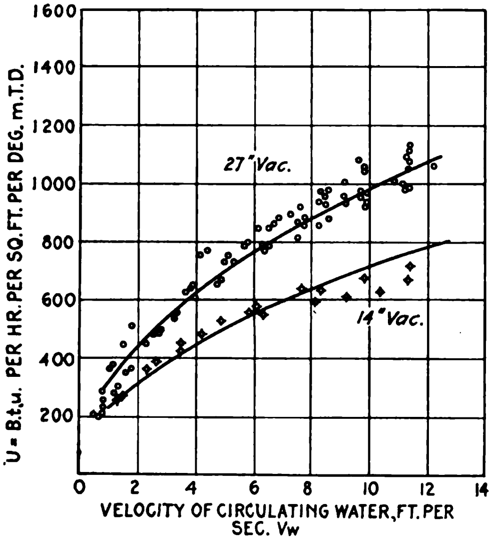

FIG. 2 HEAT TRANSFER COEFFICIENT AS A FUNCTION OF WATER VELOCITY FOR STEAM CONDENSERS, SHOWING CURVES OF ACCEPTED LAW AND EXPERIMENTAL POINTS BY ORROK

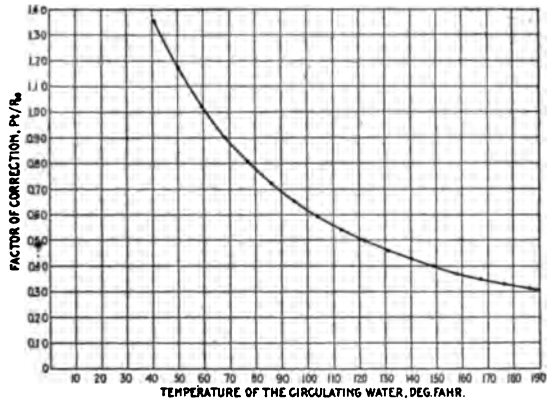

FIG. 3 CURVE OF FACTORS OF CORRECTION FOR CHANGE IN VISCOSITY WITH CHANGE IN TEMPERATURE OF THE CIRCULATING WATER

11 The view that the rate of heat transfer should vary inversely with the viscosity was arrived at through consideration of the similarity between the accepted law for heat transfer and that for the loss

TABLE 1 FACTORS OF CORRECTION FOR VISCOSITY

Temperature	Correction factor	Temperature	Correction factor	Temperature	Correction factor	Temperature	Correction factor
32	1.60	78	0.79	124	0.49	170	0.34
33	1.57	79	0.78	125	0.49	171	0.34
34	1.54	80	0.77	126	0.48	172	0.33
35	1.52	81	0.76	127	0.47	173	0.33
36	1.49	82	0.76	128	0.47	174	0.33
37	1.47	83	0.75	129	0.47	175	0.33
38	1.44	84	0.74	130	0.46	175	0.33
39	1.41	85	0.73	131	0.46	177	0.32
40	1.38	86	0.72	132	0.46	177	0.32
41	1.36	87	0.71	133	0.45	178	0.32
42	1.33	88	0.70	134	0.45	179	0.32
43	1.31	89	0.69	135	0.44	180	0.32
44	1.29	90	0.69	136	0.44	181	0.32
45	1.27	91	0.68	137	0.44	182	0.32
46	1.25	92	0.67	138	0.43	183	0.32
47	1.23	93	0.67	139	0.43	184	0.31
48	1.21	94	0.66	140	0.43	185	0.31
49	1.19	95	0.65	141	0.43	186	0.31
50	1.17	96	0.65	142	0.42	187	0.31
51	1.15	97	0.64	143	0.42	188	0.31
52	1.14	98	0.63	144	0.42	189	0.31
53	1.12	99	0.62	145	0.41	190	0.30
54	1.10	100	0.62	146	0.41	191	0.30
55	1.09	101	0.61	147	0.41	192	0.30
56	1.07	102	0.60	148	0.40	193	0.30
57	1.05	103	0.59	149	0.40	194	0.29
58	1.04	104	0.59	150	0.39	195	0.29
59	1.02	105	0.58	151	0.39	196	0.29
60	1.00	106	0.58	152	0.39	197	0.29
61	1.00	107	0.58	153	0.38	198	0.29
62	0.98	108	0.57	154	0.38	199	0.29
63	0.97	109	0.57	155	0.38	200	0.29
64	0.95	110	0.56	156	0.37	201	0.29
65	0.94	111	0.56	157	0.37	202	0.28
66	0.93	112	0.55	158	0.37	203	0.28
67	0.92	113	0.54	159	0.37	204	0.28
68	0.90	114	0.54	160	0.36	205	0.28
69	0.89	115	0.53	161	0.36	206	0.28
70	0.88	116	0.53	162	0.36	207	0.28
71	0.87	117	0.52	163	0.36	208	0.28
72	0.85	118	0.52	164	0.35	209	0.28
73	0.84	119	0.51	165	0.35	210	0.27
74	0.83	120	0.51	166	0.35	211	0.27
75	0.83	121	0.50	167	0.35	212	0.27
76	0.82	122	0.50	168	0.35	....	....
77	0.81	123	0.50	169	0.34	....	....

of head in pipes. The curves shown in Fig. 1 for heat transfer have the equation of the form $U = kV^n$, in which n varies from 1/3 to 1/2. The accepted law for loss of head in pipes is familiar as $h = kV^n$, in which n is about 2. In these expressions U is the rate of heat transfer, h the loss of head per unit of length, and V the velocity of the water in feet per second. This similarity of form led to a study of the law for loss of head, which had been determined by Osborne Reynolds and reported in the Philosophical Transactions of the Royal Society of London in 1883. Reynolds found that up to the critical velocity the loss of head varied directly with the velocity as shown in the lower branches of two of his results in Fig. 4. The coördinates here are the

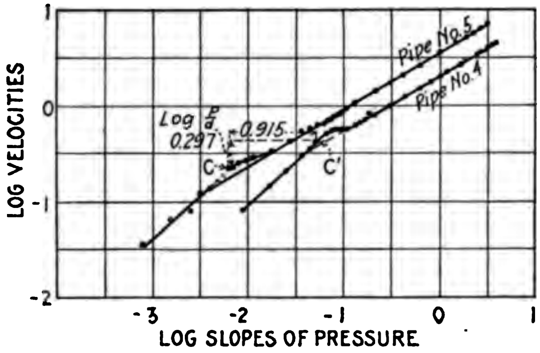

FIG. 4 VARIATION OF LOSS OF HEAD IN PIPES AS REPORTED BY OSBORNE REYNOLDS, PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY OF LONDON, 1883

logarithms of the loss of head and velocity, and the slope of the lower branches is 45 deg. At the critical points C and C' the flow becomes turbulent and the slope of the curve changes to about 1.72. Reynolds shows further that the temperature of the water and the diameter of the tubes have no effect upon the slope of the curves, but shift the position of the critical point instead. The component of this shift in the direction of the velocity is 0.297 on the figure and this corresponds very closely to log. P/d in the expression for critical velocity above.

12 With these facts in mind the results for heat transfer were investigated and those reported by George A. Orrok were selected as being the most complete and accurate. The mean temperature of the circulating water was computed from his results as shown in Table 2

TABLE 2 CALCULATIONS FOR REDUCTION OF DATA ON CONDENSERS

Reported by George A. Orrok, Transactions Am.Soc.M.E., Volume 32.

1 Velocity of water in tube ft. per sec. V_w	2 Temperature of steam deg. fahr. t_s	3 Mean temperature difference deg. fahr. t_m	4 Mean water temperature deg. fahr. t_w	5 Correction for viscosity C	6 Velocity of water reduced to 60 deg. fahr. V_{60}	7 Corrected velocity of water referred to 5/8-in. tube. $\overline{V}$	8 Reciprocal of column 7 $\frac{1}{\overline{V}}$	9 Column 8 to 0.82 power $\left(\frac{1}{\overline{V}}\right)^{0.82}$	10 B.t.u. per hr. per sq. ft. per deg. mean temperature difference U	11 $\frac{1000}{U}$ R	12 Test number No.
11.2	126.3	72.2	54.1	1.10	9.3	15.85	0.063	0.102	1000	1.00	548
9.88	126.3	71.5	54.8	1.09	9.07	15.45	0.065	0.106	967	1.03	549
9.40	126.1	71.8	54.3	1.10	8.61	14.68	0.068	0.109	945	1.06	550
9.13	126.3	72.1	54.2	1.10	8.32	14.15	0.071	0.111	938	1.07	551
7.62	126.6	69.2	57.4	1.05	9.26	15.80	0.063	0.102	864	1.16	552
6.46	126.4	71.5	54.9	1.09	5.93	10.10	0.099	0.133	788	1.27	553
5.44	127.0	70.9	56.1	1.07	5.08	8.60	0.116	0.170	722	1.39	554
4.02	127.1	69.9	57.2	1.05	3.83	6.48	0.154	0.206	618	1.62	555
2.66	126.6	65.8	60.8	1.00	2.66	4.51	0.221	0.290	492	2.03	556
1.20	127.0	65.7	61.3	1.00	1.20	2.03	0.495	0.560	273	3.66	557
11.4	124.9	70.6	54.3	1.10	10.35	17.50	0.057	0.095	1076	0.93	558
10.1	126.1	70.5	55.6	1.07	9.43	16.05	0.062	0.100	1008	0.99	559
7.63	126.3	69.1	57.2	1.05	7.26	12.35	0.081	0.126	920	1.086	560
9.17	125.4	65.8	59.6	1.01	9.07	15.45	0.065	0.106	1005	0.995	561
6.76	125.1	71.5	53.6	1.10	6.15	10.50	0.095	0.142	862	1.160	562
5.07	126.4	73.6	52.8	1.12	4.52	7.70	0.130	0.185	732	1.370	563
3.94	126.7	67.4	59.3	1.02	3.86	6.57	0.152	0.210	640	1.560	564
2.65	126.8	66.2	60.6	1.00	2.65	4.52	0.221	0.290	507	1.970	565
1.24	127.0	65.6	61.4	1.00	1.24	2.11	0.474	0.545	283	3.500	566
11.26	126.5	73.6	52.9	1.12	10.05	17.10	0.058	0.096	1040	0.960	567
9.70	125.4	71.5	53.9	1.10	8.82	15.00	0.066	0.106	950	1.050	568
9.12	125.0	71.0	54.0	1.10	8.30	14.10	0.071	0.113	947	1.060	569
7.32	126.0	70.3	55.7	1.08	6.79	11.55	0.087	0.132	895	1.120	570
6.27	126.5	71.9	54.6	1.09	5.76	9.80	0.102	0.155	842	1.19	571
4.28	126.9	70.9	56.0	1.07	4.00	6.80	0.147	0.205	760	1.32	572
4.43	126.0	70.2	55.8	1.07	4.13	7.03	0.142	0.200	769	1.30	573
11.40	127.5	71.2	56.3	1.07	10.65	18.10	0.055	0.092	1132	0.88	574
9.65	126.4	68.8	50.6	1.04	9.29	15.80	0.063	0.102	1080	0.93	575
8.67	125.6	71.3	54.3	1.10	7.89	13.40	0.075	0.117	980	1.02	576
8.52	126.4	70.7	55.7	1.07	7.97	13.55	0.074	0.115	963	1.04	577
6.98	126.0	69.4	56.6	1.06	6.58	11.20	0.089	0.135	883	1.13	578
4.01	126.1	68.7	57.4	1.05	3.82	6.48	0.154	0.207	647	1.55	579
5.82	125.5	71.5	54.0	1.10	5.29	8.97	0.111	0.162	792	1.26	580
2.67	126.6	69.6	57.0	1.05	2.54	4.32	0.231	0.300	487	2.05	581
0.86	127.4	68.0	59.4	1.01	0.85	1.45	0.689	0.740	195	5.12	582
11.3	125.5	74.6	50.9	1.15	9.82	16.70	0.060	0.099	1090	0.92	583
11.4	125.9	71.9	54.0	1.10	10.35	17.65	0.056	0.093	1110	0.90	584
9.8	126.0	71.8	54.2	1.10	8.91	15.15	0.066	0.107	1056	0.95	585
8.33	125.3	70.7	54.6	1.08	7.71	13.12	0.076	0.118	973	1.03	586
8.30	124.8	73.5	51.3	1.15	7.22	12.25	0.082	0.128	943	1.06	587

TABLE 2—Continued

1	2	3	4	5	6	7	8	9	10	11	12
Velocity of water in tube ft. per sec.	Temperature of steam deg. fahr.	Mean temperature difference deg. fahr.	Mean water temperature deg. fahr.	Correction for viscosity	Velocity of water reduced to 60 deg. fahr.	Corrected velocity of water referred to ⅝-in. tube.	Reciprocal of column 7	Column 8 to 0.82 power	B.t.u. per hr. per sq. ft. per deg. mean temperature difference	$\frac{1000}{U}$	Test number
V_w	t_s	t_m	t_w	C	$V_{60°}$	$\overline{V}$	$\frac{1}{\overline{V}}$	$\left(\frac{1}{\overline{V}}\right)^{0.82}$	U	R	No.
6.61	126.5	73.7	52.8	1.12	5.91	10.05	0.0995	0.150	852	1.18	588
4.01	126.0	69.8	56.2	1.07	3.75	6.38	0.157	0.218	648	1.54	589
5.14	124.5	68.0	56.5	1.06	4.85	8.26	0.121	0.175	741	1.40	590
2.74	126.6	67.4	59.2	1.01	2.72	4.63	0.216	0.285	497	2.01	591
2.44	128.1	67.1	61.0	1.00	2.44	4.15	0.241	0.310	470	2.12	592
1.65	126.5	64.1	62.4	0.98	1.69	2.88	0.340	0.425	350	2.86	593
0.98	126.3	61.9	64.4	0.95	1.03	1.75	0.571	0.630	229	4.37	594
11.4	126.6	75.7	50.9	1.16	9.82	16.70	0.060	0.099	980	1.02	595
9.88	126.8	75.6	51.2	1.15	8.60	14.65	0.068	0.109	922	1.08	596
8.36	125.1	74.3	50.8	1.16	7.22	12.30	0.081	0.125	852	1.17	597
7.59	127.1	74.6	52.5	1.13	6.72	11.42	0.088	0.135	818	1.22	598
6.46	125.5	72.0	53.5	1.11	5.82	8.90	0.112	0.165	767	1.30	599
4.84	126.4	72.1	54.3	1.10	4.40	7.47	0.134	0.190	652	1.53	600
3.00	126.5	69.6	56.9	1.05	2.86	4.86	0.206	0.275	498	2.00	601
3.34	125.1	67.9	57.2	1.05	3.18	5.40	0.185	0.250	531	1.88	602
1.95	126.6	68.3	58.3	1.04	1.88	3.20	0.313	0.390	370	2.70	603
0.94	127.0	68.8	58.2	1.04	0.90	1.530	0.654	0.170	208	3.73	604
10.95	187.0	133.7	53.3	1.12	9.76	16.60	0.060	0.098	1010	0.91	605
9.74	186.0	133.5	53.5	1.11	8.85	15.05	0.066	0.105	968	1.03	606
8.52	186.8	132.9	53.9	1.10	7.25	12.32	0.081	0.125	926	1.08	607
7.82	187.1	133.8	53.3	1.12	7.00	11.90	0.084	0.129	876	1.14	608
5.89	186.8	131.0	55.8	1.07	5.50	9.34	0.107	0.158	794	1.26	609
3.79	186.7	128.7	58.0	1.04	3.64	6.17	0.162	0.222	623	1.60	610
1.99	187.0	126.2	60.8	1.00	1.99	3.38	0.296	0.370	401	2.49	611
2.29	186.8	126.1	60.7	1.00	2.29	3.89	0.257	0.330	459	2.18	612
0.86	187.0	123.7	63.3	0.96	0.90	1.53	0.654	0.700	198	5.05	613
11.28	186.4	132.9	53.5	1.11	10.3	17.50	0.057	0.095	985	1.02	614
9.87	186.9	132.5	54.4	1.10	8.95	15.22	0.066	0.105	929	1.08	615
8.64	186.6	131.4	55.2	1.09	7.92	13.45	0.074	0.115	881	1.14	616
7.77	186.0	130.6	55.4	1.09	7.12	12.10	0.083	0.127	854	1.17	617
6.53	186.3	128.9	57.4	1.05	6.22	10.56	0.095	0.143	787	1.27	618
4.92	186.6	129.7	56.9	1.05	4.69	7.98	0.125	0.179	665	1.50	619
3.38	186.6	128.2	58.4	1.04	3.25	5.52	0.181	0.245	542	1.85	620
1.46	187.3	125.4	61.9	1.00	1.46	2.48	0.403	0.480	306	3.27	621
2.44	186.6	126.8	59.8	1.00	2.44	4.15	0.241	0.310	446	2.24	622
8.91	111.6	60.9	50.7	1.15	7.71	13.10	0.077	0.120	940	1.06	623
8.94	111.5	60.8	50.7	1.15	7.76	13.20	0.076	0.117	[illegible]	1.08	624
8.75	111.3	47.6	63.7	0.97	9.02	15.35	0.065	0.103	[illegible]	1.03	625
8.75	111.0	38.8	72.2	0.85	10.3	17.50	0.057	0.095	1000	1.00	626
8.86	111.9	28.3	83.6	0.71	21.5	21.25	0.047	0.080	1108	0.90	627

TABLE 2—CONTINUED

1	2	3	4	5	6	7	8	9	10	11	12
Velocity of water in tube ft. per sec.	Temperature of steam deg. fahr.	Mean temperature difference deg. fahr.	Mean water temperature deg. fahr.	Correction for viscosity	Velocity of water reduced to 60 deg. fahr.	Corrected velocity of water referred to ⅝-in. tube	Reciprocal of column 7	Column 8 to 0.82 power	B.t.u. per hr. per sq. ft. per deg. mean temperature difference	$\frac{1000}{U}$	Test number
V_w	t_s	t_m	t_w	C	V_{60}	$\overline{V}$	$\frac{1}{\overline{V}}$	$\left(\frac{1}{\overline{V}}\right)^{0.82}$	U	R	No.
8.94	112.1	12.7	99.4	0.62	14.4	24.50	0.041	0.072	1233	0.81	628
8.79	114.6	63.4	51.2	1.15	7.66	13.02	0.077	0.120	920	1.09	638
8.75	114.1	48.7	65.4	0.94	9.30	15.85	0.063	0.101	945	1.05	639
8.79	114.3	41.9	72.4	0.85	10.35	17.62	0.057	0.095	995	1.01	640
8.86	115.1	33.5	81.6	0.76	11.7	19.90	0.050	0.085	1062	0.94	641
8.72	114.4	22.8	91.6	0.67	13.00	22.10	0.045	0.078	1127	0.89	642
8.72	116.0	15.6	100.4	0.62	14.00	23.80	0.042	0.073	1130	0.89	643
8.52	114.5	6.80	107.7	1.55	15.40	26.20	0.038	0.067	1290	0.78	644
8.72	113.9	13.3	100.6	0.61	14.30	24.30	0.041	0.072	1145	0.87	645
8.63	113.9	21.7	92.2	0.67	13.00	22.10	0.045	0.078	1095	0.91	646
8.63	114.4	30.0	84.4	0.74	11.6	19.70	0.051	0.086	1055	0.95	647
8.91	113.5	41.3	72.2	0.85	10.1	17.20	0.058	0.096	985	1.01	648
8.91	115.4	63.5	51.9	1.14	7.82	13.30	0.075	0.126	930	1.08	649
8.91	116.0	53.3	62.7	0.97	9.18	15.60	0.064	0.102	975	1.03	650
8.82	114.5	41.6	72.9	0.85	10.5	17.90	0.056	0.093	1018	0.98	651
8.72	114.6	31.4	83.2	0.75	11.75	20.10	0.049	0.083	1060	0.94	652
8.82	115.1	23.7	91.4	0.68	12.82	21.82	0.046	0.079	1088	0.92	653
8.79	117.4	15.4	102.0	0.60	14.7	25.20	0.040	0.070	1158	0.95	654
8.82	115.5	3.30	112.2	0.50	17.55	29.00	0.035	0.063	1342	0.75	655
8.82	115.0	3.60	111.4	0.51	17.30	29.50	0.034	0.061	1263	0.79	656
8.82	115.0	13.8	101.2	0.61	14.50	24.70	0.040	0.070	1193	0.84	657
8.82	113.8	60.7	53.1	1.12	7.89	13.40	0.075	0.127	963	1.04	658
8.72	114.5	50.5	64.0	0.95	9.18	15.60	0.064	0.102	993	1.01	659
8.86	114.8	42.5	72.3	0.85	10.41	17.75	0.056	0.093	1036	0.97	660
8.75	115.1	31.8	83.3	0.75	11.68	19.83	0.050	0.085	1050	0.95	661
8.75	115.0	23.0	92.0	0.67	13.05	22.20	0.045	0.078	1088	0.92	662
8.72	115.3	13.1	102.2	0.60	14.50	24.50	0.041	0.072	1163	0.86	663
8.63	115.3	3.90	111.4	0.51	16.9	28.80	0.035	0.063	1295	0.77	664
8.86	115.1	3.70	111.4	0.51	17.35	29.50	0.034	0.061	1403	0.71	665
8.75	115.3	13.1	102.2	0.60	14.6	24.85	0.040	0.700	1200	0.83	666
8.98	115.4	23.1	92.3	0.67	13.3	22.32	0.045	0.078	1113	0.90	667
8.63	115.0	32.7	82.3	0.76	11.35	19.32	0.052	0.088	1035	0.97	668
8.72	114.8	41.3	73.5	0.83	10.5	17.89	0.056	0.093	1008	0.99	669
8.72	115.0	50.8	64.2	0.95	9.18	15.62	0.064	0.102	984	1.02	670
8.86	114.5	6.80	107.7	0.54	16.4	27.90	0.036	0.064	1206	0.83	671
8.72	114.4	6.60	107.8	0.54	16.15	27.50	0.036	0.064	1305	0.77	672
8.91	114.9	3.70	111.2	0.55	16.2	27.60	0.036	0.064	1335	0.75	673
8.72	114.7	3.15	111.5	0.51	17.1	29.10	0.034	0.061	1510	0.66	674
8.86	114.0	5.40	108.6	0.54	16.42	28.00	0.036	0.064	1275	0.78	675
8.72	114.6	9.60	105.0	0.57	15.3	26.05	0.039	0.068	1165	0.86	676
8.75	114.7	6.20	108.5	0.54	16.2	27.60	0.036	0.064	1342	0.75	678
8.79	113.6	10.2	103.4	0.59	14.9	25.40	0.039	0.068	1228	0.81	679
8.64	115.3	10.7	104.6	0.57	15.15	25.80	0.039	0.068	1225	0.82	680
8.86	115.1	5.60	109.5	0.52	17.05	29.00	0.035	0.068	1367	0.73	681
8.68	115.4	3.20	112.2	0.50	17.35	29.50	0.034	0.061	1395	0.72	682

TABLE 2—Concluded

1	2	3	4	5	6	7	8	9	10	11	12
Velocity of water in tube ft. per sec.	Temperature of steam deg. fahr.	Mean temperature difference deg. fahr.	Mean water temperature deg. fahr.	Correction for viscosity	Velocity of water reduced to 60 deg. fahr.	Corrected velocity of water referred to ⅝-in. tube.	Reciprocal of column 7	Column 8 to 0.82 power	B.t.u. per hr. per sq. ft. per deg. mean temperature difference	$\frac{1000}{U}$	Test number
V_w	t_s	t_m	t_w	C	V_{60}'	$\bar{V}$	$\frac{1}{\bar{V}}$	$\left(\frac{1}{\bar{V}}\right)^{0.82}$	U	R	No.
8.80	115.1	5.80	109.3	0.53	16.6	28.25	0.035	0.063	1405	0.71	683
6.12	114.6	4.40	110.2	0.52	11.8	20.10	0.050	0.085	1123	0.89	686
5.97	114.3	13.7	100.6	0.62	9.62	16.35	0.062	0.100	1047	0.95	687
5.93	113.4	21.7	91.7	0.67	8.85	15.05	0.066	0.105	1007	0.99	688
5.93	115.0	31.4	83.6	0.74	8.00	13.60	0.074	0.115	945	1.06	689
5.90	114.3	41.4	72.9	0.84	7.03	11.95	0.084	0.130	892	1.12	690
6.12	114.2	49.9	64.3	0.95	6.43	10.95	0.091	0.137	868	1.15	691
6.16	115.0	61.3	53.7	1.12	5.50	9.35	0.107	0.157	812	1.23	692
6.24	114.5	61.0	53.5	1.11	5.68	9.65	0.104	0.154	823	1.21	693
6.16	114.3	50.0	64.3	0.95	6.48	11.02	0.091	0.137	858	1.17	694
6.12	115.3	42.0	73.3	0.84	7.30	12.42	0.081	0.125	891	1.12	695
6.20	115.5	32.9	82.6	0.75	8.28	14.10	0.071	0.112	935	1.07	696
6.28	114.5	22.7	91.8	0.67	9.37	15.95	0.063	0.102	1005	1.00	697
6.31	114.5	13.0	101.5	0.60	10.52	17.90	0.056	0.093	1055	0.95	698
6.12	114.5	3.40	111.1	0.51	12.00	20.40	0.049	0.084	1272	0.79	699
3.61	114.5	3.80	110.7	0.51	7.10	12.10	0.083	0.128	786	1.27	700
3.68	113.3	12.9	100.4	0.62	5.93	10.10	0.099	0.148	741	1.35	701
3.38	114.3	20.7	93.6	0.66	5.12	8.70	0.115	0.169	733	1.36	702
3.76	113.5	30.9	82.6	0.75	5.02	8.53	0.117	0.171	723	1.38	703
3.68	114.1	39.3	74.8	0.83	4.44	7.55	0.132	0.190	700	1.43	704
3.68	113.6	47.0	66.6	0.92	4.00	6.80	0.147	0.205	665	1.50	705
3.64	114.0	55.5	58.5	1.02	3.57	6.06	0.165	0.225	632	1.58	706
3.64	114.0	56.1	57.9	1.04	3.50	5.95	0.168	0.230	627	1.59	707
3.53	113.5	49.4	64.1	0.95	3.72	6.32	0.158	0.216	625	1.60	708
3.53	113.8	39.3	74.5	0.83	4.25	7.22	0.138	0.195	660	1.52	709
3.53	114.0	30.8	83.2	0.75	4.70	7.99	0.125	0.180	693	1.44	710
3.53	114.1	21.1	93.0	0.67	5.27	8.95	0.112	0.165	722	1.39	711
3.53	113.3	11.8	101.5	0.61	5.79	9.85	0.102	0.152	740	1.35	712
3.57	115.4	5.10	110.3	0.52	6.88	11.70	0.085	0.130	812	1.23	713
2.01	112.4	55.3	57.1	1.05	1.93	3.28	0.305	0.380	392	2.55	714
1.83	113.8	47.6	66.2	0.93	1.97	3.35	0.299	0.375	383	2.62	715
1.83	113.0	37.7	75.3	0.83	2.20	3.74	0.267	0.340	410	2.43	716
1.80	113.5	30.6	82.9	0.75	2.40	4.08	0.245	0.315	420	2.38	717
1.90	112.6	19.7	92.9	0.67	2.84	4.83	0.207	0.275	478	2.09	718
2.14	113.8	11.9	101.9	0.60	3.57	6.07	0.165	0.225	500	2.00	719
2.14	113.8	3.10	110.7	0.51	4.20	7.15	0.140	0.195	650	1.54	720
2.07	112.8	3.2	109.6	0.52	3.98	6.77	0.148	0.205	632	1.58	721
2.09	112.9	12.3	100.6	0.61	3.43	5.83	0.172	0.235	510	1.96	722
1.93	113.9	22.5	91.4	0.68	2.84	4.83	0.207	0.275	467	2.14	723
2.10	113.1	29.1	84.0	0.74	2.84	4.83	0.207	0.275	500	2.00	724
2.12	113.6	54.7	58.9	1.02	2.08	3.54	0.282	0.360	418	2.39	725
2.03	113.5	46.9	66.6	0.92	2.21	3.76	0.266	0.335	433	2.31	726
1.87	114.6	39.2	75.4	0.83	2.25	3.83	0.261	0.332	418	2.39	727

and the data grouped in accord with certain average temperatures. The results are plotted in Fig. 5 from which it is seen that there is a distinct shift from the mean of the points for the lowest temperature to those of the highest temperature. Of course these results are too few and, not having been obtained with this purpose in mind, are too inaccurate for quantitative results, but they show well enough the qualitative effect.

13 Looking now at the points defining the five lines of more or less equal temperature it is noted that these form a wide band. If,

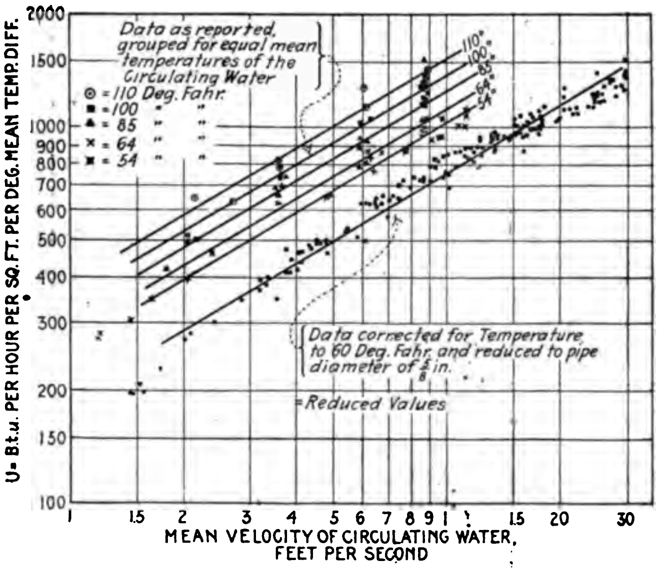

FIG. 5 RATE OF HEAT TRANSFER, VERSUS CIRCULATING WATER VELOCITY RESULTS OF TESTS BY ORROK, TRANS. AM. SOC. M. E., 1910

however, we apply a correction similar to that for the critical velocity so as to reduce the results to some common temperature, say 60 deg. the width of this band is cut in half. For instance, if the mean water temperature in one case were 40 deg. and the velocity 5 ft. per sec., the resistance to heat transfer would be the same as if the temperature were 60 deg. and the velocity

$$\frac{5 \times P_{60}}{P_{40}} = 3.62 \text{ ft-sec.}$$

This correction has been applied in Table 2 and the results as reduced from a 1-in. to a ⅝-in. pipe are plotted in Fig. 5 along with the uncorrected data. The reduction in the dispersion is quite marked and the agreement, in view of the difficulties in eliminating air from the steam in a test of this sort, becomes now quite good. There is a well marked curvature in the plots which bears out the statement in Par. 7 that no simple exponential law can represent the conditions throughout the whole range, though in this case it does very well through a wide range.

14 These results are in terms of conductivities, so that for the

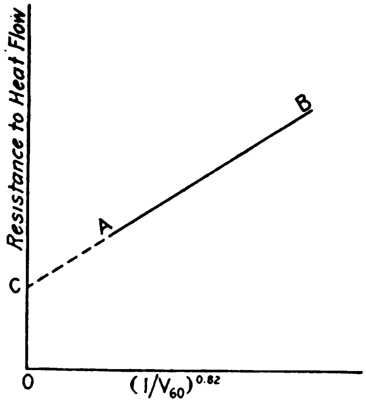

FIG. 6 RESISTANCE TO HEAT FLOW, GENERAL FORM

purpose of summation it is necessary to convert them into resistances. The water resistance does not vary directly as the velocity but the expression

$$R_w = \frac{Pk}{V^{0.82}}$$

is the value of water resistance when P is the viscosity divided by density and k is a constant for any given diameter of pipe. It now remains to define the units in which R is to be measured and R is therefore taken arbitrarily as

$$R = \frac{1000\ t_m}{\text{B.t.u. passed per hour per sq. ft. of external tube surface}}$$

This is seen to be 1000 times the reciprocal of the coefficient of heat transfer as defined above. Accepting this definition, suppose we have a set of observations on the condensation of air free steam, and that by means of Poiseuille's formula we find for each result the actual velocity of flow which would correspond to the viscosity and density at the standard temperature of 60 deg. Calling these "reduced velocities" V_{60}, we can plot values of R against the corresponding values of

$$\frac{1}{(V_{60})^{0.82}}$$

and get a straight line as shown in Fig. 6. The value of the exponent 0.82 was obtained by trial and error by the author of Fig. 7 and is such that AB becomes a straight line, when plotted as shown.

15 When, now, the velocity of flow becomes infinite $\frac{1}{V} = 0$, and if we prolong the straight line to C, the intercept on the vertical axis, OC, represents the sum of the resistances, R_s and R_t, while a line through the origin parallel to the original line gives the resistance due to the water side alone or R_w. We have, then, a linear relation between the resistance on the water side and one variable, the velocity, after the other variables have been taken care of in reducing the velocity. The resistance of the tube walls for clean tubes is well known and that on the steam side has been determined for the one case, that of steam at 212 deg. The curves resulting from plotting experimental data will lie above that due to the sum of the resistances as calculated because in practice steam is never quite air free and the resistance on the steam side is increased by that due to the air.

16 This constitutes the reduction for viscosity, but when the results of different experimenters are considered it is found that different sized tubes have been used. Now the velocity in a 1-in. pipe corresponding to that in a ⅝-in. pipe of the same thickness is in the ratio of the internal diameters, according to Reynolds' expression. In the "Engineering" paper all results have been referred to a tube ⅝ in. outside diameter of 18 S. W. G., or 0.048 in., while Orrok's experiments were conducted on a tube of 1 in. outside diameter and 18 B.W.G., or 0.049 in. The velocity in a ⅝-in. tube corresponding to that in the 1-in. tube, then, is given by multiplying the latter by the ratio

$$\frac{1 - 2\times 0.049}{0.625 - 2\times 0.048} = \frac{0.902}{0.529}$$

17 The values of R above are resistances in the arbitrary units referred to the unit of external tube surface. If ρ denotes the same

resistance referred to the internal surface will be less than R. In the 1-in. tube mentioned above

$$(\rho)_1 = (\rho)_{5/8} \times \frac{0.529}{0.902}$$

It is more convenient to deal with R than with ρ and since we have

$$\rho_w = R_w \times \frac{\text{Inner diameter}}{\text{Outer diameter}}$$

$$(R_w)_1 = (R_w)_{5/8} \times \frac{1.00}{0.902} \times \frac{0.529}{0.625} \times \frac{0.529}{0.902} = 0.55\ (R_w)_{5/8}$$

This gives a reduction factor for relating the two sizes of tubes on the basis of the resistance of the water.

18 As an example of the method of applying the corrections for working up the results, take the first run in Orrok's results. Here $V_w = 11.2$ ft. per sec.; $t_s = 126.3$ deg.; $t_m = 73.2$ deg.; $U = 1000$. Now $t_s - t_m = t_w = 54.1$ deg. From the table of correction factors, Table 1, for $t_w = 54.1$ deg., $C = 1.10$. Then

$$V_{60} = \frac{11.2}{1.10} = 9.3$$

$$V_{\text{final}} = V_{60} \times \frac{0.902}{0.529} = 9.3 \times \frac{0.902}{0.529} = 15.9$$

of which the reciprocal is 0.063. This can be raised to the 0.82 power by laying off a pair of scales, one of which will show V when the other gives $V^{0.82}$. By this method the result is 0.102.

$$R = 1000 \times \frac{1}{U} = \frac{1000}{1000}$$

or in this case, 1. The coördinates of a point on the final curve are then 1 and 0.102.

19 In Fig. 7 is shown the result of applying this method to the results of different experimenters as reported in Engineering. No experimental points have been rejected and the results are in good accord. In Fig. 8 are shown the results of the experiments by Mr. Orrok. These are the same points as those plotted in Fig. 2 in which there was an apparent inconsistency, yet here they are in quite good accord. At the same time it is noted that the slope of the curve from Orrok's results is steeper than that taken from "Engineering" and this is explained probably by the fact that the circulating water used by Orrok was taken from a salt water fire main which in turn took

its supply from a source contaminated with sewage. The density of this mass being greater than that of pure water, the total weight was partly due to foreign matter. The true water velocity must then

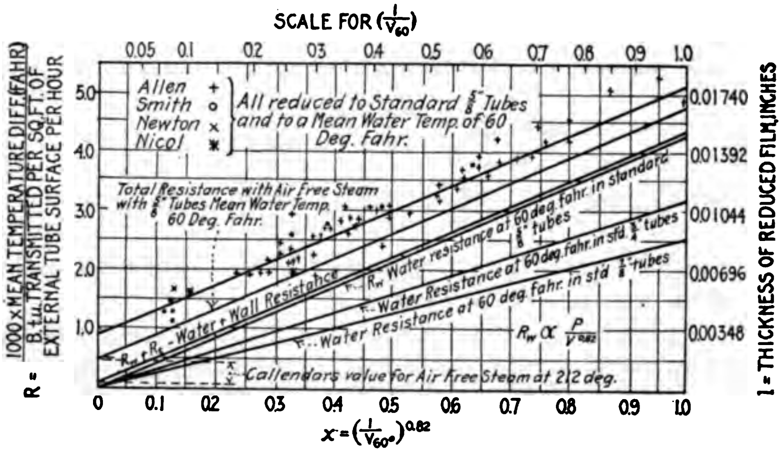

Fig. 7 Resistance to Heat Flow in Terms of "Fully Reduced" Velocity of Circulating Water. Engineering, Jan. 23, 1914

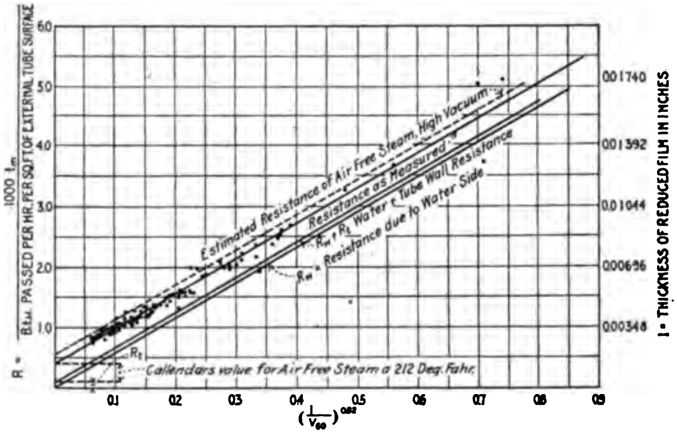

Fig. 8 Curves of Resistance to Heat Transfer in a Condenser. Reported by Orrok. Reduced to this Form for a ⅝-in. Tube and the Velocity Corresponding to 60 deg. Mean Temperature of Circulating Water

have been smaller than that reported if referred to fresh water, and without taking into consideration the difference in viscosity, $\frac{1}{V}$ must have been larger. This correction would tend to flatten out the curve, but since no data is available as to the real conditions in either case the proper correction cannot be evaluated. The quality of the circulating water then, is another factor which has apparently been neglected and we have here the results to be expected under ordinary circumstances and those encountered in the neighborhood of a large city where sewage is discharged to the waters used for cooling purposes.

20 In Fig. 9 is shown the results of applying the method of reduction to experimental results reported for a Navy feed water heater.[1] In the comment within the article it was stated that the results of the tests as reported were considered to be in error due to

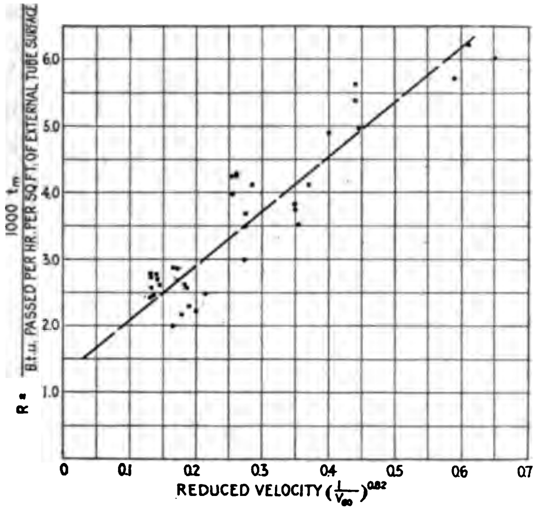

Fig. 9 Curves of Resistance to Heat Transfer in a Feed-Water Heater, Journal A. S. N. E., Vol. 32

the presence of air on the *water side*. No air cock had been fitted to remove the gases expelled from the river water by heat, with the result that they collected and interfered with the heat transmission. Not only are the results inconsistent but the resistances are extremely large due to the formation of a gas film on the water side, and this calls to our attention still another variable. This one is not likely to be met in condenser practice but it is very likely to be encountered in feed water heaters, for here the greater temperature rise results in

[1]Journal A. S. N. E., Vol. XXIV, No. 1, Feb. 1912.

the expulsion of larger amounts of gas and often the provisions for getting rid of the gas are not as good as in condensers.

21 Having reduced the results to a linear form it now becomes possible to interpret them. Referring to the curves we see that the line drawn through the plotted points gives the resistance to heat transmission in terms of a reduced velocity. The line through the origin parallel to this gives the resistance due to the film on the water side. The resistance due to the tube wall itself can be computed and is about 0.08 units. If this amount be laid off on the vertical axis and a line be drawn parallel to the water resistance line, then the resulting curve is the sum of the resistances on the water side and tube wall plotted against the same fully reduced velocities. According to Professor Callendar's result for the resistance due to air free steam condensing at 212 degrees, the resistance in our unit is about 0.34. For steam at lower temperatures this resistance is a little higher and may be taken as about 0.48 units for the conditions in condensers. If this amount be laid off on the axis and another parallel to the original plot be drawn, we have the curve of resistance to heat transfer for air free steam giving up its heat through a tube of known thickness to water moving at a certain reduced velocity. Above this lies the curve of resistances to be expected in condenser practice and we see that Orrok's curve in addition to being steeper has a smaller intercept. This difference is accounted for in the fact that Orrok took special precautions against air. The steam used was generated in a closed system and after condensation was returned for re-evaporation. The other results were obtained in larger experimental condensers and in certain cases in actual condensers under the operating conditions due to the use of a dry vacuum air pump. Just how well Mr. Orrok succeeded in getting rid of air is shown by the similarity in the intercepts.

22 So far we have dealt with these resistances in terms of an arbitrary unit because it is simpler to do so, but this is not necessary, since the resistance may be expressed in terms of the thickness of equivalent water films, by a simple transformation. Referring to our original law of conductivity we find that it may be written

$$U = \frac{1}{\rho l}$$

in which

U is now the symbol for the B. t. u. transmitted per hour, per square foot, per degree mean temperature difference.

ρ is the reciprocal of the conductivity and
l is the thickness of the film.

For water ρ is $\frac{1}{3.48}$. Referring back to the definition of water resistance, we find that R_w may be written $R_w = 1000\rho l$. For $R_w = 1$, l is found to be 0.00348 and we can now convert the scale of R_w on the curves to one of film thicknesses in fractions of an inch of the fluid. When this is done we have the thickness of the films on each side and that of the tube wall expressed as a water film thickness, and we have the film thickness on the water side in terms of one variable which has been reduced in such a manner that all other variables are accounted for. In other words, we have a linear relation between the physical object offering the resistance and the factors upon which it depends for its magnitude.

23 It now remains to express the results algebraically and this may be readily done. Since the curves are of the form $y = mx + b$ we have, measuring the values of m and b on the line for "resistance as measured" in Fig. 7,

$$l = 0.0143 \left(\frac{1}{V_{60}}\right)^{0.82} + 0.00313$$

24 Fig. 7 is used because it represents the results from a number of actual condensers rather than one experimental apparatus as in Fig. 8. Now the reduced velocity V_{60} was corrected for both viscosity and the diameter of the standard 5/8-in. tube, 0.529 in. If we let d be the inside diameter of the tube under test and P_t be the viscosity ÷ density for the mean water temperature as before and P_{60} the same ratio for the standard water temperature we can write

$$l = 0.0143 \left(\frac{P_t \times 0.529}{V \times P_{60} \times d}\right)^{0.82} + 0.00313$$

Calling the ratio of the viscosities C_w, water correction, and reducing we find

$$l = 0.0085 \left(\frac{C_w}{V \times d}\right)^{0.82} + 0.00313$$

25 If now C_w can be expressed in terms of the temperature we have an expression involving the film thickness and all the variables concerned. In Fig. 10 the values of C taken from Fig. 4, have been plotted against absolute temperature in degrees Fahrenheit on logarithmic paper, and it is seen that while the curve is not exactly straight it can be divided into two sections which are practically so. The lower section has a range from 32 to 140 deg. fahr., which em-

braces the temperatures met in condenser practice and greatly exceeds the limits in this practice. Expressing the lower section algebraically we have

$$C_w = \left(\frac{520}{T_w}\right)^{0.57}$$

Putting this value in the expression for l above

$$l = 0.0085\left(\frac{520^{6.57}}{T_w^{6.57} \times V \times d}\right)^{0.82} + 0.00313$$

From which

$$l = \frac{20.6 \times 10^{12}}{T_w^{5.88} \times (V \times d)^{0.82}} + 0.00313$$

For water temperatures outside this range, such as those met in feedwater heaters the upper section may be used and a similar ex-

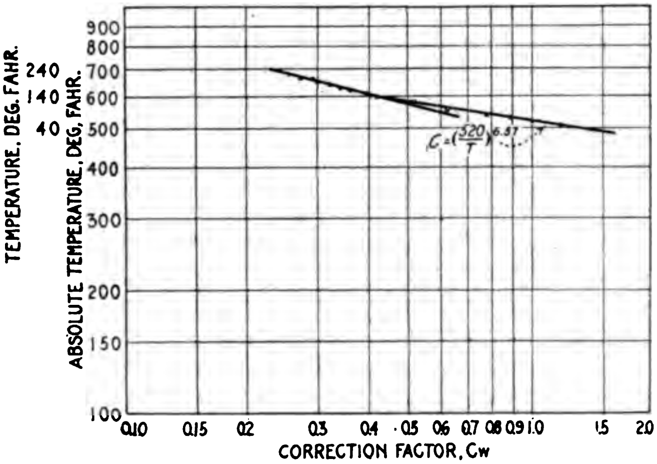

FIG. 10 CURVES OF CORRECTION FACTORS FOR VARIATION IN VISCOSITY DUE TO TEMPERATURE OF CONDENSER CIRCULATING WATER

pression obtained. This expression is a rational one as far as film thicknesses are concerned but is still not in a practical form. To reduce it to such form, the general law of heat transfer is utilized. According to this law

$$Q = AUt_m$$

in which Q is the heat transferred per hour, A, the area of the heating surface, and t_m the mean temperature difference as before. Remembering that we can now write

$$U = \frac{1}{\rho l}. \quad A = Q\frac{\rho l}{t_m}$$

from which

$$A = \frac{Q}{3.48 \times t_m}\left(\frac{20.6 \times 10^{12}}{T_w^{5.38}(V \times d)^{0.82}} + 0.00313\right)$$

Upon further reduction this becomes

$$A = \frac{Q}{t_m}\left(\frac{592 \times 10^{10}}{T_w^{5.38}(V \times d)^{0.82}} + 0.0009\right)$$

26 This is a rational expression for the area of the heating surface required in a condenser in which all the surface is active and in which steam is substantially air free, as it exists when an effective dry vacuum pump is in use. If a portion of the total surface is to be drowned for the purpose of cooling the condensate to augment the effective capacity of the wet air pump, then the above area must be increased in proportion. If air is present in the steam or the condenser is so designed that part of it becomes inactive through pockets or crowding of the tubes, then the area must again be increased, but in any case the area found from the above expression is that required to transmit the given quantity of heat under the conditions laid down. It is the law of heat transfer in condensers, and as such is a basis of rational design.

27 As an example of its application the area of a given condenser can be computed. Referring to Fig. 7 it is noted that one point of some tests by B. J. Nicol is located near the curve at a fully reduced velocity of 0.13 and a resistance of 1.4 This point corresponds to a run having the following data: Heat units transferred per hour per sq. ft., 123,722; t_m, 177 deg. fahr.; t_w, 77 deg. fahr.; actual velocity, ft. per sec. 6.95; diameter of tubes inside, 0.664. These data are taken from Table III, page 4, Engineering, January 2, 1914. If we assume a surface of say 100 sq. ft. in order to obtain Q, the total heat transferred per hour, and, by substituting this Q with the other data in the formula, solve for A and obtain the area as 100 sq. ft. we have checked the formula. On this basis Q becomes 12,372,200 and from the formula

$$A = \frac{12{,}372{,}200}{177}\left(\frac{592 \times 10^{10}}{538^{5.38}(6.95 \times 0.664)^{0.82}} + 0.0009\right)$$

$$= \frac{12{,}372{,}200}{177}\left(\frac{592 \times 10^{10}}{2.48 \times 3.50 \times 10^{15}} + 0.0009\right)$$

$$= \frac{12{,}372{,}200}{177}\left(0.000682 + 0.0009\right)$$

$$= 110 \text{ sq. ft.}$$

This is fairly close to the assumed value of 100 and the variation

TABLE 3 CALCULATIONS FOR REDUCTION OF DATA ON FEED WATER HEATERS

Reported in Journal A.S.N.E., Vol. XXIV, No. 1

1	2	3	4	5	6	7	8	9	10	11	12
Velocity of water in tube ft. per sec.	Temperature of steam deg. fahr.	Mean temperature difference deg. fahr.	Mean water temperature deg. fahr.	Correction for viscosity	Velocity of water reduced to 60 deg. fahr.	Corrected velocity of water referred to ⅝-in. tube.	Reciprocal of column 7	Column 8 to 0.82 power	B.t.u. per hr. per sq. ft. per deg. mean temperature difference	$\frac{1000}{U}$	Test number
V_w	t_s	t_m	t_w	C	$V_{60°}$	$\bar{V}$	$\frac{1}{\bar{V}}$	$\left(\frac{1}{V}\right)^{0.82}$	U	R	No.
0.588	227.1	122.3	104.8	0.59	1.00	1.70	0.588	0.650	165.8	6.03	1
0.583	239.3	131.2	108.1	0.57	1.02	1.74	0.575	0.640	156.1	6.40	2
0.595	249.2	137.3	111.9	0.55	1.08	1.84	0.543	0.610	159.3	6.23	3
0.595	257.6	142.5	115.1	0.53	1.12	1.90	0.526	0.590	175.0	5.71	4
1.190	232.5	125.9	106.6	0.57	2.09	3.56	0.281	0.351	260.7	3.83	5
1.190	240.9	139.4	101.5	0.60	1.98	3.37	0.297	0.370	243.2	4.12	6
1.185	250.9	143.4	107.5	0.56	2.10	3.57	0.280	0.350	267.6	3.75	7
1.185	262.8	149.4	113.4	0.54	2.19	3.53	0.283	0.360	284.3	3.52	8
1.785	231.9	138.1	93.8	0.66	2.70	4.59	0.218	0.285	243.2	4.12	9
1.785	241.3	143.8	97.5	0.63	2.83	4.82	0.207	0.275	271.0	3.69	10
1.790	250.9	152.9	98.0	0.63	2.84	4.83	0.207	0.275	285.6	3.50	11
1.775	259.5	160.4	99.1	0.62	2.87	4.88	0.205	0.274	337.4	2.97	12
2.390	249.3	132.8	116.5	0.52	4.60	7.82	0.127	0.182	381.6	2.62	13
2.380	258.1	149.4	108.7	0.56	4.25	7.23	0.138	0.195	379.9	2.63	14
2.385	253.9	154.8	99.1	0.62	3.87	6.58	0.152	0.212	414.4	2.49	15
2.380	264.5	156.5	108.0	0.57	4.18	7.10	0.141	0.200	451 5	2.21	16
2.860	235.3	138.1	97.2	0.64	4.47	7.60	0.132	0.187	389.1	2.57	17
2.860	239.8	144.4	95.4	0.65	4.40	7.47	0.134	0.190	433.9	2.30	21
2.860	258.3	156.3	102.0	0.60	4.77	8.12	0.123	0.177	456.3	2.19	22
2.860	269.6	160.0	109.6	0.55	5.20	8.84	0.113	0.165	498.6	2.00	23
0.600	225.4	65.1	160.3	0.37	1.62	2.75	0.364	0.440	201.0	4.98	18
0.600	238.1	78.7	159.4	0.37	1.62	2.75	0.364	0.440	178.0	5.62	19
0.600	248.9	88.5	160.4	0.37	1.62	2.75	0.364	0.440	185.6	5.40	20
0.602	258.4	87.7	170.7	0.34	1.77	3.05	0.328	0.400	204.5	4.90	36
1.210	229.9	79.0	150.9	0.40	3.00	5.10	0.196	0.261	235.6	4.26	28
1.205	242.2	93.9	148.3	0.39	3.10	5.27	0.190	0.255	234.4	4.26	29
1.205	253.3	103.7	149.6	0.40	3.00	5.10	0.196	0.261	233.8	4.27	30
1.212	258.4	107.3	151.1	0.39	3.10	5.27	0.190	0.255	250.4	3.99	31
1.790	238.1	72.1	166.0	0.35	5.10	8.67	0.115	0.167	346.9	2.88	32
1.785	247.9	84.6	163.3	0.36	4.95	8.40	0 119	0.172	346 0	2 89	33
1.785	250.0	92.1	157.9	0.37	4.83	8.20	0.121	0.174	345 5	2.89	34
1.795	259.9	96.1	163.8	0.36	5.00	8.50	0.118	0.173	373.4	2.69	35
2.380	230.5	69.8	160.7	0.36	6.61	11.20	0.089	0.135	408.6	2.44	37
2.405	240.1	83.4	156.7	0.37	6.35	10.80	0.093	0.140	358.4	2 79	38
2.380	262.0	93.7	168.3	0.35	6.80	11.55	0.087	0.132	354.6	2.81	39
2.380	257.5	95.5	162.0	0.36	6.62	11.23	0.089	0.135	408.2	2.45	40
2.870	239.3	98.7	140.6	0.42	6.82	11.60	0.086	0.131	363.8	2.75	25
2.875	250.1	107.1	143.0	0.42	6.83	11.62	0.086	0.131	386.9	2.58	26
2.865	257.8	123.0	134.8	0.45	6.40	10.89	0.092	0.140	363.7	2.75	27

may be assigned to the fact that the point does not lie exactly on the curve from which the law was derived, indicating some error in experimental work, which determined the values used in the formula. This checks the formula for one value chosen at random and it can be shown by similar substitution that the expression is true throughout the entire range of temperatures, velocities and tube diameters met in condenser practice.

28 The above rational expression for the area of condensers, the determination of the actual thickness of the dead fluid film and its variation with the flow conditions, suggests a method of dealing with all heat transfer apparatus, dependent upon the same principles. With the water resistance measured above, tests may be run on an economizer, for instance, and from the known resistance of the tube walls, the resistance of the gas film may be evaluated for mass flow variations. The actual thickness of the dead gas film may be evaluated as before and with these data similar tests may be conducted on a superheater, remembering that in this case, the controlling resistance may be on either side and may shift from one to the other. With full control of all the variables involved, it seems possible to write complete laws for each type of apparatus from the experimental data, and in this way assemble information that will be of the greatest value, not only to the designer, but to the operator as well.

CONCLUSIONS

29 In condensers and feed water heaters, using practically air free steam, the controlling resistance to heat transfer is on the water side. This resistance is a function of the water velocity, its mean temperature and the diameter of the tubes, and it varies inversely as these three variables, according to a straight line law. The rate of heat transfer cannot properly be expressed as an exponential function of the mean temperature difference or the water velocity in a general law of heat transfer, though such an expression may serve well for purposes of empiric design. Large quantities of air collecting in pockets on the water side greatly increase the resistance to heat flow while the quantity of solid matter contained in the circulating water may increase the apparent resistance by increasing the apparent velocity as determined by the weight of water passing. The viscosity of the circulating water has an important influence on the resistance, since the warmer the water the less resistance it offers. The resistance due to oily circulating water may be greatly different from that of fresh water. Finally by analysis of reliable

test data it is possible to get an expression for the area of heating surface required to transmit a given quantity of heat, in terms of the prime variables, from which it is seen that the area varies directly as the quantity of heat to be transferred per hour per square foot, and inversely as the circulating water velocity and its mean temperature, as well as the tube diameter. Such an expression, based on good experimental results, should replace empiric coefficients and become a basis of rational functional design, at the same time being of assistance in the operation of heat transfer apparatus.

DISCUSSION

Robert C. H. Heck (written). In this paper the argument is developed so much by the empirical method that the title "rational" is not fully deserved. Thus in the original discussion in *Engineering* the ratio P/P_{60} was at first computed with 0.00221 instead of 0.000221 for the coefficient of T^2, and the values of R were found to be proportional to $1/V$. When attention was called to the error the base was changed (presumably by trial calculation) to $1/V^{0.82}$, with the result shown in Fig. 7 of the paper. This flexibility or insensitiveness of method leaves one in doubt as to the finality of the conclusions reached.

The argument from analogy in Par. 11 seems rather farfetched. If $U = k_1 V^{0.4}$ and $h = k_2 V^{1.8}$, the functions U and h are not really enough alike in their manner of variation to make a good team. Further, the assumption that general velocity states are proportional to critical states is an hypothesis of the simple, first-trial, partly probable type, which must be confirmed by direct and copious experiment before it can be accepted as truly giving the overall effect of the complex actions which it covers. Even with the same tubes and with the same mean velocity, but with different water temperatures, variations in heat transfer would not surely measure variations in turbulence of flow, because the conductivity of the water probably varies with temperature, although less rapidly than does convection.

Again, consider the assumption that because, with the same water temperature, the critical velocity in a 1-in. tube is only half as great as in a ½-in. tube, therefore a unit surface of the larger tube is twice as effective as a unit of the smaller. This means that with the same temperature and velocity the convection would be twice as great in the larger tube. Of course, more water is passing the unit surface of the larger tube and the eddy currents are larger in volume; but does not the idea of complete inverse proportionality to diameter appear to be too simple and to give too much weight to size of tube? At any rate, it is open to doubt and subject to experimental trial. I believe that the difference of slant in the final lines of Figs. 7 and 8 is due more to excessive influence of tube diameters than to the causes named in Par. 19.

It is easy and helpful to define verbally the unit of R here used as the resistance through which a temperature difference of one degree will drive 1000 B.t.u. per square foot per hour.

LEO LOEB (written). The results of Lieutenant Wilson's studies are summarized in his conclusions that the controlling resistance to heat transfer from a condensing vapor to liquid warming rests on the liquid side, and that the numerical value depends on the three factors of water velocity, mean temperature and tube diameter. The most recent developments in heat transfer apparatus, all directed toward the reduction of this resistance, have produced types of apparatus in which the agitation of the liquid is carried to the highest practicable point consistent with reasonable frictional resistance, and it appears that in the near future the largest part of such apparatus will embody construction principles which make it altogether impossible to base results on either velocity or diameter.

The variations in results reported by several investigators are not entirely due to the neglect of any one variable, so much as the lack of standard methods of test and of uniformity of presentation of data, as well as in many instances to an improper conception of the meaning and application of the term *temperature difference.*

As an example of the possibility of accidental errors influencing final results might be cited a test of a feed water heater in which there was a noticeable difference in temperature across the cross section of a 6-in. pipe where the outlet temperature was measured. Only after the installation of a mixing device could the thermometer be depended upon to indicate the true temperature at outlet. Several instances have shown that water is discharged from different portions of the heating surface in streams of varying temperature and that these streams do not diffuse readily.

Another source of uncertainty is the average temperature of the heating medium. In the case of air-free condensers and feed-water heaters with generous tube spacing, the pressure throughout the steam space is substantially constant, and the average temperature therein is that corresponding to the pressure. However, in many instances the heat transfer is so great that the velocity of approach of steam toward the heating surface is considerable and it will be necessary to determine the steam pressure at exit as well as inlet or take the saturation temperature at several points within the shell.

One cause of difficulty is the assumption usually made that the law of heat transfer is analogous to the electric or magnetic circuit. This analogy can be directly carried out when there is the consideration of a single thermal resistance, the case which never occurs in practice.

The writer has recently summarized[1] a great number of tests made to determine the effect on heat transfer under air-free conditions of various forms of water film agitators produced either through the use of retarders in the tubes or by a deformation of the tube in process of manufacture. Those tests which Lieutenant Wilson cites in Paragraph 20 were the first made and represent only a few preliminary results used to modify the test arrangement in order to obtain consistent data.

Also, in order to determine the form of temperature gradient in a tube a number of tests were conducted at the Naval Engineering Experiment Station, Annapolis, Md., upon single tubes and normal heaters. The results[1] give data which may be used directly to evaluate the temperature gradient.

In view of the trend of design of heat transfer apparatus toward types in which velocity or reduced velocity is not a measure of film agitation, it is not considered practicable with the information in hand to introduce another factor of viscosity. A moderate number of tests on any type will serve to establish the few coefficients needed by a designer in the development of geometrically similar apparatus.

EDGAR BUCKINGHAM (written). While Mr. Wilson's very interesting paper has, ostensibly, only a purely practical object, it evidently also suggests some consideration of the physics of the subject of heat transmission which he has refrained from giving but which may be in place in the discussion. We may confine our attention to the dominating resistance, namely that encountered by the heat in passing from the inner surface of the tubes into the

[1]Heat transmission and tube length in Marine Feed Water Heaters. American Society of Naval Engineers—May 1915.

water flowing through them, and we shall assume that the water is moving with more than the critical speed so that its motion is turbulent. We shall also suppose that the apparatus, whatever its nature, is in a steady state of operation so that the temperature and water velocity at any point are sensibly constant with time, when measured by instruments too sluggish to respond to the rapid turbulent changes.

Against the tube wall, the water is stationary and heat must pass from the metal at first by conduction. There is no stationary film of finite thickness, but as we consider points farther and farther from the solid surface, the mean axial speed and the degree of turbulence increase rapidly, until all over the body of the cross section the water is so thoroughly mixed that its temperature is nearly uniform. The transition from zero speed and the temperature of the metal to a nearly uniform speed and temperature away from the wall is of course gradual and continuous; but it is very rapid near the wall and we may get a good approximation to the facts by idealizing them somewhat. We may, without departing far from reality, imagine that there is an absolutely stationary film of finite thickness against the wall, and that beyond this the fluid is completely mixed by turbulence. The process of heat transmission will then consist in pure conduction through the stationary film, followed by the removal of heat from the inner surface of the film by convective mixing.

Among the physical properties of the water (or other liquid) which enter into the problem and determine what happens we have therefore to consider; *first,* its thermal conductivity, and *second* those properties which determine the convective effect of a given state of motion, namely density and specific heat. Finally, the state of motion itself depends on and is fully determined by the density and viscosity of the liquid, its mean speed of flow, and the diameter of the tube, if, as we are now supposing, the tube is round, smooth, straight, and long in comparison with its diameter.

As regards the state of motion of a sensibly incompressible fluid in such a tube, it may be shown by dimensional reasoning to depend only on the value of the dimensionless quantity $(DV\rho/\mu)$, in which D is the internal diameter of the tube and V, ρ, μ are the speed, density, and viscosity of the fluid. Furthermore, it is well known that this conclusion from the principle of dimensional homogeneity is in perfect accord with the observed facts.[1] In order therefore to compare different cases of any phenomenon, such as heat transmission, which depends on the nature of the fluid motion, we ought to make the comparison at equal values of $(DV\rho/\mu)$; or in plotting results to show the effect of speed, we should use not V but $(DV\rho/\mu)$ as one coördinate. This is, in effect, what Mr. Wilson has done by making his reductions to standard viscosity and standard diameter, although for his purpose he has found it more convenient to state the matter differently. No reduction to standard density was needed because the density of water varies so little with temperature.

To see how the various quantities mentioned may enter into the problem, it is well to make a preliminary survey by means of dimensional reasoning. At any section of the tube, let ρ, μ, C be the density, viscosity, and specific heat of the liquid at the nearly uniform temperature prevailing over this section; these quantities determine the convective behavior of the liquid. Density and specific heat are so nearly constant that we may disregard their possible variations with temperature. Viscosity, on the other hand, decreases rapidly with rising temperature and is therefore much less against the wall than at a distance. But the layer where this variation takes place is so thin and the liquid there so nearly at rest, that the whole convective action of the liquid can hardly be appreciably affected by this variation of viscosity close to the wall, and it seems safe to proceed as if μ had only a single fixed value for the whole section depending on the mean temperature there.

The rate of conduction through the stagnant film depends on the temperature difference Δ at the section in question and also on the conductivity of the liquid. We know very little about the conductivity of liquids, but if we make the reasonable assumption that it may be represented well enough for practical purposes as a linear function of the temperature, we may be sure that the behavior of the liquid as regards conduction can be completely specified by λ, a, and Δ, where λ is the conductivity at the mean temperature of the section, and a is the temperature coefficient of the conductivity between this and the temperature of the inner tube surface which is Δ degrees hotter.

[1]See, for example, Stanton and Pannell on Similarity of Motion in Relation to the Surface Friction of Fluids, Phil. Trans. Royal Soc. London, *A 214*, p. 199, Jan. 1914.

If we now let Q represent the heat transmitted to the liquid from unit length of the tube in unit time, there must be some definite relation connecting Q with the other quantities mentioned, and we may represent it by writing

$$F(Q, D, V, \rho, \mu, C, \lambda, a, \Delta)=0 \qquad [1]$$

The validity of this symbolic statement of the existence of the relation in question depends only on the following assumptions: *a* that the density and specific heat are sensibly independent of temperature over the range Δ deg.; *b* that the conductivity may be treated as a linear function of the temperature over this same range; *c* that the variation of viscosity very close to the wall is of negligible importance in determining the motion and the resulting convection, in comparison with the viscosity throughout the great body of the liquid outside of the nearly stagnant film. There can hardly be any doubt that these assumptions may be regarded as entirely legitimate for the moderate temperature ranges with which we are concerned at present.

By proceeding in the manner illustrated in my paper[1], the number of variables in equation [1] may be reduced by four and the equation may be put into a great variety of forms, one of the simplest and most convenient for our present purpose being

$$Q = DV \rho C \Delta \Phi \left(\frac{DV\rho}{\mu}, \frac{V^2 a}{C}, \frac{\mu C}{\lambda}, a \Delta \right) \qquad [2]$$

in which Φ is an unknown function which remains to be determined. In order to be "rational," in the sense of "dimensionally homogeneous," any more specific equation purporting to describe the relation symbolized by equation [1], must be reducible to the form [2].

While equation [2] refers only to the process as it occurs at a single cross section where the liquid and the wall have particular temperatures, an equation of just the same form will hold for the mean transmission over the whole length of the tube, measured in B.t.u. per hour per running foot, if the temperature of the liquid does not change much along the tube and if the temperature of the tube itself is fairly uniform from end to end. We may then replace Δ by the mean temperature difference t_m and let ρ, μ, C, and λ be values for the mean temperature of the liquid in the tube.

A glance at equation [2] shows that a complete investigation of the problem of heat transmission, even for a single liquid in so

[1]Model Experiments and the Form of Empirical Equations, *post* p. 263.

simple an apparatus as a condenser tube, is a rather complicated affair. For to determine the form of the unknown function Φ, it is necessary to find the effect on Q of varying the four arguments of Φ separately. In view of our almost complete ignorance about the thermal conductivity of water, to say nothing of other liquids, it is not surprising that we do not yet know all there is to know about heat transmission; and any attempt to proceed rationally, as Mr. Wilson has done, by taking into account and allowing for the various physical quantities that may be involved in the process of heat transmission, is very much to be welcomed.

In any systematic attack on the general problem of heat transmission, whether by making new experiments or by analyzing the data already at hand, equation [2] or some one of the many equivalent forms into which it may be thrown, may be used as a guide. To illustrate, let us suppose that as a first guess we decide to leave the possible variation of conductivity with temperature as a refinement to be attended to later if necessary, and see whether we can not coördinate the observed facts, at least approximately, by assuming that these variations are of negligible importance. This amounts to setting $a = 0$, and equation [2] at once reduces to

$$Q = DV \rho C \Delta \Phi_1 \left(\frac{DV\rho}{\mu}, \frac{\mu C}{\lambda} \right) \quad [3]$$

in which Φ_1 represents a new unknown function, but one with only two, instead of four, independently variable arguments. Since the temperature head Δ has disappeared from within the unknown function, we now have $Q \propto \Delta$, as is very often assumed from the start in discussions of heat transmission.

Of the two arguments of Φ_1, the first or $(DV \rho/\mu)$, in which D is a length and V a speed, occurs very often in considering the motion of viscous fluids, and it would be convenient to have a name for it. When the geometrical shape of the solids in contact with the fluid is given—a smooth straight tube, in the present instance—the value of $(DV \rho/\mu)$ determines the nature of the motion, including the degree of turbulence, and it might, provisionally, be called the "turbulence variable." The other argument $(\mu C/\lambda)$ is a property of the fluid, and for a given fluid it depends, like other properties, only on the temperature and pressure. Physically speaking, it is the ratio of kinematic viscosity to thermal diffusivity. For lack of an obviously appropriate name, it might, in case of need, be termed the "convectivity" of the fluid.

If we write equation [3] in the form

$$\frac{Q}{DV\rho C\Delta}=\Phi_1\left(\frac{DV\rho}{\mu},\ \frac{\mu C}{\lambda}\right) \qquad [4]$$

the first member is susceptible of a simple physical interpretation. For it is easily seen that $\frac{Q}{DV\rho C\Delta}=\delta$ is the change in temperature of the liquid in moving a distance $\pi\ D/_4$, or one quarter circumference, along the tube. Hence we may write [4] in the form

$$\frac{\delta}{\Delta}=\Psi\left(\frac{DV\rho}{\mu},\frac{\mu C}{\lambda}\right) \qquad [5]$$

which states that the ratio of the temperature change in passing over any fixed number of tube diameters along the tube, to the mean temperature head Δ causing the change, depends only on the values of the turbulence variable $(DV\rho/\mu)$ and of the convectivity of the liquid $(\mu C/\lambda)$. The form [5] has the advantage over [4] that the temperature differences δ and Δ are among the quantities which may be directly measured during an experiment, and the meaning of the equation is physically plainer than that of [3] or [4].

Returning to equation [3] we may, if we like, replace Q by the somewhat artificial but practically convenient transmission coefficient U. For we have $Q = U\times\pi D\Delta$, so that [3] may be written

$$U=V\rho C\Phi_2\left(\frac{DV\rho}{\mu},\ \frac{\mu C}{\lambda}\right) \qquad [6]$$

If, as in the case of water in condenser tubes, we are dealing with a liquid for which ρ, C, and λ may be treated as sensibly constant over the working range of temperature, equation [6] may be simplified to the form

$$U=V\Phi_3\ (DV,\mu) \qquad [7]$$

If experiments at variable rates of flow show that $U \propto V^n$, equation [7] simplifies still further to

$$U=V^n D^{n-1}f\ (\mu) \qquad [8]$$

so that the effect of varying diameter is known at once from the effect of speed, without further experiment. To find the effect of viscosity, i.e. of temperature, the natural thing to do after finding the value of n is to plot observed values of $U\div(V^n D^{n-1})$ against simultaneous values of μ, draw a curve through the points, and thereafter make corrections or allowances for temperature by using the curve or an empirical equation for it. If the points do not lie on a single curve, within the errors of experiment, our neglect of the variations of λ with temperature was probably not permissible, and we must then take up a more complete analysis of the problem by treating the tube temperature as one of the variables.

Equation [1] may be treated in various ways and the problem considered from various standpoints. But the important things to be noted are: *first,* that in attacking such a problem in technical physics as that of heat transmission, it is important to start by recognizing all the physical quantities which may be involved in it, instead of trying to shut one's eyes to as many as possible; *second,* that an empirical equation which is not rational, i.e. not dimensionally homogeneous, is not safe; and *third,* that dimensional reasoning is very useful, both in suggesting and directing experimental work, and in analyzing experimental data so that they may be represented by rational formulas.

C. F. Braun (written). This paper is particularly interesting to the writer because it treats in a most thorough manner a factor in calorifier design, which is given only slight attention in my condenser[1] paper.

Unquestionably the resistance to heat flow on the fluid side of a tube increases rapidly with increase in viscosity and is a most important factor in the design of calorifiers for such viscous fluids as oils, syrups, etc., having viscosities ranging up to 50 times that of water. As the oil industry particularly is an important one in this country and the demand for oil heaters and exchangers large, the development of rational formulae relating to heat transfer and viscosity would be very valuable, and can the author extend his researches to these fields he will render a service to the profession.

The viscosity of water varies so slightly within the ranges of temperatures encountered in heater and condenser practice that its effect is, I feel, almost negligible compared with that of other variables indeterminate and difficult to control. While believing in figuring, I think we must eliminate unimportant complications, or we avoid and do not use our theory.

One of the greatest variables in determining the unit coefficient of heat transfer for comparison of different experimental results is the method of computing the mean temperature difference. Without discussing the merits of the various formulae, it is sufficient to call attention to the fact that the arithmetic mean, logarithmic mean and others produce widely varying results, particularly when

[1]The Surface Condenser, *post,* p. 203.

the ratio between the last temperature difference and the greatest temperature difference is small.

Again the amount of air present is a variable which cannot be controlled or its effect determined, and will invariably produce variations far beyond the limits of viscosity effects.

The density or mass flow of the steam, while apparently having no important effect upon heat transfer, may very probably have as much if not more effect than the viscosity of the water.

Also, I believe thoroughly that heat transfer varies inversely with tube size and to a far greater degree than with water viscosity. Experimental data on this point would be valuable.

Unaccountable discrepancies in condenser test results are very frequently the result of assuming a uniform temperature in the steam space, instead of obtaining an average which may fall considerably below that at the inlet, due to frictional pressure drop through the tube of space, and to the presence of air.

Arthur M. Greene, Jr. I would like to ask how the author proposes to get the mean temperature so as to use the results of his work, and also how he obtained the mean temperature he did use.

As Mr. Braun points out, using the arithmetic mean and the logarithmic mean gives two results; and if viscosity is affected by temperature, this must be considered in finding the coefficient of heat transfer. It seems that both methods of getting t_m are incorrect, and t_m should really be found by the method Mr. Orrok uses in his condenser work.

Mr. Leo Loeb's paper, in the Transactions of the Institute of Naval Engineers, points out clearly that this heat transfer coefficient varies with temperature. The method which Mr. Loeb uses is rather unique, and proves clearly, I think, that we must use some exponent form for computing t_m. This seems to me better than using an expression involving viscosity.

The Author. Professor Heck criticises the reference to the similarity between the laws of resistance to the flow of water and heat on the grounds of dissimilarity of exponents. The mere numerical value of these exponents is of little importance for the purpose in hand. It was the similarity in form of the accepted law that led to the study, and although this study showed the exponential law to be incorrect for heat transfer, nevertheless the remarkable facility with which all data are reconciled by the viscosity correction is justification enough for the method.

Professor Heck also objects to the use of the word "rational." My understanding is that an expression is rational when it so involves the different variables that the truth may be checked up by the fundamental dimensional equations. Taking the expression for film thickness in the paper, we have $L = (a/V) + b$ where L is the thickness, V, the reduced water velocity, a, the slope of the line, and b, the intercept on the vertical axis, Figs. 7 and 8. In this expression b has the dimension L; V is of course L/T; a is L/V or L^2/T, and we then have

$$L = \frac{L^2T}{TL} \div L = L$$

which is a rational expression. Referring to the expression for the area we have
$A = Q\,\rho\,L/t_m$ in which L is taken as the expression above; by definition $Q/t_m = U$, and $\rho = L/U$ from which

$$A = \frac{UL^2}{U} = L^2$$

which expression is again rational.

As for the influence of tube diameter to which Professor Heck takes objection, there seems no reason for endeavoring to find a more complicated function. The resistance to heat flow and water flow are both a function of the water agitation. In the flow of water the diameter is shown by Reynolds to have a certain influence on the critical point, that is the point at which the flow becomes turbulent, and this feature in turn affects the resistance to flow. It seems not unreasonable to expect the diameter to have the same sort of influence on heat transfer, which is dependent upon the agitation in the same manner. In comparing a ½-in. and a 1-in. tube as to the effectiveness of the unit surface it must be borne in mind that for the same linear velocity the quantity of water passed increases as the square of the diameter. If now the flow is turbulent, there can be a larger number of impacts on the unit surface of the larger tube than of the smaller, and this accounts for the greater effectiveness.

Professor Greene has asked the method of obtaining the mean water temperature. This is shown in the tables where I have subtracted the mean temperature difference t_m from the temperature of the steam t_s, thus getting t_w, the mean water temperature. Both Mr. Braun and Professor Greene have discussed the mean temperature difference; I am aware of the difference of opinion as to the proper way of getting this. Mr. Orrok in his calculations used the arithmetic mean for the reason that the difference in results by different methods when the temperature rise is small is not worth considering.

Mr. Loeb states that no useful purpose can be served by considering the high circulating water temperature in design where space and weight are limited. On the other hand this seems to me to be the very place careful design is needed. As an example take the condenser design of a battle cruiser developing say 120,000 horsepower, on a small displacement. Her condensers will be beyond the limits of present practice so that no comparison may be had with other vessels, yet, on the other hand, space will be so confined as to preclude guesswork.

Mr. Loeb calls attention to the fact that the variation in results reported by several investigators may be due to experimental errors, rather than the neglect of the variables mentioned. I appreciate fully the difficulties of work in this field and drew Fig. 9 as an example.

The objection by Mr. Loeb that the analogy of the law of the electric circuit can apply only when we have a single resistance is not sustained. We are fully justified in writing such a law if we remember that the resistance in this case is a variable. Mr. Loeb practically uses the same analogy in his paper, but instead of separating the resistance into its component parts he assumes that he is keeping this constant when he maintains the water velocity constant and determines an exponent for the mean temperature difference. In the general expression,

$$U = \frac{t_m}{R_w + R_s + R_n + R_t}$$

R_w is not dependent upon the velocity alone, but also on the mean water temperature, and his failure to consider the temperature is probably responsible for the value of his exponent for t_m, and its difference from unity. Any exponential law for heat transfer involving either V or t_m, however convenient it may be for the design of a limited class of apparatus, cannot be considered a valid general law. As Mr. Loeb says any one can run a few tests and use them as a basis for design for similar apparatus involving the same conditions; surely, however, we are not entitled to use these few results in writing a general law for the whole subject.

The objection that new types of apparatus have been evolved in which agitation is carried to the limit so that viscosity need not be considered is hardly valid. We have two distinct methods of reducing the resistance,—mechanical agitation and increased water temperatures. In certain types of apparatus the first method is carried to the limit, but this does not remove the influence of temperature on the thickness of the water film.

In regard to the value of the exponent of V in the paper, I have now plotted the results of Mr. Orrok's work against V with the in-

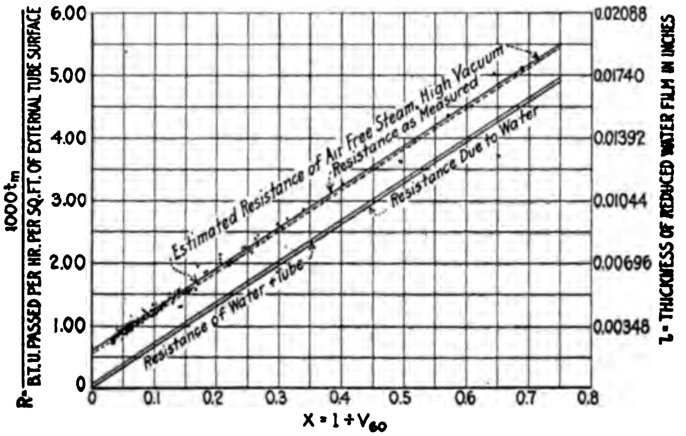

FIG. 11 REPLOTTED CURVES OF RESISTANCE TO HEAT TRANSFER IN A CONDENSER

dex unity and find excellent accord, as shown in Fig. 11. Further experimental work must determine the real value of the exponent.

In conclusion, successful design of heat apparatus has been done empirically and will continue to be so done. In all design, however, we must appreciate two great classes, (1) functional design and (2) the design of the mechanism. Mr. Braun in his paper[1] gives an excellent summary of the design of the mechanism of a condenser. My paper is intended to demonstrate a basis for rational functional design of heat transfer apparatus of all types for every flow condition, throughout the whole range of variation.

[1]The Surface Condenser, *post*, p. 203.

No. 1478

INFLUENCE OF DISK FRICTION ON TURBINE PUMP DESIGN

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SYNOPSIS OF PAPER

The mathematical survey of the problem leads to these conditions for a minimum loss through disk friction:

a Smoothness (polish) of both disk and casing. Roughness of either is equally detrimental.

b Smallest possible surface of both. Excessive extension of surface is equally detrimental whether it is the surface of the disk or of the casing.

c Outward indication of attainment of minimum is the fact that the waste-water rotates half as fast as impeller.

A gyrostatic pressure is generated by the rotation of the waste-water and added to static pressure prevailing at the center of the impeller.

From the influence of the width of the impeller it follows that it is important to keep the thickness of metal at the periphery as small as possible. Protruding rims are objectionable.

The influence of the ordinary roughness of non-machined castings has no perceptible effect on the efficiency except with high lift pumps. Painting or japanning the surfaces generally seems less desirable than machining them with a medium heavy cut. High polish seems wasted. The experiments verify conclusions of mathematical survey.

The influence of viscosity is proportional to its fifth root; yet, it is responsible for an improvement in the efficiency of hot-water turbine pumps. The effect of pumping heavy oil and tarry liquid is estimated. The influence of fluid density is almost exactly proportional to the specific gravity.

The loss through disk friction constitutes a constant percentage of the normal useful power at all speeds in one and the same pump. Generally its percentic value grows with the value of $\frac{\text{head per stage}}{\text{capacity}}$ at constant speed, and diminishes with increasing speed and constant ratio $\frac{\text{head per stage}}{\text{capacity}}$. High heads are more economically produced by high speeds or a greater number of stages than by increasing the diameter of the impellers, but the number of stages should be left

Presented at the Spring Meeting, Buffalo, N. Y., June, 1915, of The American Society of Mechanical Engineers. On account of limitations of space, the paper is here reproduced in an abridged form, approved by the Author. The complete paper is on file in the Library of the Society.

to the discretion of the makers, not fixed by specifications. A steep angle between impeller blades and the tangent at the periphery serves very considerably to improve the efficiency owing to indirect reduction in disk friction losses, especially in high lift pumps.

The disk friction reaches a minimum for a certain width of the waste water chamber, which is about ⅜ in. for disks of 12 in. diameter. The increase with increasing width is due

a To the increase in retarding surface.
b To the induction of secondary or induced hydraulic currents.

Concentric ribs are advantageous; radial ribs are detrimental.

In single-stage pumps the rotation of the waste-water reduces the tendency for leakage by about 20 to 35 per cent. In multi-stage pumps the same influence may even increase the leakage.

Inequality in shape or roughness of the waste-water chambers on both sides of the impellers produces a gyrostatic axial thrust due to disk friction which can assume very considerable values. The direction of this thrust is indicated by the rule: "The impeller is drawn to the side where the waste-water rotates fastest."

ABRIDGED PAPER

A disk revolving in water acts as a brake. The losses due to the rotation of impellers in the water surrounding them amount to one-quarter to one-third of the total of the losses occurring in a high-lift turbine pump. It is astonishing that publications on the nature and magnitude of these losses are more than scarce.[1,2,3,4,5] There is very little reliable information on the question; and that little is not arranged as designers would like to have it for use in commercial practice.

2 In the present paper the results gained by Professors Unwin[1] and Gibson[4] are mainly employed in an effort:

- *a* To find what means are at the disposal of the designer of turbine pumps for minimizing the losses due to disk friction.
- *b* To furnish reliable data and diagrams from which to estimate the influence of disk friction on the efficiency of a turbine pump.
- *c* To draw attention to the bearing which disk friction of the impellers has on the axial thrust of turbine pumps.

NOTATION

3 Referring to Fig. 1, the following symbols and terminology will be used:

κ_A and κ_B = coefficients of friction between surfaces A and B respectively on one hand and the fluid surrounding the disk on the other hand.

Waste-water* = fluid surrounding the disk.

f = Gibson's coefficient of disk friction.

v_r = abs. velocity of waste-water in ft./sec. at the distance r from the axis.

[1] "Experiments on the Friction of Disks Rotated in Fluid." Minutes Proc. Inst. Civ. Eng., London, vol. lxxx, p. 221 ff.

[2] Bulletin No. 2, Univ. of Cal., Berkeley, Cal., 1887.

[3] Wirkungsweise der Kreiselpumpen; Mitteilungen ueber Forschungsarbeiten, Verein Deutscher Ingenieure, Heft 42, Berlin 1907.

[4] Min. Proc. Inst. Civil Eng., London, vol. clxxix, 1910, part i.

[5] Journal of Electricity, Power and Gas, San Francisco, Dec. 3, 1910. Joseph Le Conte, "Friction of Flat Discs Rotated in Water."

v_ρ = absolute velocity in ft./sec. of a point situated on the surface of the disk at a distance of ρ from the axis.

$v_{\rho A}$ = abs. velocity in ft./sec. of adjacent particle of waste-water.

$v_{\rho B}$ = abs. velocity in ft./sec. of waste-water particles adjacent to casing at distance ρ from axis.

O_A = area in sq. ft. of that part of the impeller surface which is marked by fat contour in upper half of Fig. 1.

O_B = area in sq. ft. of surface of casing as marked by fat contour in upper half of Fig. 1.

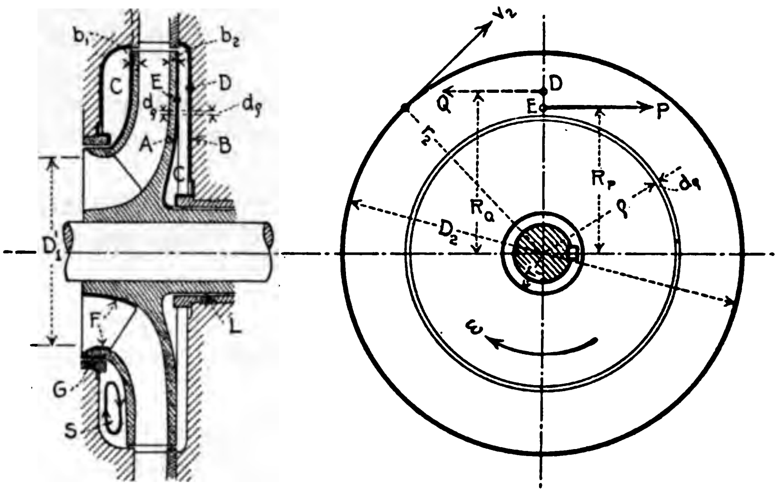


FIG. 1 KEY TO NOTATION

P = dragging force in pounds of impeller, i.e., force exerted by impeller on waste-water which it causes to rotate.

Q = retarding force, in pounds, of casing, resisting effects of P.

ω_T = angular velocity per sec. of waste-water.

ω = angular velocity per sec. of impeller.

J_A = polar moment of inertia in ft^4, of surface O_A with respect to axis of rotation.

J_B = the same of surface O_B.

R_p and R_q = distance, in feet, of P's or Q's place of application respectively from axis of rotation.

E_r = energy of disk friction in ft-lb.

h_g = gyrostatic head in ft.

h_s = static head generated by impeller proper in ft.

D_1 = diameter of slip ring clearance in ft.

D_2 and R_2 = diameter and radius respectively of impeller in ft.

P_g = total gyrostatic force in lb., see equation [1].

P_s = total axial force exerted by gyrostatic plus static pressure on the impeller in lb., see equation [2].

N_T = number of revolutions of waste-water, per min.

N = number of revolutions of pump per min.

M = moment of frictional force in ft-lb.

L = friction loss in ft-lb. per sec.

n = Gibson's functional exponent of v.

b_1 = axial thickness of left hand margin of impeller in same measure as D_2.

b_2 = axial thickness of right hand margin of impeller in same measure as D_2.

b = total thickness $= b_1 + b_2$.

L_b = loss due to b, in ft-lb. per sec.

h.p. f^1 = loss due to friction on both faces of disk without considering axial thickness, in h.p. (see formula [3]).

h.p. f_b = loss due to friction on circumference of disk or impeller (see formula [4]).

h.p. f = h.p. f_b + h.p. f^1 (see Fig. 6).

μ = absolute viscosity in grams per cm-sec. (= 0.010 for water of 65 deg. fahr.).

w = density or specific gravity (= 1 for water of 40 deg. fahr.).

s = lateral distance between wall and disk.

k = ratio of roughness of disk to roughness of wall.

k^1 = k × constant (see formula [6]).

v_{crit} = critical speed of flow.

*The term "waste-water" was chosen in order to specify unmistakably this part of the contents of the pump. It is the fluid that has leaked out from the clearance between the circumference of the impeller and the diffusor. It does not participate in the pumping process proper. The water which fills the eye of the impeller (see fat contours marked F in lower half of fig. 1) does participate in that process. The friction of that part of the impeller skin is not booked under disk friction and is not dealt with in this paper.

DEDUCTIONS

4 *a* In order to reduce the loss through disk friction as much as possible, the coefficients of friction κ_A and κ_B should be made as small as possible; i.e., both the casing and the runner should be machined or polished as smoothly as possible.

b If a smoothly finished impeller revolves within a rough casing, the result is the same as if a rough impeller revolves within a smooth casing.

c The surface of the impeller and of the stationary parts of chamber *C* (Fig. 1) should be made as small as possible.

d The effect of an extended and complicated surface is equally bad whether this surface be the stationary wall of chamber *C* or the rotating surface of the impeller.

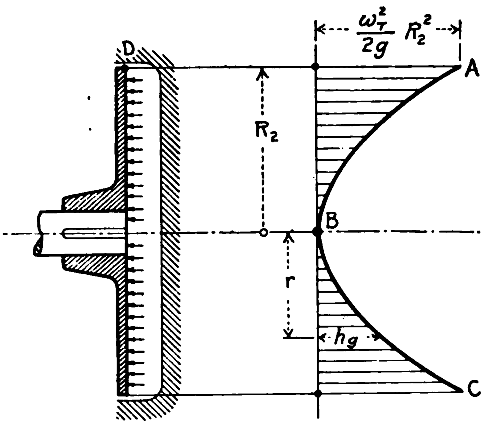


FIG. 2 GYROSTATIC PARABOLA

The outward indication of the attainment of a minimum of loss through disk friction is the fact that the waste-water rotates just half as quickly as the impeller.

THE GYROSTATIC PRESSURE

5 *The pressure due to the rotation of the waste-water is the source of important axial forces.* At a distance of *r* ft. from the axis of rotation and an angular velocity of ω_T per sec. this pressure in feet of fluid is

$$h_g = \frac{\omega_T^2 r^2}{2g} = \frac{v_T^2}{2g}$$

where v_T stands for the absolute tangential velocity of the waste-water in ft-sec. and $g = 32.16$.

6 When plotting these pressures as abscissæ over the respective radii as ordinates, the parabola $A\ B\ C$, Fig. 2, is obtained. The pressure h_g is directed towards the circumference.

7 In turbine pumps the problem is to find the influence of the gyrostatic pressure upon annular surfaces, say, e.g., that part of the impeller which lies between slip ring a and its periphery (Fig. 3).

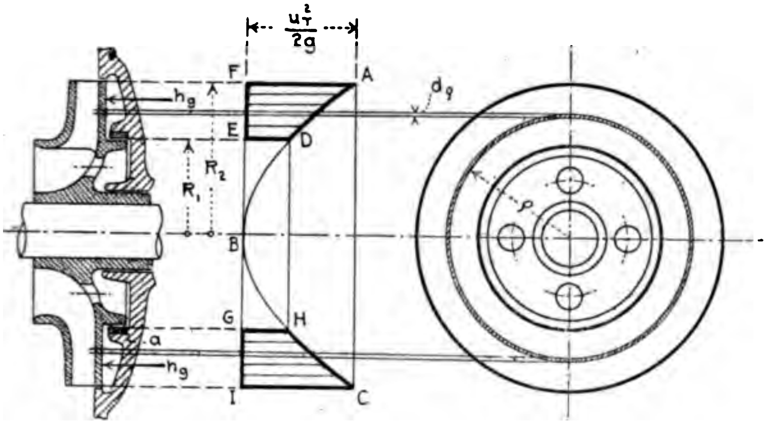


FIG. 3 GYROSTATIC FORCE ACTING UPON ANNULAR SURFACE

8 *The gyrostatic force in pounds axially exerted by the rotating waste-water upon an annular surface is*

$$P_g = \frac{N_T^2 (D_2^4 - D_1^4)}{1000} \qquad [1]$$

where

N_T = rotative speed of waste-water in r.p.m.

D_2 and D_1 = the outer and the inner diameter, respectively, of the surface in ft.

9 The *total axial force*, therefore, is

$$P_s = 49(D_2^2 - D_1^2)\left(h_s - \frac{4.25\ N_T^2\ D_2^2}{100{,}000}\right) + P_g \qquad [2]$$

where

D_2 and D_1 = outer and inner diameter, respectively, of annular face, in ft.

h_s = static head existing at circumference of impeller, in ft.

N_T = rotative speed of waste-water in r.p.m.
P_g = gyrostatic force (see equation [9]).

V. ESTABLISHMENT OF PRACTICAL FORMULA

10 The paper here surveys the experiments of Unwin and Gibson, considering the limitations of their formulae and also that of Biel for the theoretical loss through disk friction. It then develops the expression

$$\text{h. p.} f^1 = \frac{4\,\pi^{n+2} f}{550\,.\,30^{n+1}.\,2^{n+3}}\,N^{n+1}.\,\frac{D_2{}^{n+3}}{n+3} \qquad [3]$$

where

h.p. f^1 = loss due to friction on both faces of disk, without considering axial thickness, in h.p.
f = coefficient of friction, see Fig. 4, *a*, *b* and *c*.
n = exponent of friction, see Fig. 4, *a*, *b* and *c*.
N = r.p.m. of disk.
D_2 = largest diameter of disk.

11 This formula does not take into account the axial extension of the disk. Gibson introduces, for this purpose, an "Effective Radius" R_1 which is slightly larger than the actual radius of the disk so as to make up for the extra loss produced by the circumferential face of the disk. The author has found this correction to be rather inconvenient for use in connection with turbine pump impellers. He therefore developed a corrective factor specially adapted to the needs of the designer as follows:

$$h.p.f_b = h.p.f^1 \times \left[\frac{b}{D_2}\cdot(n+3)\right] \qquad [4]$$

That is, in order to find the total loss due to *friction both of the circumferential and the lateral faces* of a disk, the value of h.p. f^1 obtained for the latter (equation [3]) should be multiplied by

$$1 + \frac{b}{D_2}(n+3) \qquad [5]$$

where b is the total axial thickness of the disk, expressed in the same measure as D_2.

12 In this way the designer can always, by a simple factor to be worked out mentally, consider separately and keep apart the

influence of the diameter and that of the axial extent of an impeller on the friction loss.[1]

INFLUENCE OF ROUGHNESS OF DISK AND CASING

13 Fig. 4 *a, b* and *c* represent graphically the average variations of f and n in function of v_2* as established by Gibson and Ryan for various degrees of roughness. Their results were compared with those established by Professor Unwin's experiments and found to harmonize with them to a satisfactory degree. As set forth in Fig. 4 *a, b* and *c* the values of f and n furnish, within the range of speeds at which turbine pumps are run, figures for the losses (equations [3] and [4]) which always come within 5 per cent of the actual test results.

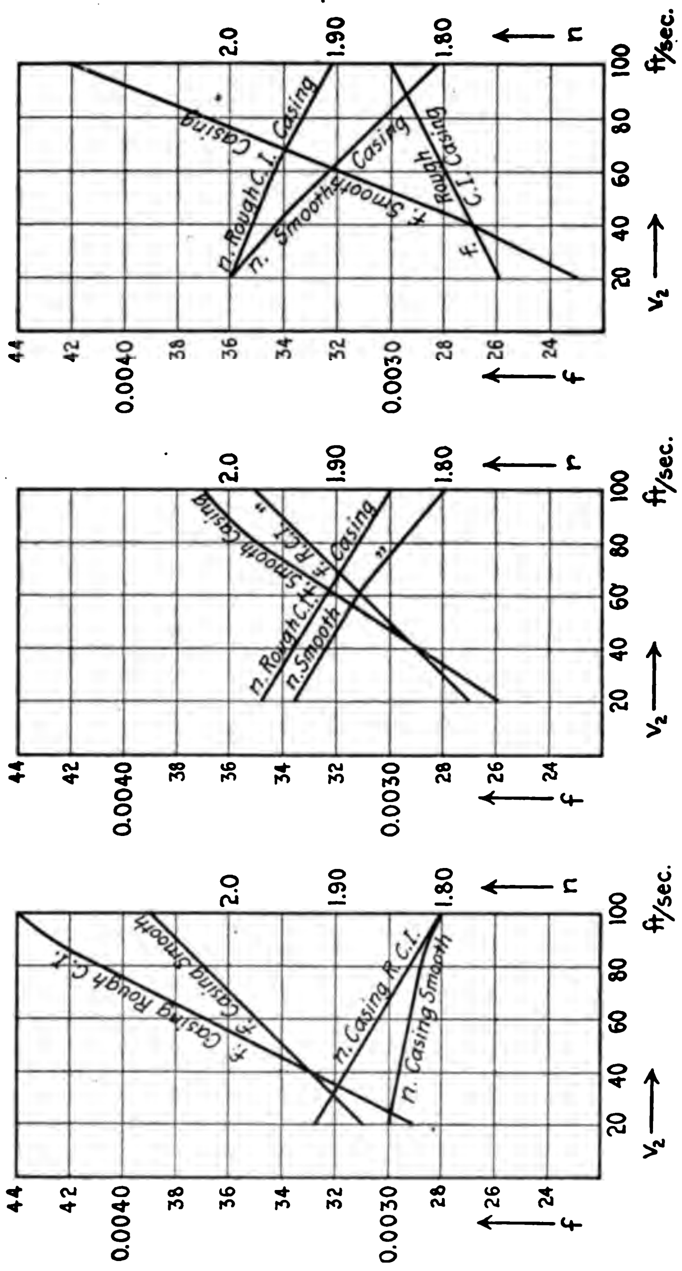
14 The second deduction at which the author arrived by the mathematical survey in this paper, viz.: "if a smoothly finished impeller revolves within a rough casing, the result is the same as if a rough impeller revolves within a smooth casing," is well corroborated by these curves. Gibson and Ryan carried out a series of special tests dealing with this question, the results of which are contained in Table 1.

TABLE 1 LOSS THROUGH FRICTION OF DISKS 1 FT. IN DIAMETER AND OF VARYING ROUGHNESS, REVOLVING WITHIN ⅝ IN. LATERAL DISTANCE FROM COVERS OF CASING OF VARYING ROUGHNESS

	Friction loss (h.p.f+h.p.f_1 equations 12 and 14)		
	h.p.		Per cent of minimum loss
	N = 1500 r.p.m.	N = 2000 r.p.m.	
Polished brass disk in varnished casing........	0.928	2.07	100
Varnished cast iron disk in varnished casing....	0.923	2.06	100
Rough cast iron disk in varnished casing.......	1.11	2.49	120÷125
Polished brass disk in rough cast iron casing ...	1.11	2.49	120÷125
Varnished cast iron disk in rough cast iron casing.	1.13	2.57	122÷125
Rough cast iron disk in rough cast iron casing....	1.25	2.9	135÷140

[1]This correction is silently based on the assumption that f and n have the same value for the circumferential face as for the lateral surfaces. They really are slightly smaller there. Still, considering all uncertainties that are connected with the application of the experimental values to practical shapes of impellers the inaccuracy is trifling.

*v_2 is the peripheral velocity of the disk, in distinction from the value v= mean, or one-half the peripheral, velocity as employed by Gibson.



a. Polished Brass Disk.

b. Painted and Varnished Cast-Iron Disk.

c. Rough Cast-Iron Disk.

Fig. 4 Diagrams of Disk Friction on Polished, Painted and Rough Surfaces

15 This table further shows that the effect of metal polish and of varnish is almost exactly the same. This result should, however, not encourage manufacturers to resort to varnishing rather than machining the surfaces. Varnishing or japanning appears only at first sight to be the cheaper process. As a matter of fact, two coats of varnish require rather a long time to dry thoroughly. If the varnish or lacquer is not applied very carefully and dried well, or if the fluid to be pumped is not absolutely clean, the varnish will peel off. The surface then acts worse than even a rough casting. Polished surfaces, on the other hand, will often lose much of their original finish by the sediments of the water in the pump. It is, therefore, of no great use to expend much time and wages on a high, shining exterior finish of the impellers. It is sufficient to turn the faces off with a medium heavy cut, especially as the turning ruts are running concentric, i.e., in the direction of the flow of the waste-water.

16 Much saving can be effected if designers and shops coöperate in producing the smoothest possible castings to surround the waste-water chamber. This chamber should offer a minimum of surface and be free of any ribs, protrusions or recesses which not only increase the surface and impede the rotation of the waste-water, but also frequently give rise to rough castings. If the designer gives the walls of the chamber a shape of utmost simplicity, the foundry will be able to produce faultless, smooth castings without extra cost.

17 Considerable though the differences due to various degrees of roughness may at first seem, still it is only with typical high lift pumps that they appreciably affect the efficiency of the pump.[1] This is especially true where the duty is small in comparison with the head per stage.

INFLUENCE OF VISCOSITY, TEMPERATURE AND SPECIFIC GRAVITY OF FLUID ON DISK FRICTION

18 Inasmuch as oil, gasolene, hot water, etc., are today pumped by turbine pumps, the consideration of these influences gains in importance. The author has found it possible to answer the question of their respective magnitudes by a simple application of the theory of dimensions, without resorting to experiment.[2]

[1]This holds good, of course, only for pumps while new. Bad water often causes the formation of very heavy sediments, which act very detrimentally on the disk friction and lower the efficiency.

[2]See also Author's closure.

19 By considering the dimensions upon which the magnitude of the resistence depends, the expression

$$\text{h.p.}f^1 = k^1 \cdot \sqrt[5]{\frac{\mu \cdot w^4}{s}} \cdot v^{2 \cdot 8}\, d^2 \qquad [6]$$

where

μ = absolute coefficient of viscosity in grammes per centimeter-second.

w = specific gravity

is deduced. It is easily seen that this is substantially identical with formula [3].

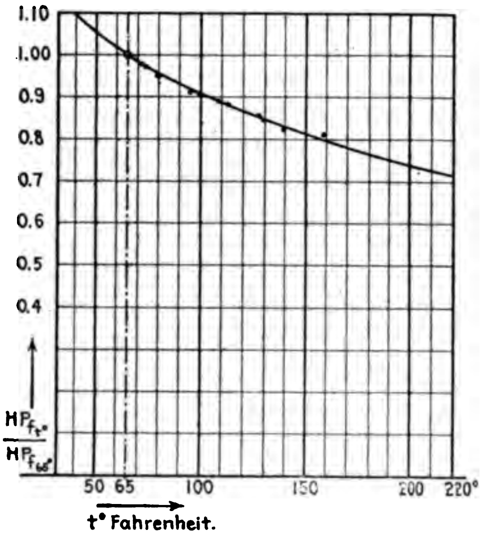


FIG. 5 DISK-FRICTION LOSS AS FUNCTION OF TEMPERATURE OF WASTE-WATER

20 Fig. 5 represents the variation of h.p.f with the temperature of the water.[1]

21 This curve may be utilized for estimating the improvement in power consumption which may be expected when a pump which has been tested with cold water (65 deg. fahr.) is eventually used, say, as a boiler feed pump and deals with water of 200 deg. fahr.

22 Of course, in a pump all the other causes inherent to

[1]Influence of viscosity disappears when $n = 2$, i.e. when the sides of the waste chamber are very rough. A sensible reduction of frictional resistance can only be expected when walls and impellers are very smooth.

the change of temperature are at work at the same time. For instance the loss of head owing to hydraulic friction in the fixed and rotative channels of the pump is likewise diminishing. On the other hand the leakage loss must increase as the fluid becomes less viscous. The balance of all influences tends slightly to improve the efficiency of a boiler feed pump in actual service as compared with the figures obtained when testing it with cold water.

23 The paper gives two examples illustrating the influence of heavy oil and tarry liquid on the disk friction in a turbine pump.

INFLUENCE OF DIAMETER AND SPEED

24 Formula [3] with $n = 1.8$ to 2, reveals at once the tremendous role the diameter plays. The influence of the rotary speed N is of relatively minor importance, though it is in proportion to about the third power of the number of revolutions.

25 As the duty of a turbine pump is in proportion to the speed, and the head in proportion to its square, the useful output is increased in proportion to the third power of the speed. It will be seen, therefore, that the *loss through disk friction must form a constant percentage of the normal useful output, (water-h.p.), of a given pump irrespective of speed.*

26 *High heads per stage are more economically produced by applying high speed than by using a large diameter of the impeller.* This economic tendency for the high-speed pump is fostered by a number of other incidental advantages, e.g., alternating current electric drive, direct steam turbine drive, small unit costs, small weight, reduction in tensile stress on casing, compactness, ease in transport and repairs, etc.

27 It might not be out of place to point out that *the angle between the impeller blades and the tangent at the periphery* has an important bearing on the loss through disk friction. If this angle is made 20 deg. the diameter must, from hydraulic reasons, be larger by 10 to 25 per cent,[1] than if the angle were 45 deg. This implies for the flatter angle an increase in disk friction of at least 60 to more than 150 per cent in some cases. As the disk friction loss expressed in percentage of the useful work amounts seldom to less than 5 and often to more than 10 per cent in high lift pumps, it is evident that the mere changing of the vane angle from 20 to 45 deg. implies a

[1]Ten per cent in pumps with guide-wheels, up to 25 per cent in pumps with a more or less efficiently shaped volute.

profit of from 3 to 8 per cent and even more in the efficiency of such pumps. The reason why many designers like to choose flat angles of about 20 deg. is that the water then leaves the runner at a relatively high static pressure and low speed, i.e., that little or no attention needs to be paid to the balance of kinetic energy. Diffusors or guide wheels may be dispensed with, and eddies due to an improperly simplified design of the casing cannot do much harm. Still, competition in regard to efficiency between various makes and the urgent demand of the salesmen to reduce the motor power in order to minimize the cost of the unit to the consumer gradually force the manufacturer and designer to abandon their preference for flat angles and to utilize the advantage that results from the consequent reduction in diameter and disk friction.

28 In order to enable designers and manufacturers to estimate the loss due to disk friction without resorting to formula [3] which is rather unwieldy for practical use, the author plotted the diagram Fig. 6, which should be self-explanatory. In basing the chart on the highest values found by Gibson and adding 15 per cent for friction of rim (supposed to be 3 per cent of D_2 thick) the author felt he should come nearest to conditions as they obtain in practice.

29 The author here gives some examples serving to illustrate the value and the mode of using this figure; the following example is a typical one:

30 Suppose 250 gal. per min. were to be pressed against a total head of 600 ft. and the speed of about 1800 r.p.m. had to be adhered to, say on account of 60 cycle a.c. current motor drive. What number of stages should be chosen?

31 As 200 ft. per stage is too much for that capacity, either a 4-stage, a 5-stage or a 6-stage pump must be employed. By using Fig. 6 the following table (Table 3) can be compiled without difficulty.

TABLE 3

INFLUENCE OF NUMBER OF STAGES ON PERCENTIC VALUES OF LOSS THROUGH DISK FRICTION

Q = 250 gal. per min. N = 1800 r.p.m.	Head per stage in ft.	Water h.p. per stage	Loss through disk friction	
			In h.p. per stage	In percentage of water h.p.
4-stage pump........	150	9.5	1.62	17
5-stage pump........	120	7.6	0.98	12.9
6-stage pump........	100	6.3	0.66	10.5

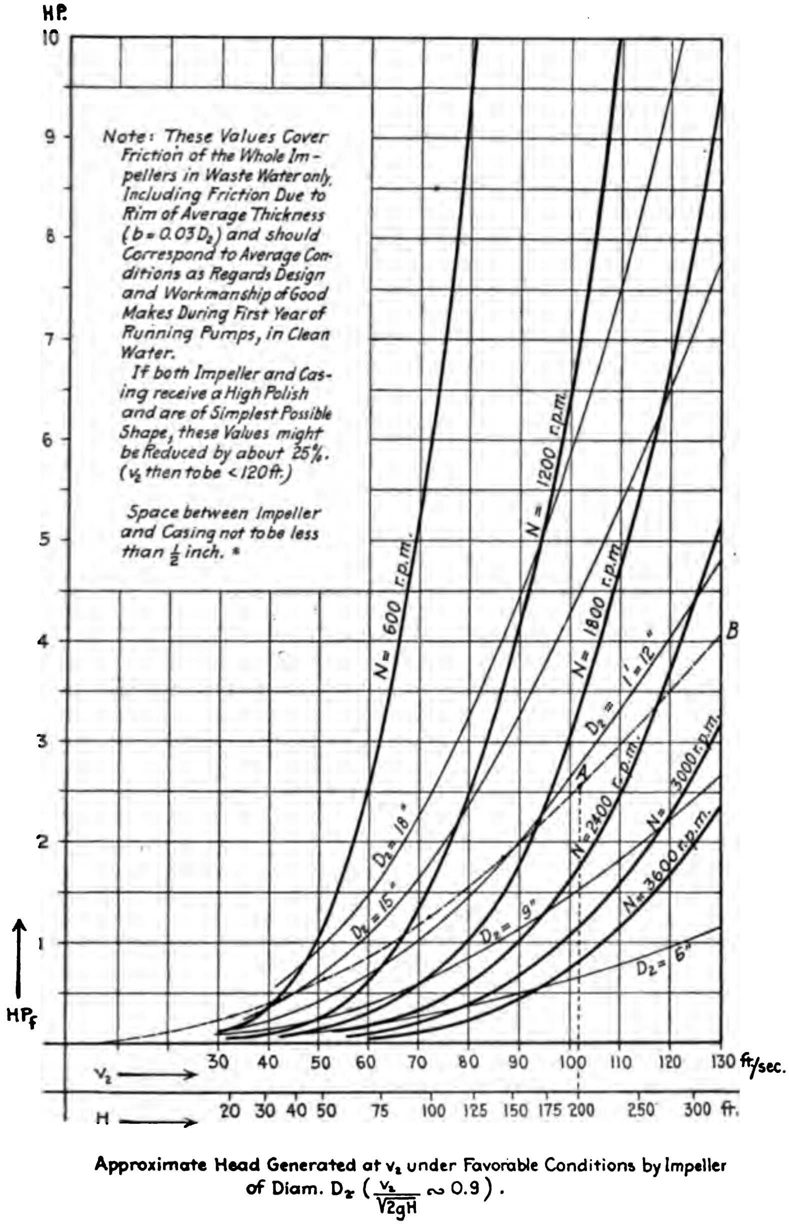


FIG. 6 LOSS THROUGH DISK FRICTION IN FUNCTION OF DIAMETER AND SPEED. BASED ON FORMULAE 3 AND 4

* Per foot of Diameter.

32 If the 4-stage and the 6-stage pump were designed equally well, the 6-stage pump would have an inherent advantage, due solely to the number of stages, of about 6.5 per cent of the water-horse power, or about 4 per cent in efficiency. The first cost would be increased in proportion to the number of stages and it would be a question of the cost of energy to the consumer whether the 4 per cent advantage in efficiency would outweigh the excess in price of the 6-stage pump over that of the 4-stage pump.

33 Looking at the question from the manufacturer's point of view, it might be possible, by abandoning certain niceties in design, for instance, the expensive guide vanes, to turn out a 6-stage pump actually as cheaply as the more elaborate 4-stage pump. If more than 4 per cent of efficiency was not sacrificed by that simplification the 6-stage pump would be as advantageous as the 4-stage pump —leaving out the question of number of spare parts, dimensions and weight.

34 One more point comes in for consideration: an impeller designed for 250 gal. only and required to pump against a 150-ft. head per stage would have to be large in diameter. As the duty is comparatively small, the passages at its circumference become rather narrow—too narrow perhaps for the cores to be properly fixed in the mould, or the passages to be well filed out in the shops. For this reason alone the head per stage is limited. The admissible minimum number in the present instance is probably 5-stage. The competition then stands between the 5-stage and the 6-stage pump. It is evident that it will be a difficult decision whether the slight increase in efficiency due to the one stage more will warrant the extra cost of that stage. Leaving out the possible advantages in manufacturing costs which might result from a greater output, a skilful designer may be quite able to surpass the efficiency of a competitive 6-stage pump with a 5-stage pump of like or even lesser cost per stage though the former has a natural advantage of 2.5 per cent less in disk friction.

35 At any rate the question will be of such nicety as to make it impossible for the buyer, judging merely from the number of stages, to have a sufficient insight into the merits of a pump. *Specifications therefore, should not fix the number of stages but leave this point to the discretion of the manufacturer.*

INFLUENCE OF LATERAL DISTANCE BETWEEN CASING AND DISK

36 Fig. 7 represents a diagrammatic compilation of the results

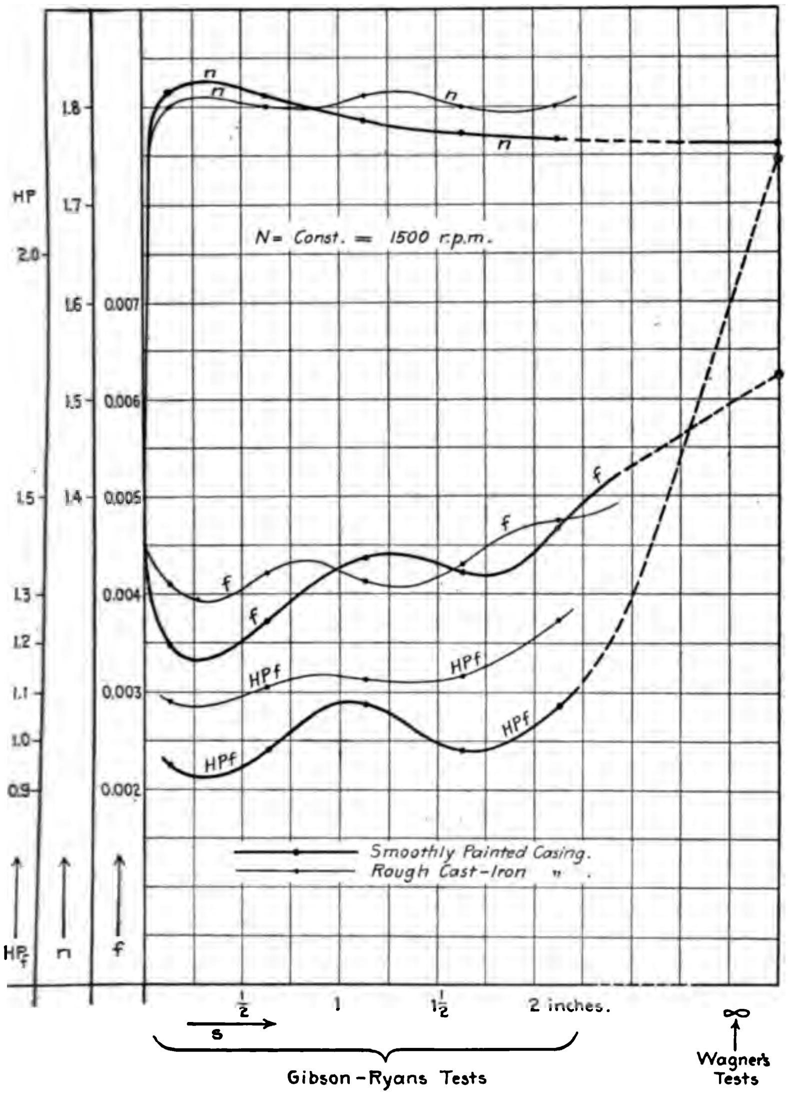


FIG. 7 INFLUENCE OF LATERAL DISTANCE BETWEEN CASING AND POLISHED 12-IN. DIAMETER DISK

which Gibson and Ryan obtained when varying the lateral distance, s, with those found by Wagner when rotating a disk in practically an unlimited basin ($s = \infty$) and by Dr. Becker[1] when rotating a polished piston within a very narrow polished casing.

37 From the diagram it is clear that the loss due to disk friction has, in general, a rising tendency with increasing width, s, of the waste-water chamber. This tendency is, for the larger widths s, due less to a variation of the exponent of friction, n, which remains fairly constant at about 1.76 to 1.8, than to a rise in the coefficient f.

38 Nothing can illustrate better than this diagram the necessity of using every caution in deriving conclusions from any mathematical survey of hydrodynamic problems.

39 The designer will draw two conclusions from Fig. 7. First, there is in general a tendency for the disk losses to rise when the width of the waste-water chamber is increased.[2] (This, by the way, follows also from the mathematical survey in this paper, for an increase of s implies an increase of O_B).

40 Second, concentric circular protrusions or ribs are not objectionable and might even be of advantage by breaking the secondary currents; while radial ribs must increase the disk losses because they favor the generation of induced currents and additional losses caused thereby.

INFLUENCE OF LATERAL DISTANCE AND ROUGHNESS UPON THE AXIAL THRUST

41 Professor Gibson, when investigating the effects of radial vanes attached to the impeller, measured the pressures generated by them. Incidentally he made a few readings from the gage attached for that purpose at the casing (which had an internal diameter of 13 in.) while rotating smooth disks. The author found that these accidental by-products of Gibson's tests are of the utmost importance for explaining and, therefore, mastering some components of the axial thrust which for a long time remained inexplicable to him and to many others.

[1]Bull. No. 48, Mitteilungen ueber Forschungsarbeiten, Verein Deutscher Ingenieure, Berlin, 1907.

[2]Whether the width indicated by Fig. 7 as connected with a minimum of loss, viz. $s \backsim \frac{3}{8}$ in., is a constant for all sizes of disks, or a function of the diameter, cannot be concluded from the experimental evidence at hand. The Author is inclined to conclude from mechanical reasons that it is a constant function of the diameter i. e. $s_{best} \backsim$ 3 hundredth's of D_2.

42 In Fig. 8 the lower three curves (curves of h_g) are plotted from Gibson's results. They represent the pressure difference due to the rotation of the waste-water between the center and points at a radial distance of 6½ in. from it.

43 From these readings it is possible to work out the rotary speeds of the waste-fluid, N_T, which, for the particular value of $r = 0.541$ ft. obtaining in Gibson's apparatus, is

$$N_T = 215 \sqrt{h_g}$$

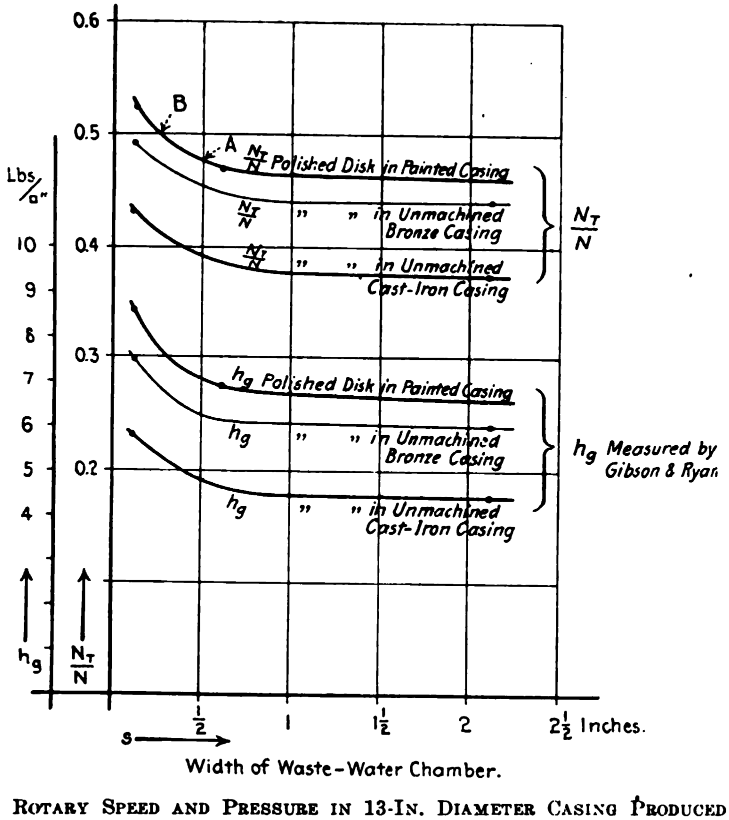


Fig. 8 Rotary Speed and Pressure in 13-In. Diameter Casing Produced by 12-In. Diameter Disk Running at 1200 R.P.M., in Function Width of Waste-Water Chamber

44 By far the most important consequences follow from Fig. 8 when used to estimate what axial pressures will result if the roughness of disk or casing, or their distance from each other, differ on the two sides of the impeller.

45 Equation [2], furnishes the total axial force resulting from the coöperation of the static pressure h_s generated by the impeller vanes at the circumference, and the gyrostatic pressure h_g. If the rotary speed of the waste-water N_T is different on the two sides of the same impeller an axial thrust results. This thrust is most striking with so-called balanced impellers, i.e., either double inlet runners, or runners having a slip ring on the back and borings leading from the inlet chamber through the eye into the chamber thus formed on the back (see Fig. 9). It shall be supposed that the static pressure generated by the impeller vanes is identical on both sides of the impeller. (This does not exactly hold true for single-inlet impellers for reasons which have no bearing upon the problem treated in this paper). When making the above assumption h_s (see equation [2]) is eliminated when subtracting the total axial force on one side from that on the other, and the effect of the gyrostatic phenomenon can be treated independently of any other causes which might affect axial thrust.

46 After subtracting the two values which equation [2] assumes for two different rotary speeds of the waste-water bodies on both sides of the impeller, the following equation results:

$$P_{s-s^1} = P_g - P_g^1 - \frac{4.25 \times 49}{100000}(D_2^2 - D_1^2)\,D_2^2\,(N_T^2 - N_{T1}^2) \qquad [7]$$

Introducing the values for P_g and P_g^1 which result from equation [9] this final expression* is obtained for the differential axial force:

$$P_a = P_{s-s^1} = \frac{N_T^2 - N_{T1}^2}{1000}[2.08\,(D_2^2 - D_1^2)\,D_2^2 - (D_2^4 - D_1^4)] \qquad [7^a]$$

where

N_T and N_T^1 = r.p.m. of waste-water bodies on two sides of impeller.

D_2 = outer diameter of impeller in ft.

D_1 = diameter of clearance of slip-rings in ft.

P_a = axial thrust resulting from difference between N_T and N_T^1, in lb.

P_a^1 depends on the fourth power of the diameter. No appreciable inaccuracy is caused by disregarding the central part of the impeller.

47 Fig. 9 shows the direction of the resulting thrust P_a. The difference in pressure between axis and periphery is greatest where the waste-water rotates quickest (Fig. 9: right hand side—

*The two members of the square brackets are reversed in sequence in order to make the contents of the brackets a positive value.

owing to small distance s). The static pressure h_s is the same on both sides. On the side where the waste-water rotates faster the gyrostatic pressure cuts out a greater parabolic area than at the other side. Therefore the total weight of pressure is smaller on the side where the waste-water rotates faster. Perhaps the following rule is easier to keep in mind: "*The impeller is drawn to the side where the waste-water rotates fastest.*"

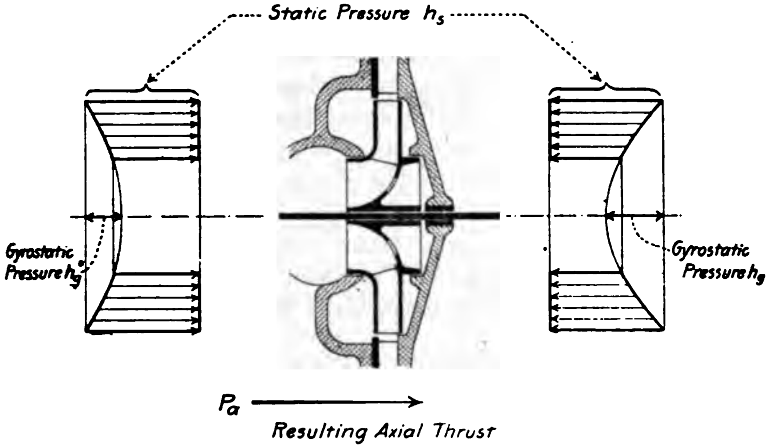


FIG. 9 DIAGRAM INDICATING AXIAL THRUST DUE TO EFFECT OF ROTATION OF THE WASTE-WATER

48 Two more examples, intentionally chosen to emphasize the necessity of the designer's attention to gyrostatic axial thrust, are here worked out in the paper. They show that an axial thrust of 440 lb. can be easily accounted for by the gyrostatic pressure differences forming in a 4-stage pump with impellers 1 ft. in diameter and the lateral width of the waste-water chambers $\frac{1}{2}$-in. on one and $\frac{1}{4}$-in. on the other side of each impeller. Mention is made that further investigation of the details of the curves in Fig. 8 would be very desirable.

49 The gyrostatic axial forces, though important, do not constitute the only hidden cause of axial thrust. There are other variable components of the axial forces due, for instance, to the flow of the leakage water, or to the difference in pressure within the cells of the impeller, etc. Designers may never hope to obtain perfect balance under more than one particular condition or independent of wear and tear. They must always rely upon some strong hydraulic

balancing mechanism or thrust bearing. The value of the above formulæ, therefore, lies principally in the possibility to foresee, by their aid, any danger that the axial thrust of the pump may act in the direction opposite or in addition to that for which the balancing device or thrust bearing is designed. It is known what strong axial forces can be exerted by disks provided with radial ribs. In this connection the results which Professor Hesse[1] derived from investigating hydraulic footsteps offer some valuable data to the designer, as does the paper of Gibson and Ryan. Not so well known, but no less remarkable is the effect of radial vanes in the fixed walls of the waste-water chamber. From the mathematical survey it is evident that their effect must about equal that of rotary vanes. The author knows of the experiences of a well-known pump manufacturer who had provided radial ribs in one, but not in the other waste-water chamber of a perfectly symmetrical double inlet impeller. The strong axial thrust which resulted was reduced to almost nil after the pockets between the ribs were covered by means of sheet iron, and incidentally the efficiency was considerably increased.

50 In case a designer should find it difficult for general constructive reasons, to shape both waste-water chambers exactly alike, he will, on the basis of the information contained in this paper, be able to counterbalance the effect of any differences in roughness and shape. It is scarcely necessary for the author to elaborate on this question which must be solved in a different way in each individual case.

DISCUSSION

C. George de Laval (written). The author draws the conclusion that roughness of surfaces of impellers and internal parts does not to an appreciable degree affect the efficiency of the pumps. In late researches by the National Physical Laboratory in England, it has been shown that the change from lamellar motion of liquid to eddying motion takes place suddenly at a definite value of the critical velocity directly proportional to the kinematical viscosity of fluid. The conclusions by Lord Rayleigh indicate that, by the principle of dynamical similarity, the relation of bodies immersed in fluids moving relatively to them is a general law of resistance of bodies and depends on the velocity, density and kinematical viscosity of the fluid expressed by the formula

[1]Bulletin No. 2, Univ. of Cal., Berkeley, Cal., 1887.

$$R = \rho v^2 F \frac{(\mu d)}{s}$$

where d = section or diameter of casing or pipe

R is resistance per unit area

S = specific gravity

F is a function of the one variable $\frac{\mu d}{s}$

ρ = density of water

v = mean velocity of water

μ = viscosity

The information given by the author is exceedingly interesting, particularly in the light thrown on the subject of fluids affected by viscosity, temperature and specific gravity. There is no doubt that little is known of this, not only on movement through pipes but also in chambers of centrifugal pumps. It is no doubt difficult to obtain a practical formula for use in dealing with very viscous liquids, nor can one estimate even approximately what can be expected in the way of power, speed and capacity.

The writer has found that results obtained are at considerable variance with what has been expected from mathematical conclusions based on existing laws of water. It would be most helpful if a commission were appointed to conduct a series of experiments to determine frictional resistances of oil and heavy viscous liquids in pipes, centrifugal pumps and reciprocating pumps, and so finally establish a practical formula which could be relied upon. The oil industry is so great and important that more research information should be made available to the engineering profession than is now. Today we have not even a standard viscometer which is satisfactory; nor is a final method settled upon as to basis of figuring results and comparing with water.

The guaranteed performance of a centrifugal pump for oil is always based on clear water, the friction of oil in pipes and pump being assumed on this basis. The final results in pumping oil with a specific gravity of 0.87 show an increase of power of about 20 per cent and a reduction in head of about 20 per cent. The internal disk friction of impellers on account of viscosity of the oil causes the efficiency to drop 25 per cent. Theoretically the pumping head, whether oil or water, should be the same, but owing to the viscosity of the liquid there is considerable heat generated by the impellers in the pump chamber and the friction loss of the liquid increases, due to the viscosity in impeller passages, so that the head produced is about 20 per cent less than with water. It is, therefore, necessary that experiments be resorted to as a considerable number of errors will creep in and give misleading results.

The author claims that he answers these important questions by theory without resorting to experiments. To the writer this appears a rather dangerous proceeding in the case of influence of viscosity, temperature and specific gravity of fluids on disk friction. Theories should be proved out by experiments. The work wasted in impeller friction, which is a wasted power due to skin friction, is the most serious loss, and can be obviated by having the impeller revolve in atmospheric air, sealing the edges of outer circumference of impellers at each side of the rim and also at hub; this would go a long way towards increasing overall efficiency and reducing power. Any leakages at these points can be automatically taken care of allowing this water, the amount of which would be very small, to go back to its original source.

Impellers could be made also to operate in air pressure between impeller and casings. Both these methods would reduce the skin friction, gyrostatic and axial losses. Credit is due the author for bringing these subjects before us, and his paper gives us considerable information on these important losses, about which there is very meager and incomplete information.

M. D. Hersey. In order to be able to state that an equation is wrong because it is not dimensionally homogeneous, it is necessary that all the physical quantities governing the fact be included in the equation; hence, while the noteworthy use which the author has made of this criterion, both in refuting Rossiter's equation and in developing his own results, is entirely legitimate, it is to be remembered that in other cases likely to arise in hydraulics if some physical quantity, such as gravity, has been suppressed, an eqnation not dimensionally homogeneous may still be correct.

The Author. In reply to Mr. de Laval, when the surface of an impeller is covered with sediments of the structure of fine sand, the friction loss may increase to three times what it would be with polished surfaces, or twice that prevailing when the disk is a smooth casting. However, such degree of roughness may safely be termed abnormal in turbine pumps, and the loss caused by an ordinary smooth impeller in most pumps is not so much greater than that caused by a polished disk that it pays to polish the impeller faces, especially as they soon lose their polish when running.

As to the influence of eddying and lamellar motion, it should be emphasized that unless the speeds of the pumped fluid are high enough to insure eddying motion throughout, the ordinary centrifugal pump is not fit to pump. The papers presented by Messrs. Buckingham and Hersey at this meeting deal more closely with the problems touched on by Mr. de Laval.

The suggestion to entrust a special commission with experimental work on viscosity is an excellent one. The author intends to direct the attention of the U. S. Bureau of Standards to the advisability of establishing more firmly the standards and laws of viscosity.

Mr. de Laval very aptly draws attention to the effect which the heat developed by impeller friction must have on viscosity. The author, however, has not committed the mistake of resorting to theoretical rather than experimental solution of the question of the influence of viscosity, temperature, and specific gravity of the fluid. As a matter of fact, he resorted to dimensional reasoning in this instance only after having very carefully investigated all available experimental evidence.

After the paper was written, the author applied formula [6] to some other tests carried out with syrup of known viscosity, gravity and temperature and found that the calculation tallied very well indeed with experiment.

Unfortunately the figures quoted by Mr. de Laval are not conclusive, as he does not state the value of the coefficient of viscosity of the oil pumped. This, by the way, is a figure which pumpmakers are rarely able to obtain from customers, but without which they are entirely unable to predict results.

The idea of rotating the impeller in air instead of water is not new. The author has himself given it a considerable amount of thought which he condensed in his application for German patent, of 1909, No. N. 11099. There also are some French and American patents covering similar efforts. The practical difficulties which stand in the way of this idea, however, are very great indeed. How, for instance, can the impeller be prevented from sucking air from the lateral chambers?

At present those high lift impeller designs which tend to diminish the diameter of the impeller, i.e. to diminish the impeller coefficient, deserve more attention. The reduction in disk friction loss is about five times the percentic reduction of this coefficient.

No. 1479

THE RELATION BETWEEN PRODUCTION AND COSTS

By H. L. Gantt, New York

Member of the Society

Manufacturers in general recognize the vital importance of a knowledge of the cost of their product, yet but few of them have a cost system on which they are willing to rely under all conditions.

2 While it is possible to get quite accurately the amount of material and labor used directly in the production of an article, and several systems have been devised which accomplish this result, there does not yet seem to have been devised any system of distributing that portion of the expense known variously as indirect expense, burden or overhead, in such a manner as to make us have any real confidence that it has been done properly.

3 There are in common use several methods of distributing this expense. One is to distribute the total indirect expense, including interest, taxes, insurance, etc., according to the direct labor. Another is to distribute a portion of this expense according to direct labor, and a portion according to machine hours. Other methods distribute a certain amount of this expense on the material used, etc. Most of these methods contemplate the distribution of *all* of the indirect expense of the manufacturing plant, however much it may be, on the output produced, no matter how small it is.

4 If the factory is running at its full, or normal, capacity, this item of indirect expense per unit of product is usually small. If the factory is running at only a fraction of its capacity, say one-half, and turning out only one-half of its normal product, there is but little change in the total amount of this indirect expense, all of which must now be distributed over half as much product as previously, each unit of product thereby being obliged to bear approximately twice as much expense as previously.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

5 When times are good, and there is plenty of business, this method of accounting indicates that our costs are low; but when times become bad and business is slack, it indicates high costs due to the increased proportion of burden each unit has to bear. During good times, when there is a demand for all the product we can make, it is usually sold at a high price and the element of cost is not such an important factor. When business is dull, however, we cannot get such a high price for our product, and the question of how low a price we can afford to sell the product at is of vital importance. Our cost systems, as generally operated at present, show under such conditions that our costs are high and, if business is very bad, they usually show us a cost far greater than the amount we can get for the goods. In other words, our present systems of cost accounting go to pieces when they are most needed. This being the case, many of us have felt for a long time that there was something radically wrong with the present theories on the subject.

6 As an illustration, I may cite a case which recently came to my attention. A man found that his cost on a certain article was 30 cents. When he found that he could buy it for 26 cents, he gave orders to stop manufacturing and to buy it, saying he did not understand how his competitor could sell at that price. He seemed to realize that there was a flaw somewhere, but he could not locate it. I then asked him what his expense consisted of. His reply was labor 10 cents, material 8 cents, and overhead 12 cents. My next question was: Are you running your factory at full capacity? and got the reply that he was running it at less than half its capacity, possibly at one-third. The next question was: What would be the overhead on this article if your factory were running full? The reply was that it would be about 5 cents; hence the cost would be only 23 cents.

7 The possibility that his competitor was running his factory full suggested itself at once as an explanation.

8 The next question that suggested itself was how the 12 cents overhead, which was charged to this article, would be paid if the article was bought. The obvious answer was that it would have to be distributed over the product still being made, and would thereby increase its cost. In such a case it would probably be found that some other article was costing more than it could be bought for; and, if the same policy were pursued, the second article should be bought, which would cause the remaining product to bear a still higher expense rate.

9 If this policy were carried to its logical conclusion, the manufacturer would be buying everything before long, and be obliged to give up manufacturing entirely.

10 The illustration which I have cited is not an isolated case, but is representative of the problems before a large class of manufacturers, who believe that *all of the expense, however large, must be carried by the output produced, however small.*

11 This theory of expense distribution is quite widespread, and clearly indicates a policy, which in dull times would, if followed logically, put many of our manufacturers out of business. In 1897 the plant of which I was superintendent was put out of business by just this kind of logic. It never started up again.

12 Fortunately for the country, American people as a whole will finally discard theories which conflict with common sense; and, when their cost figures indicate an absurd conclusion, most of them will repudiate the figures. A cost system, however, which fails us when we need it most, is of but little value and it is imperative for us to devise a theory of costs that will not fail us.

13 Most of the cost systems in use, and the theories on which they are based, have been devised by accountants for the benefit of financiers, whose aim has been to criticize the factory and to make it responsible for all the shortcomings of the business. In this they have succeeded admirably, largely because *the methods used are not so devised as to enable the superintendent to present his side of the case.*

14 Our theory of cost keeping is that *one of its prime functions is to enable the superintendent to know whether or not, he is doing the work he is responsible for as economically as possible,* which function is ignored in the majority of the cost systems now in general use. Many accountants, who make an attempt to show it, are so long in getting their figures in shape that they are practically worthless for the purpose intended, the possibility of using them having past.

15 In order to get a correct view of the subject we must look at the matter from a different and broader standpoint. The following illustration seems to put the subject in its true light:

16 Let us suppose that a manufacturer owns three identical plants, of an economical operating size, manufacturing the same article,—one located in Albany, one in Buffalo and one in Chicago,—and that they are all running at their normal capacity and managed equally well. The amount of indirect expense per unit of product

would be substantially the same in each of these factories, as would be the total cost. Now suppose that business suddenly falls off to one-third of its previous amount and that the manufacturer shuts down the plants in Albany and Buffalo, and continues to run the one in Chicago exactly as it has been run before. The product from the Chicago plant would have the same cost that it previously had, but the expense of carrying two idle factories might be so great as to take all the profits out of the business; in other words, the profit made from the Chicago plant might be offset entirely by the loss made by the Albany and Buffalo plants.

17 If these plants, instead of being in different cities, were located in the same city, a similar condition might also exist in which the expense of the two idle plants would be such a drain on the business that they would offset the profit made in the going plant.

18 Instead of considering these three factories to be in different parts of one city, they might be considered as being within the same yard, which would not change the conditions. Finally, we might consider that the walls between these factories were taken down and that the three factories were turned into one plant, the output of which had been reduced to one-third of its normal volume. Arguing as before it would be proper to charge to this product only one-third of the indirect expense charged when the factory was running full.

19 If the above argument is correct, we may state the following general principle: THE INDIRECT EXPENSE CHARGEABLE TO THE OUTPUT OF A FACTORY SHOULD BEAR THE SAME RATIO TO THE INDIRECT EXPENSE NECESSARY TO RUN THE FACTORY AT NORMAL CAPACITY, AS THE OUTPUT IN QUESTION BEARS TO THE NORMAL OUTPUT OF THE FACTORY.

20 This theory of expense distribution, which was forced upon us by the abrupt change in conditions brought on by the war, explains many things which were inexplicable under the older theory, and gives the manufacturer uniform costs as long as the methods of manufacture do not change.

21 Under this method of distributing expense there will be a certain amount of undistributed expense remaining whenever the factory runs below its normal capacity. A careful consideration of this item will show that it is not chargeable to the product made, but is a business expense incurred on account of our maintaining a certain portion of the factory idle, and chargeable to profit and loss. Many

manufacturers have made money in a small plant, then built a large plant and lost money for years afterwards, without quite understanding how it happened. This method of figuring gives a clear explanation of that fact and warns us to do *everything possible to increase the efficiency of the plant we have, rather than to increase its size.*

22 This theory seems to give a satisfactory answer to all the questions of cost that I have been able to apply it to, and during the past few months I have laid it before a great many capable business men and accountants. Some admitted that this viewpoint would produce a very radical change in their business policy, and are already preparing to carry out the new policy.

23 It explains clearly why some of our large combinations of manufacturing plants have not been as successful as was anticipated, and why the small, but newer plant, is able to compete successfully and make money, while the combinations are only just holding their own.

24 The idea so prevalent a few years ago, that in the industrial world money is the most powerful factor, and that if we only had enough money, nothing else would matter very much, is beginning to lose its force, for it is becoming clear that *the size of a business is not so important as the policy by which it is directed.* If we base our policy on the idea that the cost of an article can only legitimately include the expense necessarily incurred either directly or indirectly in producing it, we shall find that our costs are much lower than we thought, and that we can do many things which under the old method of figuring appeared suicidal.

25 The view of costs so largely held, namely, that *the product of a factory, however small, must bear the total expense, however large,* is responsible for much of the confusion about costs and hence leads to unsound business policies.

26 If we accept the view that the article produced shall bear only that portion of the indirect expense needed to produce it, our costs will not only become lower, but relatively far more constant, for the most variable factor in the cost of an article under the usual system of accounting has been the "overhead," which has varied almost inversely as the amount of the product. This item becomes substantially constant if the "overhead" is figured on the normal capacity of the plant.

27 Of course a method of accounting does not diminish the ex-

pense, but it may show us where the expense properly belongs, and give us a more correct understanding of our business.

28 In our illustration of the three factories, the cost in the Chicago factory remained constant, but the expense of supporting the Buffalo and Albany factories in idleness was a charge against the business, and properly chargeable to profit and loss.

29 If we had loaded this expense on the product of the Chicago factory, the cost of the product would probably have been so great as to have prevented our selling it, and the total loss would have been greater still.

30 When the factories are distinctly separate, few people make such a mistake, but where a single factory is three times as large as is needed for the output, the error is frequently made, with results that are just as misleading.

31 ***As a matter of fact it seems that the attempt to make a product bear the expense of plant not needed for its production is one of the most serious defects in our industrial system today, and farther reaching than the differences between employers and employees.***

32 The problem that faces us is then first to find just what plant, or part of a plant, is needed to produce a given output, and to determine the "overhead" expense on operating that plant or portion of a plant. This is primarily the work of the manufacturer, or engineer, and only secondarily that of the accountant, who must, as far as costs are concerned, be the servant of the superintendent.

33 In the past, in almost all cost systems the amount of "overhead" to be charged to the product, when it did not include *all* the "overhead," was more or less a matter of judgment. According to the theory now presented, it is not a matter of judgment, but can be determined with an accuracy depending upon the knowledge the manufacturer has of the business.

34 Following this line of thought it should be possible for a manufacturer to calculate just what plant and equipment he ought to have, and what the staff of officers and workmen should be to turn out a given profit.

35 If this can be correctly done, the exact cost of a product can be predicted. Such a problem cannot be solved by a cost accountant of the usual type, but is primarily a problem for an engineer, whose knowledge of materials and processes is essential for its solution.

36 Having made an attempt to solve a problem of this type, one of the most important functions we need a cost system to perform, is

to keep the superintendent continually advised as to how nearly he is realizing the ideal set, and to point out where the shortcomings are

37 Many of us are accustomed to this view point when we are treating individual operations singly, but few have as yet made an attempt to consider that this idea might be applied to a plant as a whole, except when the processes of manufacture are simple and the products few in number. When, however, the processes become numerous or complicated, the necessity for such a check becomes more urgent, and the cost keeper who performs this function becomes an integral part of the manufacturing system, and acts for the superintendent, as an inspector, who keeps him advised at all times of the quality of his own work.

38 This conception of the duties of a cost keeper does not at all interfere with his supplying the financier with the information he needs, but insures that information shall be correct, for the cost keeper is continually making a comparison for the benefit of the superintendent, of what has been done with what should have been done. Costs are valuable only as comparisons, and comparisons are of little value unless we have a standard, which it is the function of the engineer to set.

39 Lack of reliable cost methods has, in the past, been responsible for much of the uncertainty so prevalent in our industrial policies; but with a definite and reliable cost method, which enables us to differentiate between what is lost in manufacturing and what is lost in business, it will usually become easy to define clearly the proper business policy.

In presenting his paper, the author said:

I was moved to present this paper not because the ideas were absolutely new, but because they are of such great importance to manufacturers, and are apparently so little understood by many of them.

Since publishing this paper I have had my attention called to the work of numerous accountants, and especially manufacturers, who, within the last few years, have discarded tradition, and made excellent progress toward a rational system of expense distribution. Some have apparently solved the problem completely. Nevertheless it is a fact that the generally accepted theory of only a few years ago, was that the product of a shop must bear the total expense of owning and operating that shop. It is also a fact that this theory is still misguiding many manufacturers. In addition, the expense of selling was often added as a part of the cost of the article.

The first step taken by students of this problem was to separate the cost of manufacturing from that of selling. According to this division, the expense of manufacturing stops when the article is delivered to the shipping department or placed in the finished stock room. All expenses incurred from this time on belong to the sales department and are deductions from profits.

This separation took one variable and confusing element out of the manufacturing cost; but with a widely varying product and a relative fixed "burden" or "overhead" charge, the manufacturing cost was still subject to fluctuations over which the superintendent had no control, and hence not only gave no measure of the efficiency with which the shop was run, but was no guide at all to the salesman; and was actually misleading when business was dull.

The next step in the evolution of a rational cost system was to establish a fixed "overhead" based on past experience. This had the great advantage of making costs comparable, and of giving the salesman a definite limit by which to be governed.

It had the disadvantage that if the output was at a less rate than the usual previous rate, there was left unabsorbed a portion of this "overhead"; and vice versa, if the product was greater than at the previous average rate, more "overhead" was accounted for than was actually incurred. Of the various methods adopted to take care of this residual "overhead," the two that are best known are, first by charging it to a fund that is eventually charged back on the cost of the product, and second by charging it directly to business as a profit or loss. The second method seems the more logical of the two, and for those who are guided by what has been done in the past, seems to be entirely satisfactory. Indeed for him who is an accountant only, and not familiar with manufacturing methods, it is apparently the only possible solution.

To the engineer, however, who is not so much concerned with *what has been done* as with *what should be done,* it is not at all satisfactory.

If a plant has been built that is larger than is needed to supply the available market, the business error of building the excess portion of that plant should not be charged as a manufacturing cost, but directly to the business as a loss. For instance, if we should build two identical plants where only one was needed, the expense of owning and maintaining one of them in idleness could not be charged to the goods manufactured in the other, but would have to be deducted from the profits of the business.

In the same way the expense of any portion of a plant not needed in production should not be charged to the articles produced, but is a business expense and must be deducted from profits, or entered as a loss, if the profits will not cover it. *In other words, the only expense logically chargeable to a product is that needed for its production when the factory is running at its full or normal capacity,* which may be quite different from that used in its production in the past.

Inasmuch as the determination of this fact is primarily an engineering or manufacturing problem, and not primarily an accounting problem, it becomes evident that cost methods must be based on engineering knowledge, and the cost accountant of the future must himself be an engineer or manufacturer, or be guided by one.

Granting this, it is safe to predict the early dawning of the day longed for by Uncle John Sweet, when *the man who knows what to do and how to do it* shall gradually supplant *the man who knows what was done and who did it.*

DISCUSSION

D. B. Rushmore. In figuring the cost of any product the largest and most indeterminate item is usually the overhead expense, and the proper use of this overhead after it is once obtained is of course the difficult part of the problem. The manufacture and production of power and commodities is usually subject to considerable fluctuations, and in the extreme case of a small power house for a widely fluctuating load, in which the power may vary from zero to a maximum over irregular intervals, the instantaneous cost of power will vary enormously.

Forrest E. Cardullo. I think the idea advocated in the paper is correct, but there are two points to which I would call attention: One is that the loss incurred and charged to profit and loss must be made up in the selling price. The other is whether it is better to charge that loss annually against the product, or to capitalize it and get rid of it once and for all.

W. N. Polakov. The question is whether the cost at which a product has been manufactured in the past is the cost at which it shall and can be manufactured. If the overhead is not properly differentiated from the production cost, and if the production cost is not known,—not as it was in the past but as it ought to be,—we shall not be in a sound position.

For example, there is an electric plant which carried $200,000 overhead a year. The records show that the plant generated current at between 0.72 and 0.68 cent per kw-hr., depending on the load factor. With an annual output of 60,000,000 kw-hr. (about half capacity), the overhead per kw-hr. would be ⅓ cent; with an output of 80,000,000 kw-hr., it would be ¼ cent and with a full output of 120,000,000 kw-hr., allowing a margin for peaks and breakdowns, it would be 0.166 cent.

Consider that the capacity of the plant may not have been sufficient for the increased business, and that a public utility company was willing to sell any amount of current for 0.83 cent per kw-hr. one would immediately go to the old records and get the lowest figure, 0.68 cent, for which current had been manufactured, and then add the minimum overhead of 0.166 cent, giving 0.85 cent, against 0.83 cent which the public service corporation offered. On this basis, buying would seem to be warranted. New records would show, however, that cost of production had risen to 0.91 cent, because output was reduced and overhead was 0.25 instead of 0.166.

We finally come to the point where it appears to be no use operating our plant any longer, so it is shut down and all the power purchased outside at 0.83 cent. If this were done it would mean that we would get the power at 0.83 cent, plus the overhead on the idle plant of $200,000. In this case the cost per kw-hr. would be 0.996 cent, and we would then remember that while the plant was in operation current had been manufactured for 0.85 cent. Where would be the expected saving? One fallacy is that we use past records, instead of the exact scientifically established standard of what the cost ought to be.

In fact, I know from actual experience that in this plant the cost was brought from 0.72 to 0.44 cent, which, even added to the overhead in accordance with the old method, would give only 0.60; consequently the expected saving of some $25,000 on the plan of buying power would be turned into a loss of over $250,000.

William W. Bird. This paper shows in a way the evolution of the mechanical engineer. Twenty or thirty years ago we heard Thurston talk about the steam engine indicator; the mechanical engineer of those days made boiler and engine tests. Ten years ago Taylor told about the stop watch and routing cards for workmen; the mechanical engineer of that period went not only into the power house, but also into the shop.

Now the author tells us it is not good engineering to put all the shop burden on productive labor all the time. In other words, the mechanical engineer of today is applying the general principle of cause and effect in the office of our industrial plants, the same as he has in the other departments.

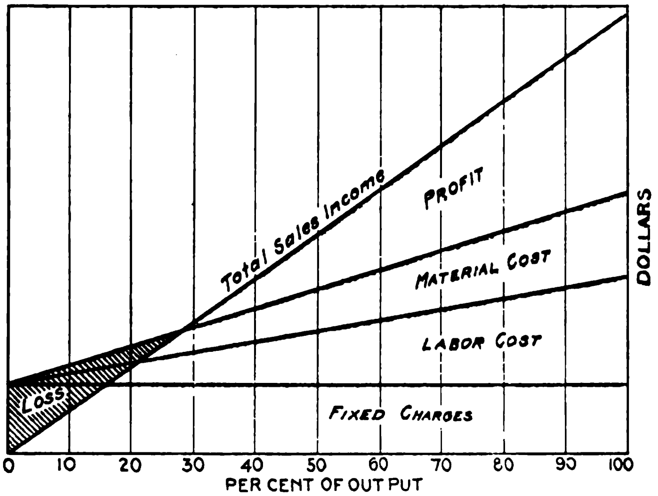


FIG. 1 DIAGRAM SHOWING RELATION OF FIXED CHARGES, PRODUCTION COST AND INCOME

JAMES A. WHITE said he had found the diagram, Fig. 1, useful in determining the effect of fixed charges. The point where the sales income line crosses the total cost line indicates where it is advisable to shut down the factory, the shaded space at the left indicating loss, and the upper space at the right, profit. This diagram is easy of application to a concern whose fixed charges can be accurately determined.

CHARLES PIEZ (written). In our own business we have assumed normal output to be the output of each department secured by its full complement of men working 2500 hr. per year, which represents 90 per cent of the possible working hours per annum on a 54-hr. week basis. The average factory expenses are distributed on a payroll representing this degree of activity, and our average or standard departmental and general expense factors are based on the output which this degree of activity represents. Costs are based on the average factors, and are therefore fairly uniform and wholly independent of the fluctuations of business.

At the end of each month the standard or average factors are compared with the actual factors for each department, and the total expenses compared with the total obtained by distributing the standard rates. In times of depression the standard rates fail to distribute the total expenses, and the debit balance goes to reduce the profits. When the business is running considerably above the assumed normal, the standard factors produce a credit balance which goes to increase the profits.

Sheets showing the comparison between actual and standard expense rates, which the cost department submit to the management each month, keep the management informed and enable it to correct the standard rates when advisable. By following the method of distributing expenses through carefully ascertained standard factors, we get in a ready and practical way the results the author advocates.

Frank H. Neely (written). This paper does much to clear up the uncertainty with which most manufacturers consider indirect expense and its application to costs. Further, it shows in clear relief the urgent need of the engineer in laying out and deciding upon the business issues of manufacturing.

About three years ago, in standardizing the processes in a factory making a complete line of candies and crackers, I found it necessary to develop my cost records very much along the line of the author's paper. This business is seasonal, and in the three fall months the volume is practically double what it is in any other similar period in the year. This, of course, made the costs appear high in the first part of the year and extremely low in the busy season, which forced the standardization of costs on the lines described in the paper for the individual product; and each department, as the year progresses, shows its profit or loss, at the same time absorbing its just pro rata of indirect expense, regardless of the volume manufactured.

Sterling H. Bunnell (written). The reason why cost systems are generally of so little use to the manufacturer is that they are planned for no other purpose than to meet the needs of the accountant in balancing his books. The fact that cost keepers are accountants and not manufacturers is the only explanation of the belief so prevalent that the cost of the product of a given month is equal to the total expenditure of the month.

Cost records should show the management that the cost of product is within proper standards. For this purpose, the items of material and direct labor are sufficient, and the burden figure of secondary importance. But the cost figures should also show the sales department the minimum selling prices which will cover the total of operating cost. Material, direct labor and factory burden furnish only part of the total which selling price must cover in order to insure continued profit.

The cost of today's product is the result of the whole past existence and future purpose of the factory organization. Experience is expensive before it becomes a direct source of profit; equipment, working force, financial resources and goodwill all enter into the cost, not only of the factory product of today, but also of the product of future days and years. The cost of the day's product, therefore, includes more than the chance portion of the total cost of operating the plant that happens to get into the day's accounts.

Keppele Hall gave an example to illustrate the author's contention that overhead or indirect charges cannot be arbitrarily distributed over product without regard to conditions, and still have costs serve as a useful guide to the management or superintendent.

He stated that an objection which might be raised to the author's proposed solution of overhead expense distribution is that it does not hold the superintendent responsible for such items of the indirect expense as he has under his control during slack times. In such cases is it not advisable to divide the indirect expense, hold the product responsible for such portion as the superintendent can control, and relieve it of the portion over which he has no control?

A good arrangement for proportioning indirect costs is to have a machine or work place rate fixed so as to cover all indirect expense under normal conditions. Each job is then charged with the direct labor and material and the machine or work place hours. The balance of the indirect expense, which is not absorbed by the machine and work place rates, is placed in an account known as an under-absorption account. This account increases in dull times and decreases in very busy times. The net result at the end of a given period shows the under or over-absorption of the indirect expense. The cost of the product is estimated from the sum of the three items (labor, material and machine or work place hours), and, except for variations in the efficiency with which the work is done, holds the costs practically constant.

Carl G. Barth (written). The question the author endeavors to answer is only another form of the old question of how low we may take orders in dull times, and it does not seem that the author has reached the bottom of it. The true answer may be had no matter what the policy regarding the distribution of overhead expenses, so long as these are definitely known and properly analyzed.

This is by means of what Mr. Taylor called dull time "limit costs," the making up of which is practised by all concerns which fully understand the true nature of manufacturing costs.

In very dull times it is unfortunately not so much a question of how much money we can make by taking orders, but how to hold our organization together and lose as little money as possible, for the fixed charges go on even if we take no orders at all and allow the organization to disband. In making up a "limit cost," we therefore leave out all consideration of the fixed charges of a plant, and also such other overhead expenses as, without being absolutely fixed, become so for the time being, because we purpose not to disrupt our organization entirely.

The overhead expenses to be added to flat labor and material in making up such a cost are, therefore, such only as will actually be incurred by virtue of undertaking the work under consideration.

If this limit cost is less than the market value of an article, it will then be correct to manufacture the article ourselves rather than to buy it, or offer it in the market for anything, however little, above this limit cost, for this margin will help carry the fixed charges.

Suppose the limit cost of the article in the case cited in the paper could have been shown to figure up to 20 cents only, this would have constituted a still stronger argument against the buying of the article at 26 cents, even if the full and true cost at the time of manufacture was 30 cents.

Ralph E. Flanders (written). In the firm with which I am connected, the plan is followed of setting the overhead rate to agree with average business conditions over a long period, taking into account both good times and bad. An overhead account is carried, to which are charged all the items that go to make up the shop overhead expense; and in like manner to this account are credited all sums apportioned as overhead charges to work in process.

In busy times there will evidently be a deficit in this account. In dull times, on the contrary, the continuance of the heavy expenses, coupled with the small volume of productive labor to which they may be applied, will produce a heavy unapportioned balance. The plan is so to set the average overhead rate that the deficits and excess balances will about cancel each other. From time to time this overhead rate requires adjustment to meet changed business conditions, both internal and external.

It is worth while to compare this plan, which we may call the *average rate* plan, with that set forth by the author, which we may call, for simplicity, the *proportional rate* plan.

In the first place, the average rate, being practically unchanging, is the more easily applied. It has the same advantage as the proportional rate in the matter of avoiding sudden and violent fluctuation in cost figures due to corresponding fluctuation in output.

The average rate offers the most direct method of distributing what may be called the "cataclysmic" overhead expenses, such as taxes, etc., which impose a disastrous load on the period in which they fall, unless apportioned piecemeal over the full term to which they apply.

The main difference between the two plans is that with the average rate the burden of carrying idle equipment and organization through dull times is distributed into the cost of work in good times, while the proportional rate takes it out of the cost system entirely and charges it to profit and loss.

I contend that there is good reason for absorbing this periodically recurring expense in costs, rather than in profit and loss. This charge has not the nature of an extraneous calamity, like an embezzlement or unwise investment. We are forced, unfortunately, to reckon with cycles of boom and depression as one of the conditions of doing business and this condition is therefore a regular factor in the cost of production, and should be so treated. This argument becomes all the stronger when it is remembered that cost figures have a two-fold use. Not only are they employed for comparison with previous costs, but they are used as well to determine whether articles can be profitably manufactured at a given selling price; in some cases, in fact, they are used for setting selling prices. There is nothing like having all unavoidable expenses firmly imbedded in the cost figures, instead of rattling around loose in the ledger.

To sum up the matter, it may be said that the use of the average rate directly disagrees with what the author states as a fallacy, that *all of the expense, however large, must be carried by the output produced, however small.* In fact, it seems to me to be the prime merit of the average rate plan, when based on an overhead account, that no legitimate expense escapes from distribution to costs. The errors which the author sees in this principle are not inherent in the principle at all, but are caused by an illogical application of it. The *average rate* seems to me to answer nearly all his objections.

The proper solution of problems such as outlined by Mr. Gantt, of a factory running below normal capacity, is independent of any particular method of applying overhead charges. You can increase the output without perceptibly increasing the overhead charges, and you may safely reckon the cost of the increased production as equal to labor plus material only. Forget about the overhead. Any margin between the cost and the price at which you can buy or sell may be considered as profit, in the sense that it will, by that much, help to carry your overhead and thus reduce your expenses. This is not a matter of accounting, but of common sense.

D. C. FENNER (written). Mr. Gantt's interesting paper suggests hope for a branch of cost accounting that is still struggling for intelligent analysis and even a semblance of uniformity.

Every alternate step in production, conversion and distribution of any product is that of transportation. Raw stock, stock in process, finished stock must be moved on to the ultimate consumer. At many points motor driven road trucks, shop trucks, crane trucks, tractors and trailers can be used to advantage; but present methods are built around equipment very limited as to capacity and sales value, but strong in associations, sentiment and book valuation.

Depending on how good a horse trader the stable boss may be, the manager figures he can use horses and hand trucks for several years to come. He hesitates to adopt machine equipment on account of its initial cost, and its cost of operation. He has never kept accurate costs of horse delivery and the very limited amount of machine costs that are available are based on conditions that do not fit his business. He finds too that each machine is loaded with a fixed portion of the yearly overhead charges of the installation, whether the machine has been in operation all or a portion of the time. In other words a fixed charge is made against each machine working or idle and at the end of the year the total fixed charges have figured prominently in the "cost of operation per mile" or ton.

Following the author's suggestions, if a motor truck is laid up for lack of work, its fixed charges or overhead should be charged against profit and loss, and at such times the manager should find outside work for his trucks. Further, if a truck is laid up for repairs, the fixed charges for that period of time should be added to the cost of repairs, and should not appear as fixed charges against the actual cost of operation.

By a proper analysis of operating and maintenance costs, a guide can always be found for reducing the idle time and increasing the earning capacity of each machine and the installation as a whole.

C. Bertrand Thompson contributed a written discussion, stating that in periods of depression and subnormal operation, costs should be figured on the basis of the equipment actually used, and the cost of idle equipment should be determined and charged simply in the profit and loss account, to be taken care of in the selling price so far as competitive conditions permit.

Mr. Gantt notes that this method may affect the policy of the plant, but unfortunately does not offer a suggestion as to what the new policy should be. Merely charging the cost of unused plant and equipment to profit and loss does not really solve the problem, which is—How can this loss be made good? When it is a question of closing whole plants, there is a possibility of selling them and thus at least cutting off a part of the loss; but when it is a question of part of a plant being unused it is practically impossible to end the loss by merely disposing of the superfluous part.

Viewed broadly, the condition of subnormal operation in a plant is due to the sales organization rather than the producing organization, not overlooking the fact that the sales organization has a perfectly legitimate excuse for not keeping the plant busy up to its fullest capacity. Special emphasis, nevertheless, should be laid on the fact that it rests on the sales organization to reduce or eliminate the loss.

Here is a field for a further application of scientific management. There is a clear call for the application to the marketing problem of the same type of analysis, scientific research and accurate determination of laws and principles that has characterized the development of scientific management in production.

William Kent (written). Mr. Gantt's paper is an admirable presentation of the evils that result from the adoption of a system of costkeeping, usually advocated by accountants, in which all the indirect expense, burden, or overhead, in a given period of time, such as a month, is charged as part of the cost of the output of that period, even if the amount of that output is, on account of depression of business or other cause, far below normal. The only excuse for such a system is an accountant's, that it enables the cost ledger to be balanced each month.

The author's statement of the general principle or theory of the correct method of charging indirect expense against product is strictly sound and logical, but it is not a new theory or principle. I have been acquainted with it and have believed in it for many years, although I do not recall having seen it in print. I have often made a statement of the principle something like this: "The burden to be charged against any product is the average burden of a normal year for the same quantity of product. If the total cost of keeping a certain machine in a shop for a year, including cost of light, heat, power, repairs, depreciation, rent, etc., divided by the number of hours the machine may be expected to run in a normal year is say 20 cents per machine hour, then the charge for burden to be made against the product of that machine is fixed at 20 cents per hour for the time the machine runs in the following year, whether it runs the normal number of hours or not."

In this connection attention may be called to an example of incorrect reasoning which sometimes follows a strict adherence to distributing burden on the machine hour system. An owner of a machine shop who had a tabulated hourly burden charge for each machine, varying with the size of the machine, the cost of running it and the number of hours that the machine was expected to run in a year, noticed that a small piece was being turned in a very large lathe. He told the foreman that he should not use the lathe for that piece because the burden charge on it was too heavy, and it would make the piece cost too much. The foreman replied that all the other lathes were busy and that there was no heavy work on hand for the large tool, and he thought he would make the big lathe "do something for its keep." The foreman was right, and, moreover, the burden that should be assessed against that piece in making up its cost, if the cost was to be used as a basis for estimating on future orders for similar pieces, is not the machine hour rate of the big lathe, but only that of a small one, on which the work would ordinarily be done.

THE AUTHOR. If I am to draw any conclusions from the discussion of this paper, it has had the effect which I hoped it would have, namely, to make clear that a cost accountant to be really useful to a manufacturing company must understand the manufacturing process.

There is one point, however, which does not seem to have been clearly grasped by some, and that is that what I propose as the real cost of an article is not what it apparently has cost in the past, but what it should cost if the proper manufacturing methods were used and the shop were run at full capacity. This might be called the *ideal cost,* and toward its attainment all efforts should be directed. Mr. Polakov's discussion illustrates this most clearly.

It was perhaps twenty years ago when the great necessity for a knowledge of costs began to be apparent, and manufacturers in general began to give the subject careful consideration. The demand for "cost accountants" soon became so great that almost any clerk who had had experience in a manufacturing plant was able to get a job as cost accountant, much as, today, almost anybody who calls himself an "efficiency engineer," even though he may never have had any engineering experience whatever, seems to be able to gain the confidence of some manufacturer.

Such cost accountants, with a few high-sounding theories and a little bookkeeping experience, but with absolutely no shop knowledge, have too often been able to gain the confidence of the financier, whose policy has been governed by the reports obtained from such sources. The result of such an epidemic of cost accounting has undoubtedly been seriously detrimental to our industries, and it is with a great deal of satisfaction that I see the best accountants of today absolutely repudiating false theories and, if not actually keeping pace with engineers on the subject, at least traveling the same road.

The class of people that advertised themselves as "cost accountants" when "costs" was the watchword, today follow the slogan of "efficiency." This is certainly a step in advance as far as their work is concerned, but before we sacrifice everything on the altar of "efficiency," let us ask whether efficiency is a *means* or an *end,* and get the answer.

It is our duty to ourselves and to society to do well, or efficiently, whatever we do but are we not in danger of losing sight of our object if we lay too much stress on the efficiency with which we strive for it?

It does not take much thought to convince us that *efficiency* is not an *end,* but a *means;* and that it may be beneficial or detrimental as the end is worthy, or unworthy.

To do efficiently something that should not be done at all, benefits nobody. Would it not be better to do something worth while, however inefficiently? Let us stop, therefore, in this wild cry for efficiency long enough to ask what its proper aim is.

If its object is to enable the few to accumulate wealth at the expense of the many, it is not worth while, for an industrial system that allows this will finally fail. If its aim is to enable one man to take unfair advantage of another in any manner it is not suitable to a democratic nation; and it is the country as a whole that must be considered, when we discuss such a broad question as this.

The greatest problem before our industrial world today is the establishment of harmonious coöperative relations between employer and employee. Efficiency is one of the most potent factors in the solution of this great problem, but it can be directed either for or against this solution.

Should we not know on which side it is to be used before we commit ourselves to it?

Before we support too strongly, then, this striving for efficiency, let us be sure that it is to be directly toward a worthy object. Efficiency alone will not cure our troubles, for misdirected efficiency *may be* just as detrimental in the future, as misdirected *cost accounting* has been in the past.

On the other hand, a combination of properly directed efficiency and proper cost methods are absolutely essential to the solution of our industrial problems; and the hopeful thing about the newer ideas of cost keeping is that they point the way of measuring not only the efficiency of the workmen, but that of the manager and of the financier.

Past methods have too often not only failed in this respect, but have frequently been so devised as to relieve the man at the top of the responsibility that was justly his, and to saddle it on the subordinate. The introduction of methods that will relieve this situation will be a long step in the solution of our industrial problems.

No. 1480

THE EFFECT OF RELATIVE HUMIDITY ON AN OAK TANNED LEATHER BELT

BY

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It has long been a recognized fact that the weather has a more or less noticeable effect on leather belts. In experimental work it has often been found impossible to duplicate results when testing the same belt on different days. In practice, those who are familiar with the behavior of leather belts have noticed a difference in the action of belts from day to day, under varying conditions of the weather. However, when the generally accepted rules for belting are consulted, it will be found that the discussion of the weather has been entirely omitted for once. Of the several weather conditions which can be noted readily, it was thought that the variation of the relative humidity of the atmosphere would offer the most promising field and therefore the effect of this variation was chosen as the subject for a special investigation.

2 After several months of preliminary work, with uncertain conclusions and very discouraging prospects, it was decided that, if anything definite was to result from the experiments, the effect of many of the variables would have to be eliminated by keeping most of the conditions constant.

3 The most noticeable effect of an increase in the humidity was found to be in the lengthening of the belt. If the distance

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Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

between the pulley centers remained constant, this lengthening of the belt would decrease the sum of the tensions. On the other hand, the sum of the tensions could be maintained by varying the center distance as the humidity was changed. Thus the field being narrowed to these limited conditions of constant initial length, width, thickness and speed of the belt; diameters of the pulleys;

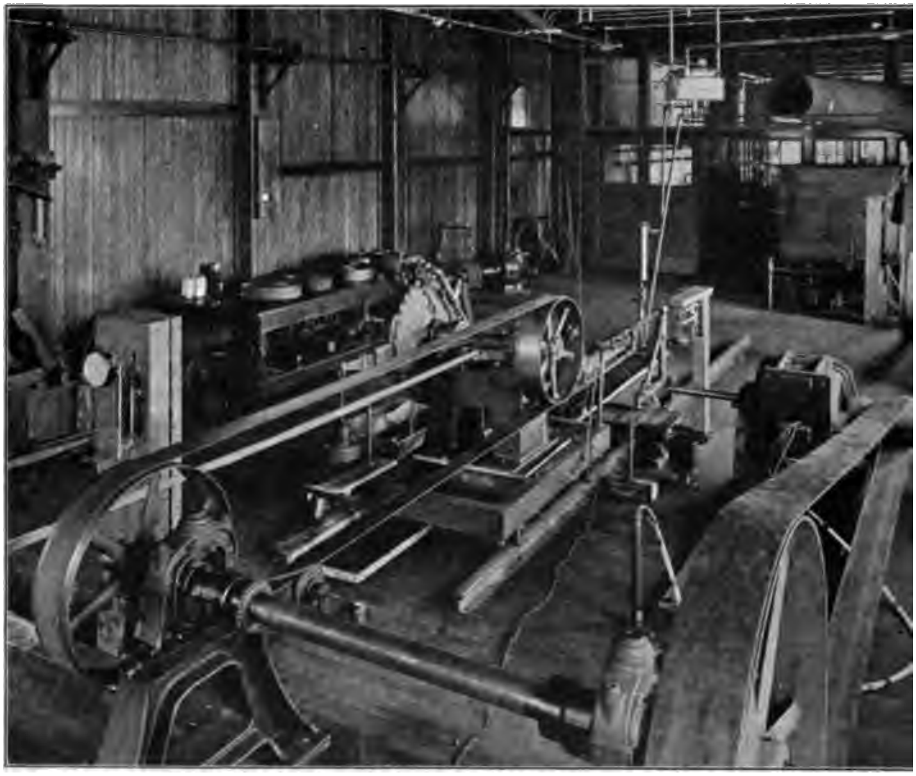

Fig. 1 General Layout of the Apparatus

horse power transmitted, and temperature, the investigation was carried out to determine:

a The effect produced on the center distance by varying the sum of the tensions, the relative humidity remaining constant.

b The effect of the relative humidity on the center distance, the sum of the tensions remaining constant.

c The effect of the relative humidity on the sum of the tensions, the center distance remaining constant.

4 A general layout of the apparatus used in these experiments, which were conducted at the Worcester Polytechnic Institute, is shown in Fig. 1. It may be described as follows:

5 A shaft which carries a pulley on one end and an Alden dynamometer on the other is mounted on a carriage which is free to move in a horizontal direction at right angles to the shaft. On the same level with this first shaft and parallel to it, is a jack shaft driven at constant speed. This jack shaft has a pulley on one end

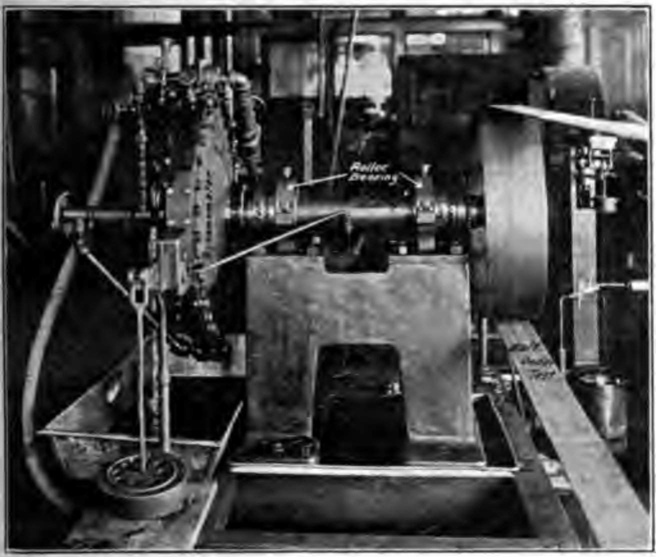

Fig. 2 Detail View of the Dynamometer

the same size as the pulley on the dynamometer shaft and the "belt under test" runs over these two pulleys. The Alden dynamometer furnishes the load which is equivalent to $T_1 - T_2$ or the difference between the tensions of the tight and slack sides and the platform scales weighs the sum of the belt tensions or $T_1 + T_2$, as shown in the figure.

6 In order to measure all of the power transmitted by the belt, the shaft bearings were so designed that they formed a part of the dynamometer. Fig. 2 shows this arrangement in more detail. The bearings consist of a pair of S. K. F. ball bearings in which the shaft

turns. The ball bearings are carried in a housing which is free to turn inside of a pair of Standard roller bearings. Thus what little friction there may be in the ball bearings will tend to turn the housing which is attached to the dynamometer casing, and thereby becomes a part of it.

7 In the rear of the room, Fig. 1, is a Sturtevant heater and blower, the heater to keep the temperature under control and the blower to circulate the air in the room. A live steam jet inside of the heater and a humidifier hung from an overhead beam were used for humidity control. The use of the humidifier had to be discon-

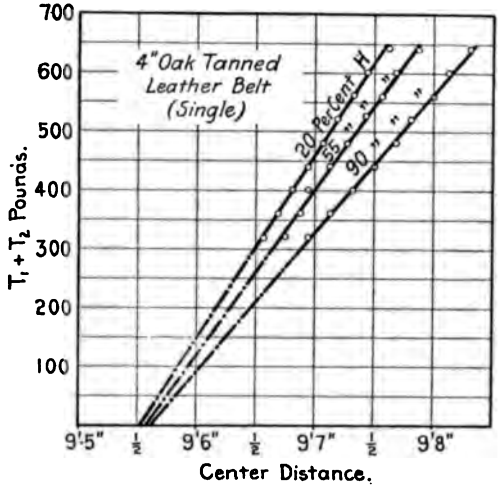

FIG. 3 RELATION BETWEEN $T_1 + T_2$ AND CENTER DISTANCE AT THREE CONDITIONS OF RELATIVE HUMIDITY. HORSE POWER CONSTANT

tinued because it was found to give fog instead of true humidity and consistent results could not be obtained, while with the steam jet it was possible to duplicate results at any time. The degree of relative humidity was measured by a precision hygrometer of the hair type, a wet and dry bulb thermometer and a sling psychrometer, (a modification of the pattern developed by the U. S. Weather Bureau), the last method giving very satisfactory results.

8 The belt used in this investigation was a four-inch, single, oak tanned leather belt furnished by the Graton & Knight Mfg. Co. of Worcester. The pulleys were a pair of cast iron crown face pulleys, twenty-four inches in diameter with a six inch face. The initial length of the belt was such that the center distance at 20 per cent

humidity, T_1+T_2 equalling 320 lb., was 9 feet $6\frac{3}{16}$ in. This makes the belt approximately $25\frac{1}{2}$ ft. long.

9 Standard conditions were assumed to be $T_1=240$ or 60 lb. per in. of width, and $T_1/T_2=3$, where T_1 = the tension in the tight side of the belt and T_2 the tension in the slack side. This gives $T_1-T_2=160$ lb. and as the belt speed remained constant at about 1900 ft. per minute, the horse power was approximately 9.21 all of the time. This condition will give between 0.8 and 0.9 of one per cent slip, or creep.

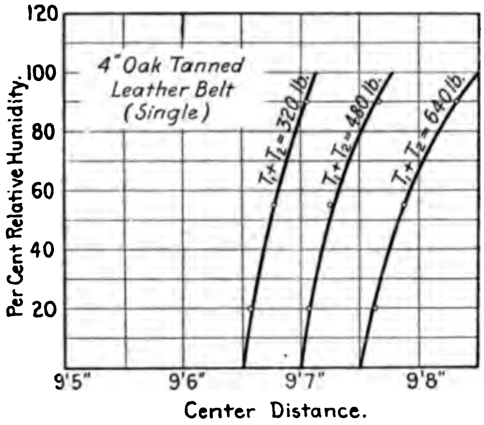

Fig. 4 Relation between Per Cent Relative Humidity and Center Distance at Three Conditions of T_1+T_2. Horse Power Constant

First. Experiments were run to see if a difference in the modulus of elasticity of the belt, when running, could be detected at 20 per cent, 55 per cent and 90 per cent humidity. These relations as found are shown in Fig. 3. Tests were also made at these humidities to see if a difference in the slip due to different values of the modulus of elasticity could be shown. No noticeable effect could be detected.

Second. The results of the tests under the second condition are shown for three different values of T_1+T_2 in Fig. 4.

10 The results of these two sets of experiments were investigated and plotted as a surface as shown in Fig. 5.

11 From Fig. 5 the data for the curves of Fig. 6 were taken. After these curves were drawn, experiments were performed and the circles shown in Fig. 6 indicate the results obtained. These

results, as will be seen were a very good check on the calculations.

12 The three black spots indicated on the surface of Fig. 5 all occur at $T_1+T_2=320$ lb. Now starting at any one of these points and keeping the center distance constant, take the course indicated by the line along which the printing occurs. This line is seen to cross the lines of constant tension, the tension increasing as the relative humidity decreases, or vice versa.

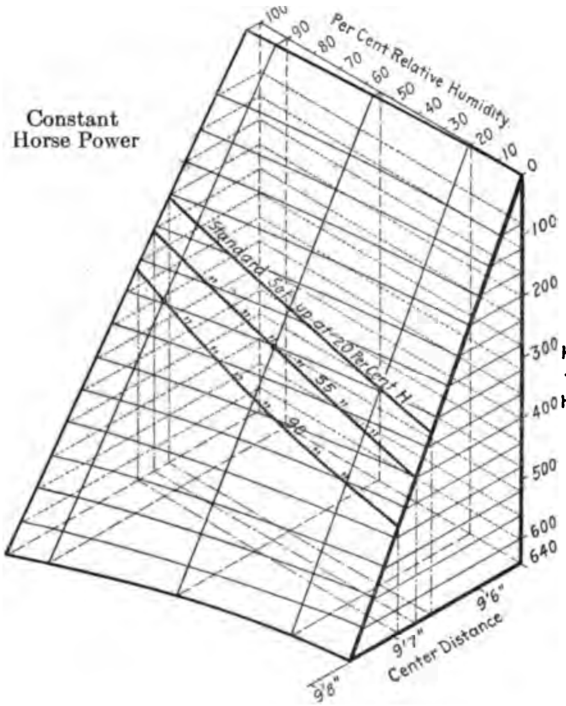

FIG. 5 BELT PERFORMANCE SURFACE FOR AN OAK TANNED LEATHER BELT 4 IN. WIDE, SHOWING RELATION BETWEEN CENTER DISTANCE, SUM OF TENSIONS AND PER CENT RELATIVE HUMIDITY

CONCLUSIONS

13 The surface shown in Fig. 5 might well be called the characteristic of this belt, and it indicates in a general way what might be expected from similar belts. Leather itself will vary; the tanning is different; the quantity and quality of belt dressing is never twice the same. All of the factors being more or less unknown, it will be impossible to make definite prediction regarding other belts.

14 However, in a general way, it may be stated that the effect of a change in relative humidity is greater at high humidities than at low, that the effect is shown more rapidly in a single than in a double belt, and that increasing the humidity shows immediate results while a decrease takes some little time to be effective.

15 If a belt be set up at 20 per cent humidity under standard conditions, then, the load remaining constant, and the center distance being fixed, an increase in the humidity will decrease the sum of the tensions, while the difference of the tensions will remain the same. Therefore, the ratio of the tensions, T_1/T_2, will increase

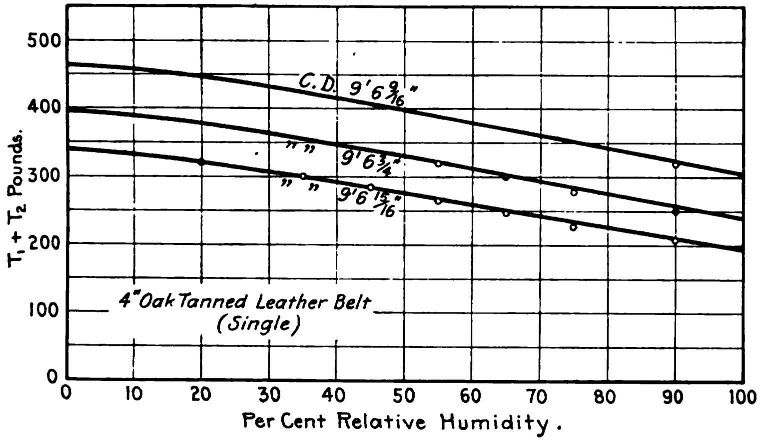

FIG. 6 RELATION BETWEEN $T_1 + T_2$ AND PER CENT RELATIVE HUMIDITY FOR THREE CONDITIONS OF CENTER DISTANCE. HORSE POWER CONSTANT

with the humidity, and at about 90 per cent humidity the belt will be on the point of slipping. This variation is shown in Fig. 7. The data for the curve are taken from Fig. 5.

16 If the belt be set up at 90 per cent relative humidity under standard conditions, a decrease in humidity will increase the sum of the tensions, provided the distance between centers remains constant. This will result in an excessive pressure on the bearings at low humidity and also tend to stretch the belt beyond the elastic limit. This is shown in Fig. 8 which is in reality a section of the surface in Fig. 5.

17 If the standard set up be made at 55 per cent relative humidity, which is somewhere near the normal, any possible increase in relative humidity will not produce an excessive ratio of tensions,

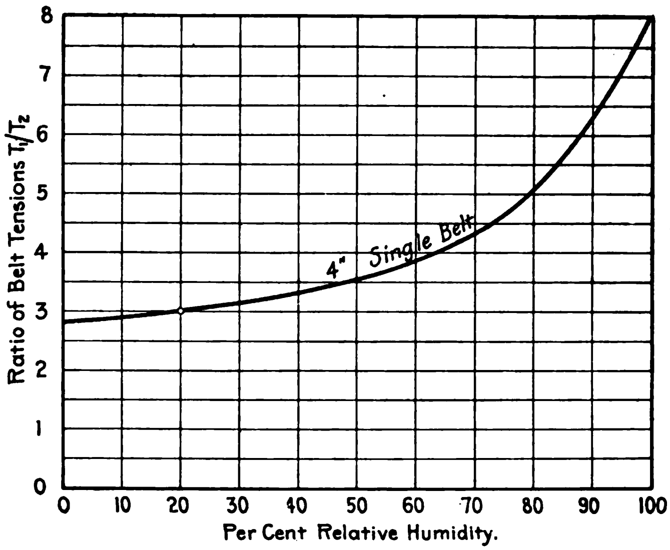

FIG. 7 EFFECT OF CHANGE OF HUMIDITY FROM 20 PER CENT. HORSE POWER CONSTANT

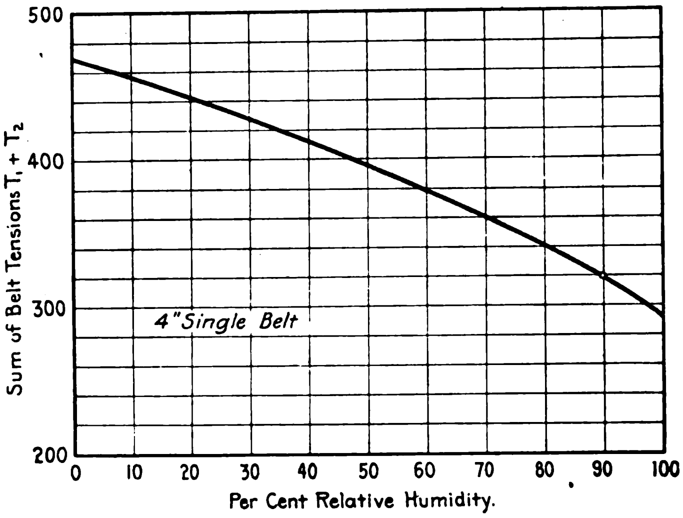

FIG. 8 EFFECT OF CHANGE OF HUMIDITY FROM 90 PER CENT. HORSE POWER CONSTANT

and any possible decrease in relative humidity will not cause an undue sum of tensions. Both of these relations are shown in Fig. 9.

18 As the higher relative humidities generally occur at temperatures above 70 deg., it was thought best to extend the field of investigation by varying the temperature. Accordingly a series of experiments was run at 50 deg. temperature with the relative humidity varying from 20 per cent to 90 per cent and another series at 90 deg. temperature, T_1+T_2 and h.p. being constant for all of these tests.

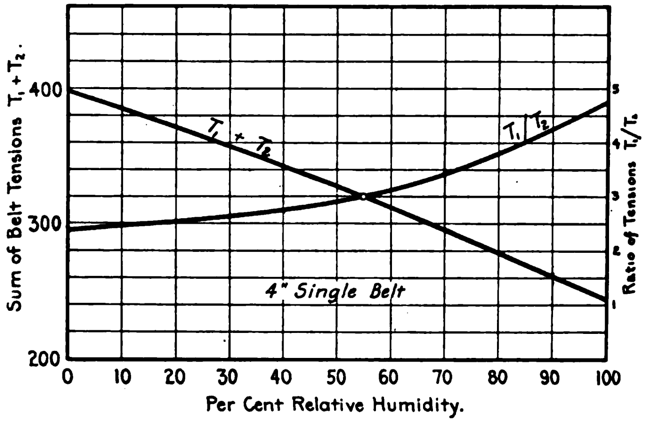

FIG. 9 EFFECT OF CHANGE OF HUMIDITY FROM 55 PER CENT. HORSE POWER CONSTANT

19 These results are shown in Fig. 10, which also has the corresponding results at 70 deg. from previous experiments. This would indicate that the belt lengthens as the temperature increases, the relative humidity remaining constant; that the amount of this lengthening of the belt is somewhat greater at high relative humidities than at low relative humidities; and that the lengthening of the belt due to an increase in the relative humidity is greater at temperatures higher than 70 deg. and less for temperatures under 70 deg.

20 It would appear from our experiments that the lengthening of the belt which takes place when the humidity increases is very nearly proportional to the relative humidity while no definite relation to absolute humidity exists.

21 The fact that the lines in Fig. 10 are not parallel would

indicate that either there is some slight effect due to changes in the absolute humidity or, what is more probable, that the coefficient of expansion is greater at 90 deg. than at 20 deg. temperature.

22 The general conclusions are:

First. If a belt be set up at low relative humidity, slipping will probably occur if the relative humidity increases to any great extent, especially if the increase be accompanied by a rise in temperature.

Second. If a belt be set up at high relative humidity, excessive pressure on the bearings and stretching of the belt will result from a decided decrease in relative humidity, especially if the decrease be accompanied by a fall in temperature.

Third. If a belt be set up at a medium relative humidity, the tensions will not be excessive at lower relative humidities, nor will

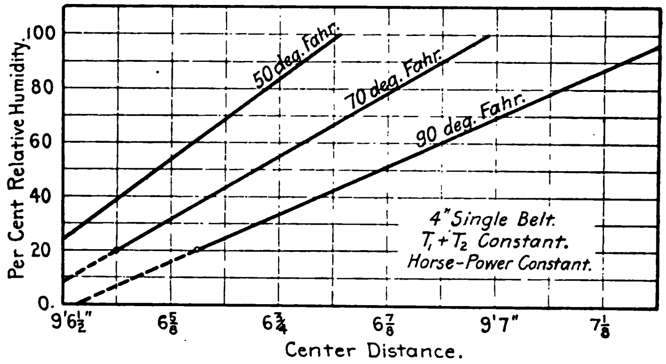

FIG. 10 RELATION BETWEEN CENTER DISTANCE AND RELATIVE HUMIDITY AT THREE DIFFERENT TEMPERATURES

there be any great danger of slipping at high relative humidities unless accompanied by excessive temperature changes. In other words, the factor of safety in the ordinary belt rules is sufficient to take care of the effect of changes in the relative humidity if the set up be made at a medium per cent of relative humidity.

Fourth. If a belt be set up at any relative humidity with a spring or gravity tightener, a load 50 per cent greater than the standard can be transmitted at either high or low humidity without danger of stretching the belt, slipping or excessive pressure on the bearings.

DISCUSSION

GEO. N. VAN DERHOEF (written). This paper goes far in explaining some of the peculiar actions of belt drives. It is strange that, in all the previous experimental work on transmission of power by leather belts, the effect of humidity should not have been investigated.

The results again show the importance of using spring belt clamps in tightening belts, and also that when these are used attention should be given to temperature and relative humidity. Those who have not found the use of spring belt clamps all they expected may find the real explanation in the disregard of these two points.

The results also show the advantage of more frequent use of tighteners. Highest efficiency of drive and maximum life of belt can only be secured by keeping the slack side tension as low as possible. The injury to the belt by reverse bending round the tightener pulley is practically negligible if the pulley is made of large diameter; in fact, this loss and that due to the friction of the tightener is far less than that due to the extra load on the main shaft bearings from the belt without a tightener never being at its condition of minimum stress, except just before taking up. A belt tightener is, however, of little value unless it is used to keep the belt just as loose as possible, and it is very unfortunate that this important device was given the name of *tightener* instead of *loosener*.

Nearly all belted electric generators and motors are arranged with sliding bases, and the belts used with them are considerably smaller than would generally be used with equal loads for other machinery.

The great success of the continuous system of rope transmission is due very largely to the fact that the tension can be kept at a minimum by means of the automatic tension carriage; and while it is impossible to secure as favorable results with a belt drive, they can frequently be more or less approximated by the intelligent use of belt tighteners.

CARL G. BARTH wrote that some fifteen years ago, while with the Bethlehem Steel Co., he attempted to study the influence of humidity on the tensions of two belts in the shop, by plotting daily simultaneous humidity readings and readings of belt tension scales applied to the belts. However, due to the unlooked-for extraordinary variations in the loads transmitted by these belts (at times they would carry heavy loads and again they would run idle for days at a time), no definite results were obtained, whereas he could not help believe that results of some value would have been secured if the belts had transmitted a fairly uniform load day and night.

Previously, the drop in tensions of two other belts had been studied during the winter months, when the shop was heated and both the temperature and the humidity thus kept within fairly small limits of variation. The results[1] were very satisfactory when the crudeness of the scales used is considered.

He believed the results given in the paper could be more readily applied if the experiments were repeated along the following slightly different lines:

Take a brand new first class belt and put it under an initial tension of 240 lb. per sq. in. of cross section, over revolving pulleys transmitting no power. Measure its length under this tension while the humidity is kept constant. Keep this up until the tension has fallen to 120 lb. Re-tighten the belt over the same pulleys to 240 lb. by cutting out the necessary fraction of its length, note this length, and proceed as before; repeat this procedure for at least one school year. During the next two school years, repeat the process under different degrees of humidity, and with belts of the same size and make, and preferably cut from the same roll as the first; or, build and equip three separate rooms for the purpose and do all the work in one year. Next, thoroughly impregnate the belts with some good belt dressing, such as Kling Surface or Plome, and repeat the experiments. It is claimed, and it is undoubtedly true, that belt dressings keep out the moisture to a considerable extent.

He was sure that results obtained from a constant length of belt under no load transmission would be more rapidly applicable in practice than would those obtained by a constant load transmission with variable belt length.

F. B. GILBRETH thought we paid too much attention to the cost of the belt; it is to the cost of the up-keep of the belt and its effect on the achievement of the task of the worker that we should look. He would like to know the effects of these experiments if carried on in practice on those two features.

WM. S. ALDRICH (written). The 55 per cent relative humidity chosen by the authors for their standard test comparisons seems reasonable, but the 70 deg. temperature chosen as an accompanying

[1]Transmission of Power by Leather Belting, Barth, Trans. Am. Soc. M. E., vol. 31, p. 43.

standard shop temperature would be uncomfortable in practice. The best working temperature is still a mooted question and depends on the class of workmen, the kind of physical work and the humidity. The best range is probably a little under or over 60 deg., according to local circumstances. 60 or 62 deg. is also about the standard normal temperature for comparisons of engineering data, in English measures.

The barometric pressure must be taken into consideration in standardizing atmospheric conditions with regard to the relative and absolute amount of moisture present. It is not unusual for this to range over 2 in. of mercury in the course of a day in very changeable weather. Were barometric readings taken throughout and all observations reduced to standards of comparison for the conditioned relative humidity selected?

A careful study of the results in Fig. 10 will show that it is probably the actual amount of moisture present in the air which most influences the stretch of the belt, and this is, therefore, the determining factor. In the diagrams, the scale values of the relative humidity may be interpreted as directly proportional to the actual moisture, assuming, however, that the barometric pressure was constant throughout the test.

From Meteorological Tables[1], the absolute amount of moisture present in the air under standard conditions, is:

Deg. fahr..............	50	70	90
Gr. Troy...............	4.076	7.980	14.780

In other words, at the standard condition of the air in the shop of 55 per cent relative humidity, the actual moisture in the air, at the above temperatures will be 2.24, 4.39 and 8.23 gr. respectively. These weights are not quite in geometric progression, but they are sufficiently cumulative to suggest interesting comparisons. They show to what extent the belt can absorb moisture as the temperature rises—how hygroscopic it really is. In short, the belt seems to have almost unlimited capacity to absorb moisture as the temperature rises, and in comparison with the accompanying equal increments of belt stretch under test.

The authors have well pointed out that the difficulties inherent with so many variables as naturally arise in belt testing indicated constant speed and constant load as prerequisites. It would be interesting to know how these latter might vary under varying

[1]Smithsonian Institution, Washington, D. C.

humidity with constant center distance, since it is this latter condition which is imposed on the belt in actual service. For precise work in certain driving operations, it may even be desirable to go to the expense of waterproofing the belt if this should prove feasible.

A. F. Nagle (written). This is a laboratory experiment and as such has an educational value, but its practical value may be questioned. Practical considerations, that is, men and materials, do not admit of too great refinements. Belt tensions should be adjusted by a mechanical engineer, with spring scales to guide him; but the operating mechanic will cut out an inch, more or less, if he finds a belt does not drive his machine. When the works are large enough to employ special men to attend to all belts, something like the refinements alluded to in this paper may be carried out, but even then the practice is liable to fall into disuse.

The authors should give the actual thickness of the belt used. "Single thickness" is not specific enough, for belts in the market under this designation vary nearly two to one in thickness.

W. W. Bird replied that if a belt is fitted up with a spring or gravity tightener, it practically adjusts itself, and a very material difference in up-keep results. He was running a great many machines with an idler or spring arrangement to take up and tighten the belt, and this is done automatically; the arrangement not only lengthens the life of the belt, but also has a bearing on the question of upkeep. A few dollars for a belt is nothing in comparison with the loss of use of a machine.

The authors do not think best to give here the results of their study of the comparative effects of relative and absolute humidity on leather belts. They have proved to their own satisfaction, however, that the statement in Par. 20 is correct.

No. 1481

ON MEASURING GAS WEIGHTS

By Thomas E. Butterfield, South Bethlehem, Pa.

Member of the Society

The Society has spent much effort in helping to establish standard designs for construction details and standards of procedure in testing. The author is interested in accurate methods of determining gas quantities, such as the quantity of gas delivered by a fan, a blower, or a compressor, or the quantity of gas generated by a producer, furnace or other combustion apparatus, or finally the quantity of gas used or consumed for various purposes. The following remarks are intended to raise discussion on points which the author thinks should not be omitted in a new standard codé on procedure in gas measurement, and also in the hope that they may be of some immediate use in the practice of the members generally.

2 In reporting results on gas measurement, the use of *volume* as an expression of quantity or mass should be eliminated Gas quantities should be expressed by *weight*. Volumes of ordinary standard gases even at standard pressure and temperature are useful in determining quantities, but it is almost always misleading to use such volumes as measures of quantity or mass.

3 The various gases used in the industries are not simple chemical compounds, but each one is a mixture of such simple compounds in varying proportions. The name of a gas even to the engineer is no exact indication of its constitution or physical properties, because commercial gases made by the same process or indeed in the same apparatus are subject to important variations in the proportions of their principal constituents.

4 Density of a gas may be readily calculated from the chemical analysis, but the result of such calculation giving quantity in pounds should always be reported by the investigator, whose business it is to present his results in a form permitting the reader most easily to

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

make accurate comparisons with other reports. For fuel gases this would also imply that thermal quality be expressed in heat units per pound, and in general all volume data should be regarded as of only collateral interest.

5 It is usual to consider that after generation commercial gases retain their composition unaltered. Even gases containing no condensible tarry constituents, however, suffer considerable changes in their moisture content with changing temperature, due to the corresponding change of vapor tension of steam. Gases which are permitted to come into contact with water when hot and are then used hot or even warm, contain large quantities of moisture. If a gas be used at the same temperature with different treatment after water contact or with no water contact, the moisture content will be quite different in each case. The ordinary volumetric analysis gives the composition of the dry gas, which is different from the actual composition as generated or used. An arbitrary assumption sometimes made is that the gas is saturated with moisture. This will be correct in but few cases.

6 Where accuracy is of importance the moisture content should be measured. A simple approximate method suggested is to use wet and dry bulb thermometers in the gas main, the amount of water carried being determined from the thermometer readings after reference to a hygrometric table. For great accuracy the hygrometric table should be made for the gas being measured.

7 Gasometer measurements furnish the most accurate method of determining gas volumes and weights. It is essential that the temperature of the gas should be uniform in every part of the gasometer. This requires a uniform temperature of the air surrounding the gasometer, protection from all unequal radiation interchanges, and exact control of the temperature of the entering gas.

8 Displacement gas meters, whether of wet or dry type, are very accurate where the volume of the measuring chamber is unalterable or accurately known at every instant, and where the pressure, temperature and humidity of the gas at the instant of filling are also known, or nearly the same as at calibration. This is not true in fluctuating flow.

9 The pitot tube, venturi meter, and orifice methods of measurement depend for accuracy on the preservation of a constant relation of velocities over an entire cross-section, accurate measure of this cross-section, and accurate measurement of gas density. It is evidently as easy to calculate weights as volumes from the readings

of such meters. They are not at all adapted to measure a rapidly fluctuating flow or a flow accompanied by eddies.

10 Where the specific heat of the gas is known its weight may be calculated by the change in temperature produced by the addition or abstraction of a known quantity of heat. Heat may be supplied by passing a known electric current flowing through a known resistance, or probably with greater accuracy by the flow of water. Either of these methods is cumbersome and requires a large amount of attention.

11 Where large volumes of gas are to be measured reliable shunt methods could be developed for measuring part of the flow, just as electric current is measured. A rational form for such a shunt would be a double walled diaphragm placed in the main through which the gas passes. The diaphragm is pierced full of holes all of the same size, say of about one inch diameter. From one in twenty to one in one hundred of these holes communicates with the interior of the diaphragm; the remainder pass through both walls, and are short, slightly flaring tubes with sharp edges. The gas from the interior of the diaphragm is carefully metered and returned to the main, and gives a measure of the total amount flowing. This method should give quite accurate results with either continuous or fluctuating flow.

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cent, $H_2 = 46$ per cent, and $N_2 = 2$ per cent. After burning with air containing 1½ per cent moisture the volumetric analysis of the dry products of combustion is

$CO_2 = 8.8$ per cent, $O_2 = 4.5$ per cent, and $N_2 = 86.7$ per cent.

It is required to find the ratio by weight or volume of the burnt gas to fuel gas. Call this ratio a_w for weight, or a_v for volume. Then, necessarily, the weight of air is $a_w - 1$, if fuel gas weight be taken as unity.

15 The four elements, carbon, hydrogen, oxygen and nitrogen are the only ones present in the three gases, and the volumetric analyses are reduced to weight analyses giving proportions of these four elements. The actual volumetric analysis of the fuel gas including moisture is

Constituents.......	CO	CO_2	CH_4	C_2H_4	H_2	N_2	H_2O
Per cent..........	5.9	2	39.2	3.9	45	2	2

The total carbon weight is

$$12\ (5.9 + 2 + 39.2 + 2 \times 3.9) = 659.$$

The hydrogen weight is

$$1\ (4 \times 39.2 + 4 \times 3.9 + 2 \times 45 + 2 \times 2) = 268.$$

The oxygen weight is

$$16\ (5.9 + 2 \times 2 + 2) = 190.$$

The nitrogen weight is

$$28 \times 2 = 56.$$

The density of the gas is

$$\frac{659+268+190+56}{200} = \frac{1173}{200} = 5.86 \text{ times that of hydrogen.}$$

The weight analysis by elements is: carbon $= \frac{659}{1173}$, hydrogen $= \frac{268}{1173}$, oxygen $= \frac{190}{1173}$, and nitrogen $= \frac{56}{1173}$.

16 In the equations following the subscript a denotes fuel, b burnt gas, c air, and d dry burnt gas.

17 Then let C_a be pounds of carbon in one pound of fuel. N_a pounds nitrogen in one pound of fuel, etc.

18 The weight analysis of fuel will be $C_a = 56.1$ per cent, $H_a = 22.9$ per cent, $O_a = 16.2$ per cent, and $N_a = 4.8$ per cent.

19 Similarly the weight analysis of air is: $H_c = 0.1$ per cent, $O_c = 24$ per cent, and $N_c = 75.9$ per cent, with a density $= 14.34$ times that of hydrogen.

20 The weight analysis of the "dry" products of combustion is: $C_d = 3.6$ per cent, $O_d = 14.4$ per cent, $N_d = 82$ per cent and the density is 14.8 times that of hydrogen.

21 The weight of any element in the burnt gas (before condensation of moisture) is the sum of the weights in fuel gas and air, giving rise to the four fundamental equations:

$C_a+(a_w-1)\ C_c=a_w\ C_b$, carbon equation . . . . [1]

$H_a+(a_w-1)\ H_c=a_w\ H_b$, hydrogen equation . . . . [2]

$O_a+(a_w-1)\ O_c=a_w\ O_b$, oxygen equation . . . . [3]

$N_a+(a_w-1)\ N_c=a_w\ N_b$, nitrogen equation . . . . [4]

22 Since the weight of carbon in air is negligible

$$C_a=a_wC_b \text{ or } a_w=\frac{C_a}{C_b} \qquad \text{[5]}$$

23 Since the relative proportions of carbon and nitrogen in the burnt gas cannot be altered by the separation of moisture, we may write

$$\frac{\text{carbon in fuel plus carbon in air}}{\text{nitrogen in fuel plus nitrogen in air}}=\frac{\text{carbon in dry burnt gas}}{\text{nitrogen in dry burnt gas}}$$

or

$$\frac{C_a+(a_w-1)\ C_c}{N_a+(a_w-1)\ N_c}=\frac{a_wC_b}{a_wN_b}=\frac{C_d}{N_d}.$$

Simplifying

$$\frac{C_a}{N_a+(a_w-1)\ N_c}=\frac{C_d}{N_d}\text{, and } a_w=1+\frac{C_aN_d-N_aC_d}{N_cC_d} \qquad \text{[6]}$$

24 Equation [6] gives a simple expression for the weight ratio from the analyses by weight of fuel and "dry" burnt gas.

The weight of moisture in the burnt gas is nine times the total weight of hydrogen in fuel and air, or

$$9\{H_a+(a_w-1)\ H_c\}$$

and the weight of "dry" burnt gas per pound of fuel is

$$a_w-9\{H_a+(a_w-1)\ H_c\}.$$

Then

$$\frac{C_b}{C_d}=\frac{N_b}{N_d}=\frac{a_w-9\{H_a+(a_w-1)\ H_c\}}{a_w}$$

and

$$C_a=a_wC_b=[a_w-9\{H_a+(a_w-1)\ H_c\}]C_d \text{ and } a_w=\frac{C_a+9\ (H_a-H_c)\ C_d}{(1-9\ H_c)\ C_d} \qquad \text{[5a]}$$

Similarly for nitrogen

$$N_a+(a_w-1)\ N_c=[a_w-9\{H_a+(a_w-1)\ H_c\}]\ N_d \text{ and}$$

$$a_w=\frac{N_c-N_a-9\ (H_a-H_c)\ N_d}{N_c-N_d\ (1-9\ H_c)} \qquad \text{[4a]}$$

25 The oxygen in air and fuel less the oxygen that separates

from the burnt gas as moisture is equal to the oxygen in the "dry" burnt gas, or

$$O_a + (a_w - 1)\, O_c - 8\{H_a + (a_w - 1)\, H_c\} = [a_w - 9\{H_a + (a_w - 1) H_c\}]\, O_d$$

or

$$a_w = \frac{O_c - O_a + (8 - 9\, O_d)\,(H_a - H_c)}{O_c - 8\, H_c - O_d\,(1 - 9\, H_c)} \quad \text{[3a]}$$

26 Applying equations [3a], [4a], [5a] and [6] to the solution of the problem given we use the recapitulation of analyses by weight.

	Fuel	Air	Dry burnt gas
O	16.2	24	14.4
N	4.8	75.9	82
H	22.9	0.1	....
C	56.1	....	3.6
Density	5.86	14.34	14.8

Then from [3a]

$$a_w = \frac{O_c - O_a + (8 - 9\, O_d)\,(H_a - H_c)}{O_c - 8\, H_c - O_d\,(1 - 9\, H_c)} =$$

$$\frac{0.24 - 0.162 + (8 - 9 \times 0.144)\,(0.229 - 0.001)}{0.24 - 8 \times 0.001 - 0.144\,(1 - 9 \times 0.001)} = 18.05$$

From [4a]

$$a_w = \frac{N_c - N_a - 9\,(H_a - H_c)\, N_d}{N_c - N_d\,(1 - 9\, H_c)} =$$

$$\frac{0.759 - 0.048 - 9\,(0.229 - 0.001)\, 0.82}{0.759 - 0.82\,(1 - 9 \times 0.001)} = 18.13$$

From [5a]

$$a_w = \frac{C_a + 9\,(H_a - H_c)\, C_d}{(1 - 9\, H_c)\, C_d} = \frac{0.561 + 9\,(0.229 - 0.001)\, 0.036}{(1 - 9 \times 0.001)\, 0.036} = 17.79$$

From [6]

$$a_w = \frac{C_a N_d - N_a C_d}{N_a C_d} = 1 + \frac{0.561 \times 0.82 - 0.048 \times 0.036}{0.759 \times 0.036} = 17.79$$

27 The percentage error $\frac{18.13 - 17.79}{18.13} \times 100 = 1.9$. The error evidently lies in the too rough approximation in determining $C_d = 0.036$. The value of the check given by the separate determinations is evident. The author has had this method in use for some years with very satisfactory results.

DISCUSSION

SANFORD A. MOSS (written). It is greatly to be doubted if the universal custom of expressing gas quantities by volumes can ever be superseded. The writer therefore recommends the following as being the nearest practical attainment of the author's idea.

Amounts of air or gas delivered by fans, blowers or compressors should be given as cubic feet, referred to a fixed pressure and temperature. I have used 14.7 lb. per sq. in. abs. pressure (at sea level and 45 deg. north latitude, if such precision is needed) and 60 deg. fahr. temperature, and have called the quantities so given "cu. ft. of standard air," or "standard gas." 70 deg. fahr. and 62 deg. fahr. have also been used. A cubic foot of standard air or standard gas is a unit of weight, just as the author suggests, but does not involve a serious departure from customary practice.

Another definite unit, which is not a unit of weight, however, is a cubic foot at atmospheric conditions. The weight of such a unit varies with barometer, altitude and atmospheric temperature. The unit most interesting to a designer of a fan or compressor is the cubic foot at the average atmospheric conditions of the point of installation, and this is usually understood when cubic feet of a fan is specified without qualification. The term "cu. ft. of free air" or "free gas" is often used without stating whether cubic feet of standard air or cubic feet of air at average atmospheric conditions is meant. Hence, the use of this term without definition should be avoided.

ARTHUR WEST (written). I agree with the author in his contention that gas measurements should be made by weight rather than by volume. I believe this custom would greatly reduce the possibilities of error in matters involving the measurement of gases of various kinds.

THE AUTHOR is convinced that the compound unit for measuring gas mass can be safely used only for laboratory work. He believes that it will be displaced by the more rational and simple unit,—weight.

In the codes for Tests of Power Plant Apparatus, Par. 184a, the long equation and the paragraph following give an incorrect method for calculating pounds of gas per pound of fuel. In effect it is stated that this ratio equals the ratio of the percentage of carbon in the fuel divided by the percentage of carbon in the gas. Another term must be subtracted, equal to the product of the weight of ash per pound of fuel by the ratio of the percentage of carbon in the ash to the percentage of carbon in the gas. The omission of this term was, of course, accidental, but would be practically impossible if the author's method were used.

In addition the method in the paper provides three independent calculations to check. It is applicable to any furnace reaction. In brief, it substitutes a simple, general method in place of the casual and partial methods now in use.

No. 1482

A STUDY OF AN AXLE SHAFT FOR A MOTOR TRUCK

HOW IT WAS STRENGTHENED BY HEAT TREATMENT

By John Younger, Buffalo, N. Y.

Member of the Society

What is described herein is an investigation which the writer made to determine the cause of failure of a very important detail in large motor trucks. The remedy adopted is also described.

2 A detailed investigation is valuable in future designs of similar structures. What follows has a very much wider application than to motor trucks, and is testimony to the value of heat-treated steels.

3 The design of the shaft and its adjacent parts is shown in the diagrammatic sketch, Fig. 1. The flutes or splines at the ends are slightly loose so that it is under no constraint except to move in a rotary path. In other words it is intended to be subject to pure torsional stresses, with no complicating bending effects.

4 The shaft itself is shown in Fig. 2. It was made from 2¼-in. diameter chrome nickel bar, turned all over to 2.1235 in. diameter. The specifications of the steel read as follows:

Chemical	Per cent	Physical	
			Lb. per sq. in.
Carbon about	.20	Elastic limit	90,000
Chromium about	1.5	Maximum strength	105,000
Manganese about	.30		
Nickel about	4.00		Per cent
Silicon about	.20	Reduction in area	66
Phophorus and sulphur below	.04	Elongation	25

The shafts broke in service as shown by the illustrations, Figs. 3, 4, 5, and 6.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

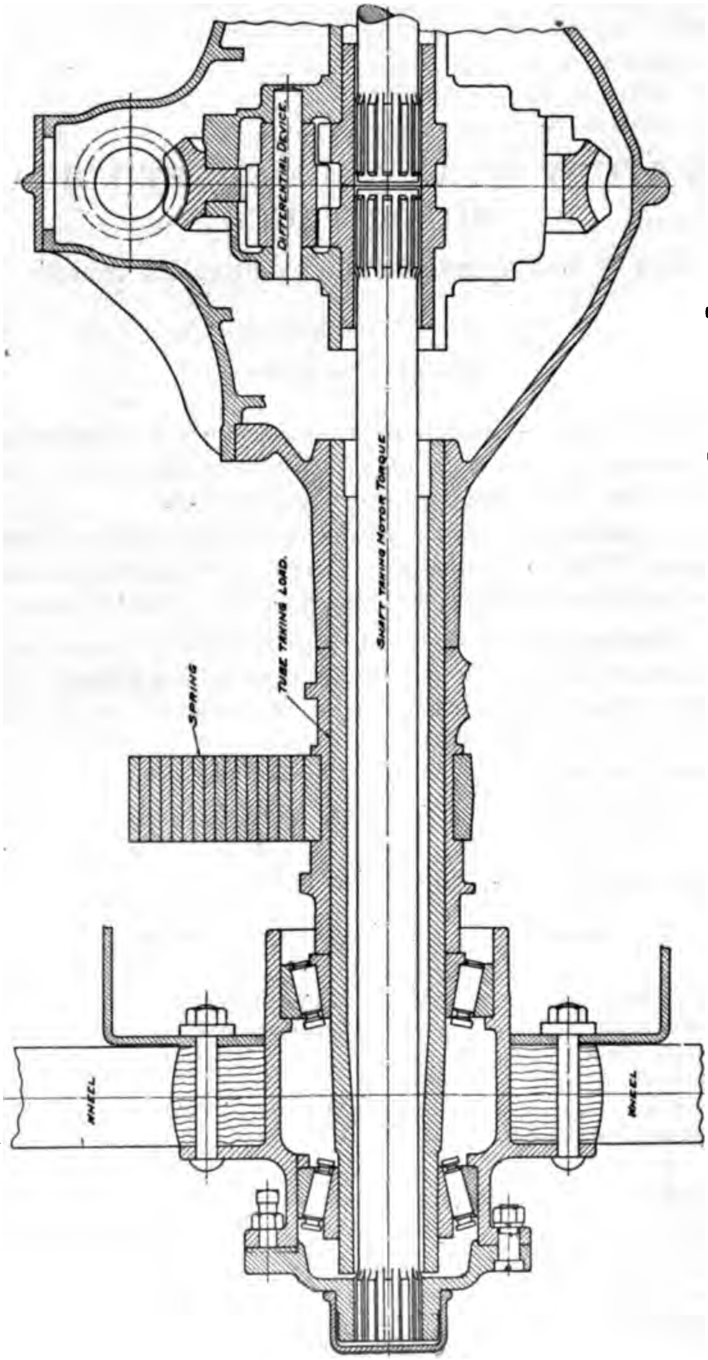

FIG. 1 DIAGRAMMATIC DESIGN OF TRUCK REAR AXLE SHOWING LOCATION OF SHAFT

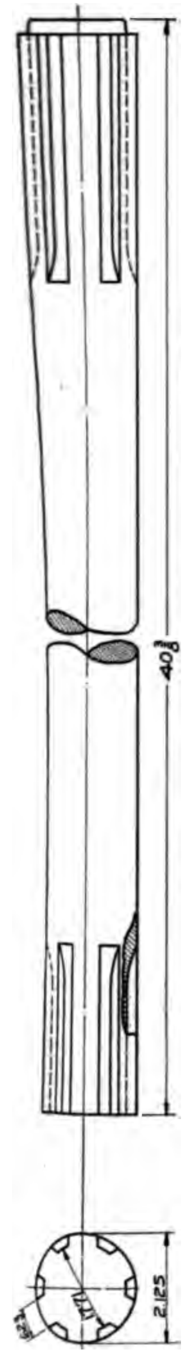

Fig. 2 The Original Shaft

Fig. 3 Characteristic Fracture Shaft turned Down

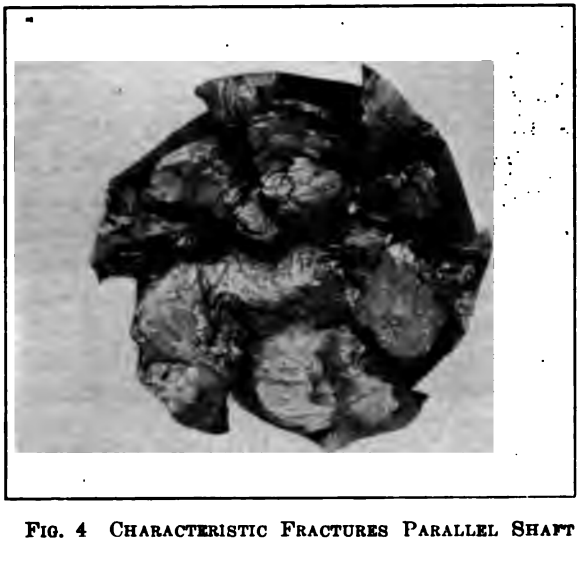

Fig. 4 Characteristic Fractures Parallel Shaft

CALCULATIONS OF STRENGTH

5 The following are the particulars of the motor truck upon which all calculations must be based:

Motor develops 44 h.p. at 1000 r.p.m. under average conditions.
Transmission reduction is 3.77 to 1.
Worm gear reduction is 9.75 to 1.
Total reduction, 36.8 to 1.

6 From various tests into which it is needless to enter in this

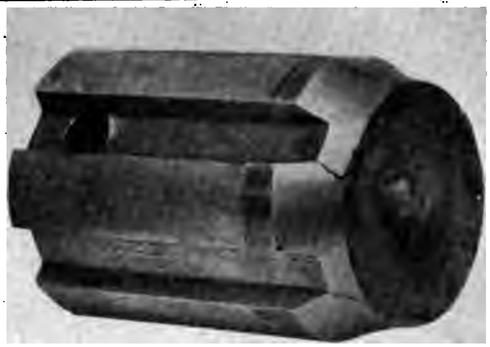

FIG. 5 SHOWING EFFECT OF SPLINE

paper, the efficiency of the transmitting mechanism between motor and axle shafts on this gear ratio is about 75 per cent. There are therefore 33 h.p. transmitted as a maximum by the shafts at 27 r.p.m.

7 In common with usual road vehicle practice, a differential gear is fitted, so that the power transmitted by each shaft at each wheel is only half the total. The shaft shown in Fig. 2, therefore, is under maximum pure torsional stress due to the transmission of 16½ h.p. at 27 r.p.m.

EFFECTIVE DIAMETER OF SHAFT AND EFFECT OF SPLINES

8 The next question that arose was as to the effective diameter on which to base calculations of stress. The large diameter of 2⅛ in. would not be accurate inasmuch as the flutes at the ends weaken the shaft to a considerable extent. That they do weaken the shaft is

obvious from Figs. 3, 4, 5, and 6, where the lines of cleavage are clearly seen to start from the corner of the spline. While it is true that a sharp corner is a very dangerous source of weakness, it was impossible to get a radius of more than 0.02 in. at the fillet without reducing the area of bearing surface. Key seats or splines are always weakening elements, but it is unfortunately impossible to do without them.

9 Mr. C. E. Larard, member of the Institution of Mechanical Engineers, (London), read a paper in 1911 before the Institution of Automobile Engineers on "Strength of Castellated or Splined Shafts." The results of his investigations are shown in Table 1.

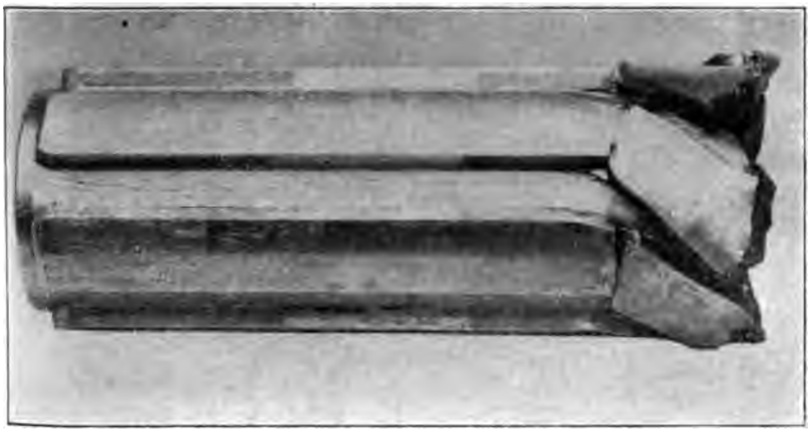

Fig. 6 Characteristic Fracture

10 The diameters of the plain shafts were the same as the diameters of the splined shafts at the bottom of the key-ways. The limit of elasticity and the torque at fracture for the splined pieces are seen to be slightly greater than that for the plain. Yet, when we turn to the work required to produce fracture, we find it takes more energy to break the plain shaft than it does the splined. Mr. Larard merely points out this difference, but does not account for it. It would certainly seem as if the projections should give an added strength to the shaft; yet in the ultimate their presence seems to hasten its downfall. This is bourne out by the appearance of the fractures in Figs. 4, 5, and 6, and it would seem to the writer that while the splines add a slight extra strength to the shaft under static

TABLE 1. MR. LARARD'S TABLE OF RESULTS

No. of specimen	Form of specimen	Diameter of specimen: Outside	Diameter of specimen: Bottom of keyway	Particulars of keyway: Width	Particulars of keyway: Depth at edge	Limit of elasticity in pound-inches	Ratio of elastic torque to elastic twist	Modulus of rigidity: Pounds per square inch	Modulus of rigidity: Tons per square inch	Diameter of equivalent elastic shaft	Elastic resilience to limit of elasticity in inch-pounds: Per unit length	Elastic resilience to limit of elasticity in inch-pounds: Per unit volume	Torque at fracture in pound-inches	Work to produce fracture in inch-tons: From instrument readings on 8 in.	Work to produce fracture in inch-tons: Per unit length	Work to produce fracture in inch-tons: Per unit volume	Ratio of plastic work to elastic resilience	Angle of torsion: By instrument on 8 in. length	Angle of torsion: Per unit length	Angle of helix	
		In.	In.	In.	In.					In.								Deg.	Deg.	Deg.	Min
108	Castellated	$2\frac{1}{2}$	2	$\frac{1}{2}$	$\frac{3}{16}$	16100	53080			2.144	5.266	1.251	135000	457.16	57.14	13.58	24300	545	68.1	33	57
109	Plain	2				14800	40000	11.68×10^6	5215		5.970	1,9	90800	438.6	54.83	17.44	25890	740	92.5	41	29
110	Castellated	$1\frac{57}{64}$	$1\frac{33}{64}$	$\frac{7}{16}$	$\frac{3}{16}$	7400	17420			1.623	3.429	1.484	57700	257.0	32.12	13.89	20980	708	88.5	34	17
111	Plain	$1\frac{33}{64}$				6400	13720	12.01×10^6	5380		3.256	1.804	41600	281.5	35.19	19.51	24200	1034	129.25	30	20
112	Castellated	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{32}$	$\frac{1}{8}$	4430	9166			1.382	2.334	1.283	38000	161.7	20.21	11.10	20490	690	86.25	39	16
113	Plain	$1\frac{11}{32}$				3850	8387	11.79×10^6	5265		3.049	2.127	28100	195.6	24.45	17.07	17960	1040	130.0	33	9
114	Castellated	$1\frac{1}{4}$	1	$\frac{1}{4}$	$\frac{1}{8}$	2750	3300			1.070	2.498	2.39	16550	101.1	12.64	12.11	11330	999	124.9	36	17
115	Plain	1				2680	2568	11.47×10^6	5120		3.048	3.812	11700	122.6	15.32	19.16	11260	1563	195.4	30	9
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	

conditions, they subtract from it under dynamic conditions where fatigue is likely to result.

11 Taking the energy required to fracture as shown in column 17, the plain shaft could be considered 30 per cent to 60 per cent stronger than a shaft with splines added. The immediate conclusion therefore would be that calculations for strength should be based at most on the diameter at the bottom of the splines.

FIG. 7 "CRADLE" IN WHICH TRUCK WAS SURGED TO AND FRO BY ITS OWN POWER

12 In order to try out the value of this conclusion in practical fashion, a truck was loaded up to full 5 tons capacity. Its rear wheels were then anchored somewhat in a cradle, which would allow about 6 ft. of travel, as shown in Fig. 7. A driver then made the truck surge to and fro, the extent of the alternations being measured. The idea was to give the maximum stress on the axles by obtaining the full force of the motor as well as the rotational energy of the flywheel acting against the inertia of the truck. One shaft was left parallel in its length, the other was turned down in the centre. The former broke first and the latter some little time after, 5141 blows being necessary. The total twist in the second exceeded 700 deg., whereas the twist in the first was through an exceedingly short length, all

being concentrated near the end of the splines (Fig. 6). It is interesting to note in Fig. 8 the beginning of the planes of cleavage.

13 From this test it is evident that the conclusion is justified that the diameter at the bottom of the splines is the one to be taken in calculations for strength. Accordingly, the design was changed to that of Fig. 9 in which the diameter at the bottom of the splines is 1.75 in. The shaft formula therefore gives

$$\text{Maximum stress per square inch} = \frac{321.000 \text{ h.p.}}{nd^3}$$
$$= 36{,}500 \text{ lb. per sq. in., approx.}$$

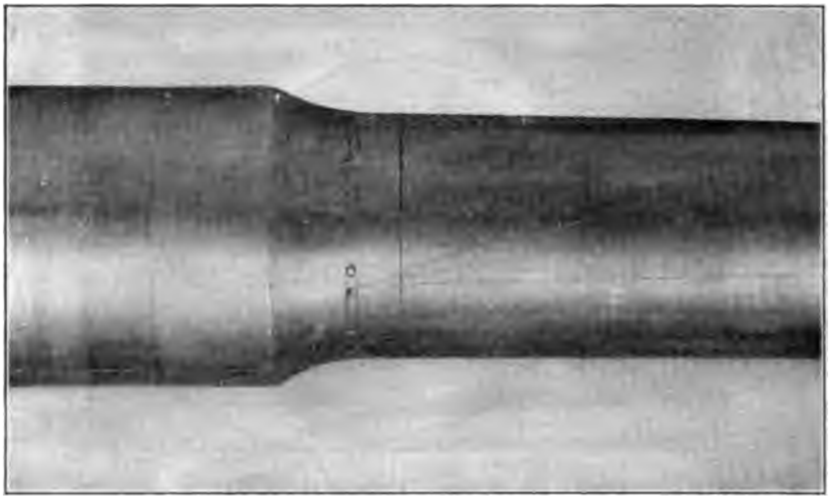

FIG. 8 LINES SHOWING PLANES OF CLEAVAGE IN A SHAFT ABOUT TO BREAK

This gives a factor of safety of 2.5 based on the elastic limit, or 2.9 based on maximum tensile strength.

BRAKING PROBLEMS

14 This particular design of truck has a powerful brake located near the transmission, so that all the braking effort is transmitted by the rear axle shaft to the wheels. We are quite safe in assuming a maximum load on each rear wheel of 8000 lb. and can assume a coefficient of adhesion of the rubber tire to the road, of 0.6. The diameter of the wheels is 40 in., so that torque on shaft is 96,000 in-lb. The writer does not believe that we really get this torque owing to the elastic rubber tires absorbing part of this blow; but assuming that it

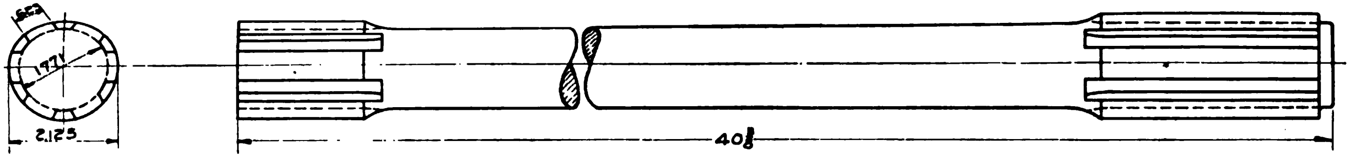

FIG. 9 SECOND DESIGN OF SHAFT

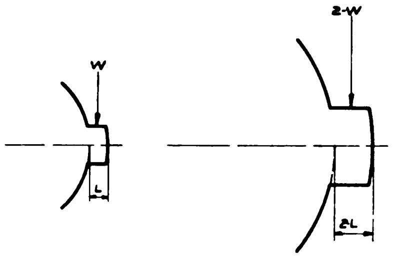

FIG. 10 ILLUSTRATING EFFECTS OF DOUBLING SPLINES AROUND THE CIRCUMFERENCE

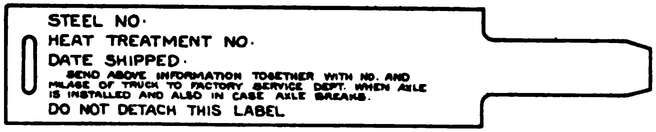

FIG. 12 LABEL ATTACHED TO EACH HEAT TREATED SHAFT

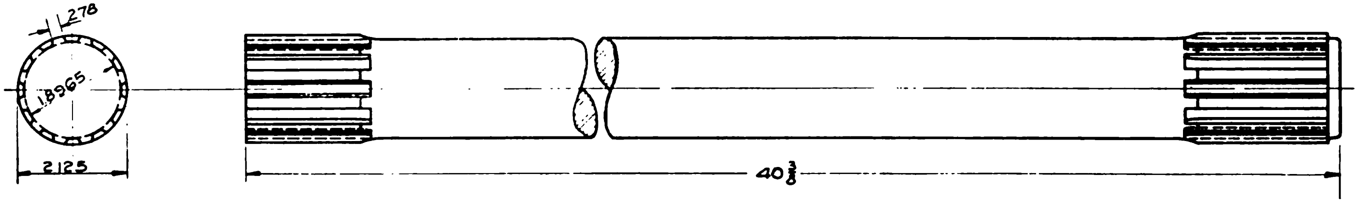

FIG. 11 FINAL FORM OF SHAFT

were obtained, we would have a maximum stress of 93,000 lb. per sq. in. set up in the axle from the formula

$$\text{Twisting moment or torque} = \frac{\pi f r^3}{2}$$

(It must be understood that the above loads are not normal working loads, but are the maximum that can be expected.)

15 It is obvious, therefore, that the shafts which have broken may have been considerably overstressed, although a large number of these actually ran high mileages up to 30,000 and 40,000 miles before breaking. For this reason, it was felt that a comparatively small increase in strength might be sufficient to prevent such breakages and a set of experiments was put in hand to determine the governing factors. The necessity for this lay in the fact that it was almost impossible to increase the diameter of the shafts, which was of course the obvious step to take.

DESIGN

16 A study of the results of Mr. Larard's paper, and of the fractures shown in Figs. 3, 4, 5, and 6, convinced us that the large splines were a very grave source of weakness. If each spline is considered as a small cantilever of breadth $2b$ jutting out from the shaft, with a load of $2W$ upon it, the bending moment at the junction will be $2Wb$.

17 If now we take double the number of splines, but keep the same total bearing area, each spline will have half its former breadth and carry half its former load. The bending moment will then be $\frac{Wb}{2}$, or only ¼ of what it was before (Fig. 10). In addition to the gain in strength by this change, the diameter of the shaft at the bottom of the splines can be increased by the height of one spline. and yet keep the outside diameter the same.

18 It was felt, also, that as the strength should be calculated from the diameter at the bottom of the splines, it would be advisable to turn down the shaft in the middle and so reap the advantage of having it uniformly strong throughout its length, and avoid trouble due to sudden changes in torque-resisting values.

19 This, therefore, gave a shaft (Fig. 11), in which 12 splines were used, with the effective diameter increased from 1.75 in. to 1.8965 in. This was an increase in strength of

$$\frac{1.9^3}{1.75^3} = \frac{6.86}{5.36} = 28 \text{ per cent, approx.}$$

20 The body of the shaft was ground and polished to avoid all scratches. It was found from examination of fractures that the slightest flaw was enough to start a breakdown. Particular attention was given to the junction of the main body of the shaft with the parts of larger diameter used for the splines, to avoid grooving of the filleting tool. It is interesting to note in this connection that the cause of failure of an axle shaft on a locomotive tender in September 19, 1914, involving the wreck of a passenger train on the B. & O. Railroad, was directly attributed to the presence of serrations or score marks on the surface of the axle. Their presence caused the axle to fail gradually under the influence of the dynamic stress to which it was subject. A full report of this was given in the Iron Trade Review for March 20, 1915, and is very interesting reading.

ERRORS IN MANUFACTURE

21 The designer naturally looks first to the manufacturing end as the cause of his troubles and one of the first investigations was with the object of assuring ourselves that the axle was subjected to pure torque and not a combined bending and twisting stress. Half-a-dozen complete axles, some of which had run up large mileage with no fracture, and the remainder had fractured in their early life, were inspected in detail. No difference was found, the only errors present being those that fell within what could only be considered as limits of tolerance. This disposed effectually of improvement being effected in machining process. Unfortunately our improved shaft, while somewhat stronger and of longer life, also broke down in the long run, and further improvement was desired.

MATERIAL

22 This left only the material question to be studied. It had first been the impression that the best material available was being used and certainly the physical tests given at the beginning of the paper indicate a high grade of alloy steel. The writer had noticed the curious fact, however, that the energy required to fracture a piece of ordinary soft mild steel was practically the same as that required to fracture an exactly similar piece of high-grade alloy steel.

23 Mr. Larard in his paper also noted this fact, finding that the work required to twist 3 per cent nickel steel shafts, oil hardened, was about the same as that required with soft carbon-steel shafts.

Several tests which the writer has witnessed have even shown that a soft steel bar requires more energy to break it than does a hardened bar. The question, therefore, at once suggested itself: "Should the shaft be of a softer steel capable of twisting more under a suddenly applied load, and afterwards returning to normal conditions; or should it be of a very much harder steel, which would be more rigid and not deflect so much, but which would require a greater load to break?"

24 A careful examination of printed matter shed no light on the subject. Engineers habitually specified maximum elongation and reduction of area, with the idea that they were getting great toughness. A large number of tests of pieces under fatigue seemed to give all kinds of results; and after witnessing many tests the writer came to the conclusion that the reason for such variation was in all probability that the elastic limit was not what it seemed, but very much lower, and that steels stressed just over the real elastic limit behaved differently from those that had not been stressed to this point. By far the greater number of tests of steel have been made to see how they will break, and very few as to how they will not break.

25 The writer therefore reasoned that if the elastic limit could be raised materially, the shafts would stand up better. Experiments made with small heat-treated specimens, which could be bent by hand, indicated that this was on the right path and accordingly, after some experiments, two shafts were heat-treated to 175,000 lb. elastic limit (measured in the usual way), and tested on a truck in very severe service. These stood up.

26 Further tests showed the desirability of increasing the carbon content somewhat, and several shafts were made to the new specifications and sent out on hard service with excellent results.

27 The material was a domestic steel of the following characteristics:

Chemical	Per cent	Physical	Lb. per sq.in.
Carbon about	.30	Elastic limit	175,000
Manganese	.50	Tensile strength	185,000
Chromium	1.5		Per cent
Nickel	3.5	Elongation in 2 in.	14
		Reduction of area	53

28 These experiments proved so satisfactory that in September 1913 a number of these heat-treated shafts were sent out to replace

others. Not one of these nor the original experimental shafts have broken, to the best of the writer's knowledge.*

29 The shafts are machined from hot-rolled bars already heat-treated to show an elastic limit of about 100,000 lb. They are then heated in a gas furnace to a temperature of between 1450 and 1500 deg. fahr., and quenched in oil. (This double treatment is to insure the grain being properly refined.) They are then reheated to a little over 700 deg. fahr., and allowed to cool slowly in air. Some trouble was experienced at first with warping, but slight experimenting showed they could be readily straightened when hot, under a press. Each individual shaft is then put under the brinell hardness machine, and its Brinell number read at the ends and the middle of the bar. This should be 402 to 444. It is then wrapped around with a metal tag, (Fig. 12), used with the idea of getting reliable results on the different steels we were trying out in our search for a suitable material. In this connection it is interesting to note that 5 per cent nickel steels, chrome vanadium steels, and air hardening steels were tried out and so far all have been standing up to service. The physical specifications of all these steels are very much alike. The success, therefore, seems due entirely to the higher elastic limit, especially as a number of these axles had six flutes and were not of the later 12-flute type.

SECONDARY RESULTS OF INVESTIGATIONS

30 It was thus definitely established that under pure torsional conditions the strength of a shaft is increased by increasing its elastic limit by heat treatment.

31 It would naturally follow, therefore, that similar results could be expected from other components under tensile or shear stress, and accordingly greater attention has been paid to heat treating. The elastic limit has been raised in many pieces with corresponding advantage and in addition as a special measure of precaution each important structural forging or bar is submitted to test on either the scleroscope or the brinell machine to insure that the piece has been properly treated. Both these machines give very reliable readings which can be directly compared with the elastic limit and by their use a number of forgings seemingly all right, but actually either too hard or too soft, have been detected.

32 As regards the economic aspect of the use of heat-treated steels, it is found that this process costs between 2 and $2\frac{1}{2}$ cents per

*This is true to date, April, 1916.

pound and as the strength can in many cases be nearly doubled it clearly effects a large saving.

33 Alloy steels, properly heat-treated, are indispensable for motor truck work, as their capacity to resist fatigue is very great. The writer looks forward to their extended use among other mechanical engineers, just as the ball bearing first introduced for cycle and automobile work is now becoming a universal friction saver. As a permanent repair the heat-treated piece of steel is in many cases worth its weight in gold.

DISCUSSION

Radclyffe Furness[1] (written). Not being a mechanical engineer, it would be a presumption for me to criticise the early part of the paper, but, in passing, I should like to point out that by increasing the carbon in the material Mr. Younger increased the true elastic limit, and, therefore, the proportion between this and the yield point, thus having a much greater factor of safety in the higher carbon material than existed in the low carbon material.

I should like to emphasize two points which I feel I may be permitted to do from my experience with the metallurgy of steel:

First, the desirability of an engineer making a thorough and careful study of the physical properties of the metal which he has in mind to use, in conjunction with the work that he expects this metal to perform, the test conditions being exactly similar to the actual conditions. This has been done by the author, to my mind, in a most thorough manner.

Second, Mr. Younger mentions that alloy steels properly heat-treated are indispensable to motor truck work. Since this is so, it immediately comes to the mind of any one familiar with metallurgy that the converse is true, namely, that alloy steels improperly heat-treated are actually a menace to all work where they are used, and, in addition, should never be used when not given a known and thoroughly tried heat treatment.

In the fabrication of any steel forging, one is obligated to heat the steel above the critical temperature (which is the temperature at which remarkable changes take place in the molecular arrangement), and, as the final physical properties depend upon the rate at which the steel cools through the critical range, the number of degrees heated above the critical range, or the time that it has been

[1]The Midvale Steel Company, Philadelphia, Pa.

held at the temperature above the critical range, when forgings are being manufactured one can readily see that the material is actually being heat-treated.

Alloys are added to steel because they emphasize the changes that can be brought about in the molecular arrangement by heat treatment, and thus increase the physical properties that can be obtained from the material in hand. Thus, one is emphasizing, by the addition of alloys, the difference that can be obtained by varying heat treatments, and, if the forgings are given no definite heat treatment and are being turned out at different rates, the varying heat treatments which are given to the material are infinite. Although it is possible in fabrication to turn out an article which has all the properties of a piece that has been put through a carefully thought-out and prescribed heat treatment, the likelihood of accomplishing this, when the chances of producing other results are infinite, is so small as to be negligible. In addition, one would have many forgings of varying physical properties from the varying heat treatments which the forgings of necessity received. Therefore, it becomes necessary to heat-treat the material to bring it to a known standard, and, since one has increased the susceptibility by addition of alloys, it is evidently more necessary to heat-treat alloy steels than simple carbon steels, and the danger of un-heat-treated alloy steels is far greater than of simple carbon steels, as I have endeavored to make plain, by their increased susceptibility to treatment.

I take this opportunity of mentioning the above, which may be well known to all the members, as I feel it is the duty of all those who are interested in the success of alloy steels, and the improvement of the arts in which they may be used, to emphasize on all occasions the desirability of putting the steels in that condition which will give the best possible results, and removing the common, but fast-dying impression that a steel, because it is a nickel, nickel-chrome or chrome-vanadium steel, is better than a plain carbon steel, no matter what heat treatment it may have received, when, as a matter of fact, the contrary is really true.

Cornelius T. Myers (written). The author has treated his subject with thoroughness, and his conclusions are worthy of more than mental note. The strides made by the automobile industry, comparatively a young industry in this country, have not been accomplished without much careful application of mechanical and metallurgical engineering. Analogous problems confront us in the older industries, and the suggestions that can be gotten from even a cursory study of the methods in vogue in automobiledom should make such a study well worth while.

H. WADE HIBBARD asked the author to define in his closure what he means by "elastic limit." Is it the commercial elastic limit obtained by drop of beam, or is it a more scientific elastic limit obtained by extensometer, electric contact or otherwise?

Regarding the strains which occur in connection with reversal of load, he said we have the results of Wöhler's experiments and of the experiments of those who have followed Wöhler, accounts of all of which are to be found in any good book on machine design.

THE AUTHOR. In reply to Mr. Furness, one method of guarding against making a mistake in heat-treated alloy steels is to subject each individual forging to a brinell or scleroscope hardness test. Forgings of alloy steel are expensive, and I believe that it is just as necessary to test them for heat treatment as it is to ascertain if they measure correctly.

Answering Professor Hibbard, the elastic limit of 175,000 lb. is by drop of beam,—the ordinary commercial elastic limit. Extensometer tests show a slightly lower elastic limit; and from observation of tests of the axles and from breaking pieces of steel, I feel that the real elastic limit is even lower. For the reason of this lower elastic limit, many of the axles broke after running perhaps 8,000 to 15,000 miles, although they were calculated right, taking into account what might be called a normal elastic limit.

The stress in a material subjected to alternating stresses is not the stress based on what might be called the normal line, but is the summation of the two stresses; and the elastic limit based on this summation should be taken as very much lower than we take it. I should say the elastic limit is probably roughly anything from about one-half to two-thirds of what we think it is.

I have found that in cases such as are discussed in the paper, it is of no use making breaking tests on the testing machine of a few meagre-sized specimens. The only real test is that made of a full size piece under practical working conditions. All the theoretical tests, including Sankey bending machine tests, Avery impact tests and the ordinary tensile tests, showed this high tensile steel was not as satisfactory as what engineers call tough material, that is, material of a lower elastic limit, with a higher elongation and higher contraction.

No. 1483

ON THE LAWS OF LUBRICATION OF JOURNAL BEARINGS

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I. INTRODUCTION

Purpose and scope of the paper.[1] This paper deals with the *physics* of lubrication. The principal purpose is to set up certain general relations which may be taken as a guide in the planning and interpretation of experiments to secure data for design. Readers interested only in results available for immediate use are referred to the discussion of the *Ideal Bearing* in Section V. That part of the paper is intended merely as an illustration of the way in which the foregoing general relations may be applied to particular cases, but the results are doubtless applicable as a first approximation to bearings of whatever type.

2 **Relation of lubrication laws to design.** After the requirements of strength and rigidity have been met, there may remain a question as to length and diameter which must be settled by reference to the laws of lubrication. Evidently too short a bearing is in danger of abrasion, while too long a bearing entails needless dissipation of power.

Let the coefficient of friction, f, be defined by the equation

$$f=\frac{F}{L} \qquad [1]$$

in which F is the frictional resisting force and L the load on the bearing perpendicular to its axis. Let the bearing pressure, p, be defined by the equation

$$p=\frac{L}{l\,D} \qquad [2]$$

[1]See page 198 for complete Index to Notation.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

in which l is the length of the bearing and D the diameter of the journal. Let p_o denote the carrying power or greatest permissible bearing pressure. Then the shortest permissible length of a bearing l_o, may be calculated from the equation

$$l_o = \frac{L}{D} \cdot \frac{1}{p_o} \qquad [3$$

while the power dissipated in this bearing at a speed of n revolutions per unit time will be

$$P = \pi \, D \, n \cdot L \, f \qquad [4$$

3 Equations [3] and [4] are purely formal and their practical use demands a knowledge of some relation

$$f = \varphi \, (p, n, D, l, \text{etc.}) \qquad [5$$

between the coefficient of friction and all the physical quantities governing the action of lubrication; together with some relation

$$p_o = \psi \, (n, D, l, \text{etc.}) \qquad [6$$

between carrying power and the factors which it may depend upon.

4 Equations [5] and [6] symbolize the most important laws of lubrication needed in design; and they may be termed the *law of friction* and the *law of carrying power* respectively. There are, of course, still other laws of lubrication which may be of interest to the designer, for example that giving the pressure distribution around the bearing; but this paper is exclusively devoted to the law of friction and the law of carrying power.

5 Before proceeding to the physical discussion of these two laws, in which the object will be to deduce the general form of the functions φ and ψ, the role which they play in bearing design may be indicated by a numerical example. In this example φ and ψ will arbitrarily be given particular forms, in order to contrast the results obtained on two different assumptions as to the form of the carrying power function ψ.

6 *Example* 1. Suppose that a machine carrying a load of 300 lb. on each of its two bearings, and running at a speed of 1000 r.p.m. has been found by experience to be designed with the smallest safe bearing surface when the bearings are made 1 in. in diameter and 3 in. long; the total dissipation under those circumstances having been found by an actual test to be about 1/20 h.p. Let it be required to design the bearings of a new machine of the same type carrying the same load, but running 2000 r.p.m. Let two solutions be made, first assuming the carrying power to vary inversely as th

surface speed, second assuming it to vary as the cube root of the surface speed; in each case taking the coefficient of friction to be directly proportional to the square root of the surface speed and inversely proportional to the bearing pressure.

7 Expressed in symbols, we are to assume the friction has the form

$$\varphi\,(p, n, D, l, \textit{etc.}) = \textit{const.}\ \frac{\sqrt{n\,D}}{p} \qquad [7]$$

while on our first assumption ψ has the form

$$\psi\,(n, D, l, \textit{etc.}) = \frac{\textit{const.}}{n\,D} \qquad [8]$$

and on our second assumption it has the form

$$\psi\,(n, D, l, \textit{etc.}) = \textit{const.} \sqrt[3]{n\,D} \qquad [9]$$

8 In order to simplify further the calculation, we will assume that considerations of strength and rigidity permit the diameter to remain unchanged.

9 By the use of equations [1] to [9] in conjunction with the data given, the two solutions are readily obtained. Our first assumption leads to the conclusion that the bearings of the new machine should each be 6 in. long, while the power dissipated will be about ¼ h.p. On our second assumption the length of each bearing should be 2⅝ in., while the total dissipation will be about 1/10 h.p.

10 Thus, on our first assumption for carrying power, the new machine must have its bearings twice as long as the prototype machine, and it will dissipate five times the power; while on our second assumption the new machine may have its bearings a little shorter than the prototype, and will dissipate about twice the power only.

11 Evidently then, it is important to decide whether carrying power shall be assumed to be less or greater at a higher speed. The former assumption is the one commonly given in text books. But the latter assumption appears to represent the practice of the General Electric Co.,[1] and will be shown in this paper to be deducible from physical facts.[2]

[1] Alford, Bearings and their Lubrication, 1911, p. 81.

[2] See equation [63].

II. PRELIMINARY PHYSICAL CONSIDERATIONS

12 **Definitions of friction and carrying power.** Already the coefficient of friction has been defined as the ratio of frictional resistance to the load on the journal perpendicular to its axis; while carrying power has been defined as the greatest permissible bearing pressure. In order then to reduce the subject to a physical basis, it remains to define unequivocally the terms *frictional resistance* and *greatest permissible bearing pressure.*

13 The frictional resistance of a lubricated bearing, conceived as a single force, is a fictitious quantity. It is the resisting torque or so-called moment of friction which we actually measure in an experiment. To be exact therefore we may define frictional resistance as the quotient of resisting torque by the mean radius of the journal.

14 A satisfactory definition of permissible bearing pressure demands a definite understanding of the type of failure we wish to avoid. If we wish to avoid failure by seizing, we must investigate thermal expansion; if we wish to avoid failure by overheating the lubricant, we must investigate the temperature rise; if to avoid creating tension in the lubricant, we must investigate the pressure distribution; if to avoid simple abrasion, we must investigate the minimum film thickness. The last type of failure is the only type we shall undertake to discuss in this paper.

15 If c denotes *radial clearance* or mean difference in radii between journal and bearing, while x denotes the *film thickness* or thickness of the film of lubricant at the point of nearest approach, the fraction $\frac{x}{c}$ may be called the *relative film thickness.* In this paper, we shall make it a matter of definition that all bearings are equally *safe* which are running with the same relative film thickness. Accordingly, carrying power may be defined as that bearing pressure which reduces the relative film thickness to some prescribed value $\left(\frac{x}{c}\right)_0$.

16 Determination of the laws of lubrication symbolized by equations [5] and [6] therefore simmers down to the investigation of the effect of various conditions on F and x.

17 **Restrictions necessary to exclude unfamiliar phenomena.** In order to narrow the problem to as simple a one as possible

without excluding any important practical circumstances, we may impose the following restrictions:

1 The bearing must be in a steady state.

2 The lubricant must be homogeneous.

3 The bearing must be running below the critical speed at which eddy motion would be set up in the lubricant.

4 The effect, on the motion of the lubricant, of any other forces than hydrostatic pressure and shearing stress must be negligible.

5 The metal surfaces must always be separated by a film of lubricant which is thick enough to have the same mechanical properties it would have in bulk.

6 There must be no resultant couple acting on the bearing in the plane of its axis.

18 The first of these restrictions excludes consideration of speeding up or slowing down; heating up or cooling off; and intermittent load. The second excludes dirty or badly emulsified oil or any mixed solid and liquid lubricant. The third and fourth restrictions, which are introduced to justify us in ignoring the density and surface tension of the lubricant, need not exclude any important case, for it may be shown that the clearance would have to be very wide and the speed extraordinarily high to create eddies, while the capillary, centrifugal, and gravitational forces acting on any part of the oil film are very small compared with the viscous drag exerted on that same part. The fifth restriction excludes the initial friction on starting up a machine from rest, even though it is accelerated slowly enough to sensibly be in a steady state. The sixth restriction excludes any upward belt pull on a journal mounted in a single bearing. However, we need not exclude forced lubrication, eccentric load (e.g., downward belt pull on journal in single bearing), oil-grooves, worn bearings, or any other purely geometrical irregularity.

19 **Qualitative discussion of action of lubrication.** We shall undertake to show qualitatively, that, under the foregoing restrictions, F and x are completely determined by the following physical quantities:

a The viscosity, μ, of the lubricant.

b The revolutions per unit time, n.

c The load, L.

d The degree of lubrication; which, in the case of *stationary* lubrication (i.e., the limiting case when no lubricant enters or leaves the bearing) may be specified by V, the volume of lubricant in the bearing; and which in the case of forced lubrication may be specified by the quantity, Q, of the lubricant flowing through the bearing in unit time.

e The absolute size of the bearing,[1] which may be given by the diameter of the journal, D.

f The line of action of the load, defined by some length ratio r' such as the ratio of its distance from the middle point of the bearing, to the diameter.

g The shape of the bearing: specified by the relative clearance $\frac{c}{D}$, the relative length $\frac{l}{D}$, and such other length ratios r'', r''', etc., as may be needed to fix the shape of the oiling arrangements, deviation from circular section due to wear, departure from cylindrical form due to strain, and all other geometrical irregularities.

20 Thus, we wish to show that F and x are uniquely determined by the quantities $\mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r$; in which r denotes all the ratios r', r'', r''', etc.

21 Consider the exaggerated cross-section of a typical bearing shown in Fig. 1. In such a bearing the action taking place under ordinary conditions may be summarized thus: The lubricant will adhere to the metal surfaces while they move. A shearing stress will then be set up at every point within the lubricant, and the work done shearing it, will continually be dissipated into heat. If we divide up the lubricant into prismatic elements like the one whose cross-section is shaded in on the diagram, we may think of the frictional resistance F as the sum of all the tangential resisting forces ΔF exerted on the journal by the elements which are being sheared. Owing to the wedging action of the lubricant as it is dragged under the journal at the point of nearest approach, the load will not be able to force the journal quite into contact with the bearing; instead, the journal will find a position of equilibrium with a certain eccentricity such that at some point, whose location we are not concerned with, the lubricant will have a minimum film thickness x.

[1]The term *bearing* will frequently be used for short to denote the entire system: bearing, lubricant, and journal.

22 While it might be stated categorically as a consequence of the recognized facts of fluid motion, that F and x cannot depend on any other quantities than $\mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r$, it is more profitable to consider these one by one, and to visualize the effect which each of them has, or may have, on the action of lubrication.

23 *Effect of viscosity.* The fundamental empirical law of fluid motion which underlies the reasoning in this paper, is that force, ΔF, required to shear a layer of fluid of area Δa and thickness X, is directly proportional to the relative velocity v of the two surfaces, and to the area, and inversely to the thickness, provided the velocity

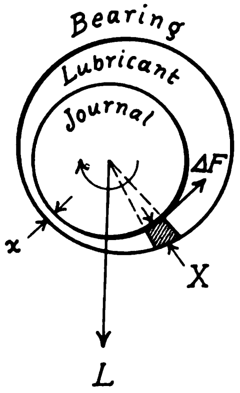

FIG. 1 EXAGGERATED CROSS-SECTION OF BEARING

at every point within the film is parallel to the two bounding surfaces. If the relation be written

$$\Delta F = \mu \frac{v \Delta a}{X} \qquad [10]$$

the constant of proportionality μ is, by definition, the viscosity. The viscosity is of course different for different fluids, and it diminishes rapidly with rise in temperature; but it does not depend appreciably on the pressure, nor, so far as has yet been discovered, does it depend at all on the rate of shear.

24 Let the lubricant be divided into elements by radial planes which are equally spaced. For every such element equation [10] is approximately true, the approximation being closer the less the clearance, eccentricity, and angular width of the element.

25 From equation [10] we see that increasing the viscosity will increase the resistance of each element and hence of the bearing as a whole, provided the X's, and hence the film thickness, x, be unchanged.

26 However, x itself will increase with increasing viscosity, the journal approaching a concentric position, owing to the greater wedging action of the more viscous lubricant.

27 This in turn tends to decrease F. For, any given displacement of the journal will change the thickness of the thin elements by a larger per cent than it will that of the thick elements; hence, any

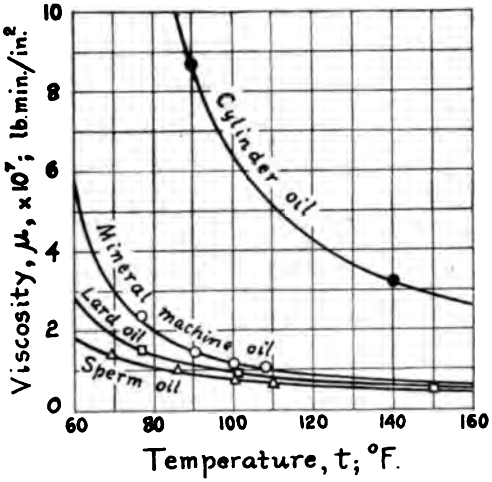

FIG. 2 VISCOSITY-TEMPERATURE CURVES

given increase in x increases the small X's, and so decreases F, by a greater per cent than F is increased by the simultaneous decrease in the large X's. We are not yet in a position to judge which effect will preponderate, and it is conceivable that, if a bearing is running so heavily loaded as to have a very eccentric journal, we may actually decrease the friction by feeding in a more viscous oil.

28 We conclude that in general both F and x depend on μ.

29 *Effect of speed.* From equation [10] we see that at constant x, F would increase with the surface speed v and hence with n. However, x itself must increase when n increases, owing to the greater wedging action at higher speeds. Just as in the case of viscosity,

we cannot yet say which effect will preponderate—the increase in resistance due directly to the speed, or the indirect decrease in resistance due to the increase in film thickness with speed. It is clear, however, that at high enough speeds so that the journal is sensibly concentric with the bearing, the total resistance, other factors remaining constant, must increase directly as the speed.

30 Hence we conclude that F and x depend on n.

31 *Effect of load.* L does not appear in equation [10], and at constant x there is no conceivable way the load can have anything to do with the frictional resistance. This explains why the coefficient of friction, which is merely a name for F divided by L, has been found in practice to vary inversely as the load in high speed bearings; that is, in bearings running fast enough or loaded lightly enough to remain sensibly concentric. The load, however, must have some effect on film thickness, x approaching zero as L becomes indefinitely great; hence the load may affect the resistance indirectly, but this effect will be appreciable only at low speeds.

32 We conclude that in general both F and x depend on L.

33 *Effect of amount of lubricant in the bearing.* At constant x, decreasing V diminishes F because it reduces the number of elements to be sheared. However, the occurrence of open spaces will change x, for it entirely alters the hydrostatic pressure components on the journal.

34 Thus F and x depend on V.

35 *Effect of forced lubrication.* The effect of increasing the quantity of flow Q may either be to increase or decrease x according to where the lubricant is introduced. If it is fed into the bearing at the point of nearest approach, the local pressure there will tend to relieve the load, thus increasing x and decreasing F: an important consideration in practice.

36 F and x then depend on Q.

37 *Effect of diameter.* At constant x, if the diameter be increased without changing the clearance, F will increase in proportion to D owing to the corresponding increase in all the sheared areas Δa of equation [10]. The change in D may also change the film thickness by virtue of its effect on the pressure distribution. For example, in a large bearing we should not expect the local pressure at the oil inlet, due to forced lubrication, to extend over so large an arc as in a small bearing. Hence, to be on the safe side, we must say that in general both F and x may depend on D.

38 *Effect of clearance.* At constant x, increasing the clearance, c, would decrease F owing to its effect on all the X's. If the bearing were concentric, so that X = constant = c, equation [10] shows that F would vary inversely as c. But this change in clearance may also alter the pressure distribution. Hence both F and x may depend on c, and hence on $\frac{c}{D}$.

39 *Effect of length.* If there are no end-effects, F will increase directly with l at constant x owing to the corresponding increase in all the $\Delta\, a$'s; but x itself will increase when l is increased owing to the diminished intensity of the load. And, if there are end-effects, their influence will be diminished by lengthening the bearing.

40 Thus F and x depend on l, hence on $\frac{l}{D}$.

41 *Effect of geometrical irregularities.* Equation [10] being applicable only to elements bounded by stream lines, cannot be used to calculate directly the tangential resistance except when the bearing surfaces are perfectly smooth, circular cylinders. The tangential

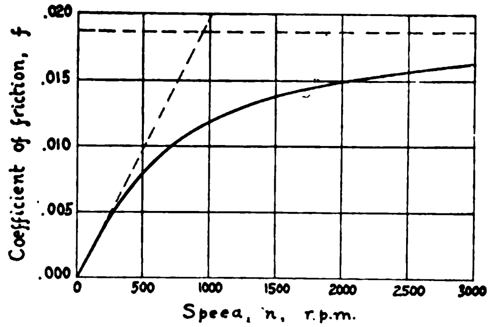

FIG. 3 WORKING CURVE FOR COEFFICIENT OF FRICTION, NUMERICAL EXAMPLE 2

resistance F must, however, depend in some way on the individual shearing resistances of all the elements into which we may properly divide up the lubricant, and these will depend, in turn, on the shape of the metallic surfaces, specified by r'', r''', etc. Likewise the pressure distribution, and therefore also the film thickness, may depend on the shape of these surfaces. For example, in a new bearing the frictional resistance is locally more intense at the point of nearest approach than elsewhere, but in a worn bearing the resistance is equally great over a considerable arc. In general then F and x depend on r'', r''', etc.

42 *Effect of line of action of load.* If the load is shifted nearer one end of the bearing, we should expect the film thickness at that end to decrease and the local friction there to increase. hence F and x depend on r'.

43 *Other conditions.* The assertion that F and x are uniquely determined by $\mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r$, can be disproved only by discovering some physical quantity independent of those nine which can influence F or x. Consider, for example, whether changing the entrance head of the forced lubrication, or the density of the lubricant, or the temperature of the bearing can make any difference in F or x while $\mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r$, remain fixed. In the first place, increasing the head under which the lubricant is fed in will increase the rate of flow through the bearing, unless the clearance be simultaneously decreased, or some analogous change made in at least one of our independent variables. After all, then, it is meaningless to ask what effect changing the entrance head would have at constant μ, n, L, etc., because it is not possible to change the head and still keep the other factors unchanged. Again, the density of the lubricant can influence its action only when turbulent motion, centrifugal force, or gravity come into play; and these cases have been specifically excluded from the present paper. Finally, temperature can influence friction and film thickness only indirectly, through its effect on the clearance and on the viscosity of the oil and, if there is local heating, through its effect on the shape; therefore if $\mu, \frac{c}{D}$, and r are fixed, F and x are independent of temperature.

44 Should any doubt arise as to the justification for assuming viscosity to be the only mechanical property of the lubricant which can affect F and x, attention must again be directed to restriction five. Doubtless new and obscure properties do make their appearance when the film has been crushed to molecular dimensions. The present paper is limited to the consideration of ordinary well lubricated bearings.[1]

[1]Fig. 6 shows approximately what the minimum film thickness in a bearing may be at any given viscosity, speed, and pressure; and it is seen that the clearance must be very wide indeed, or the speed very low, or the pressure very great, to reduce the film thickness to less than say one-quarter of the clearance; which, in bearings of any ordinary size, is hardly thin enough for molecular properties to make their appearance, unless the metallic surfaces are very rough, or badly fitted.

45 *Summary.* The conclusion that F and x depend only on $\mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r$, may be symbolized by the equations

$$\Phi\left(\frac{F}{L}, \mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [11]$$

and

$$\Psi\left(\frac{x}{D}, \mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [12]$$

in which, for convenience later on, F and x have been specified by their ratios to other quantities of the same kind, namely, L and D. A knowledge of these unknown functions Φ and Ψ would suffice to determine the functions φ and ψ of equations [5] and [6]. As they stand, equations [11] and [12] represent only the qualitative facts about lubrication. We shall undertake now to draw from them

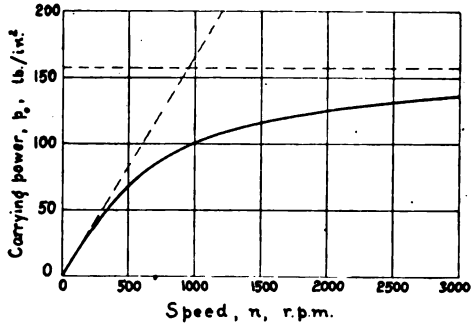

FIG. 4 WORKING CURVE FOR CARRYING POWER, NUMERICAL EXAMPLE 2

some quantitative conclusions, treating the dynamics of the problem separately from heating effects.

III. GENERAL DYNAMICAL RELATIONS

46 **Derivation of the general form of the laws of lubrication by dimensional reasoning.** In order to transform a qualitative statement into a quantitative one, we must of course get additional information from somewhere. In general, a complete, quantitative relation cannot be established without conducting experiments; but an incomplete quantitative relation may often be of great service in planning and interpreting such experiments. Such a relation may be deduced by starting with a complete, qualitative relation, and applying the principle of dimensional homogeneity.

47 If, then, under the restrictions which have been named, equa-

tions [11] and [12] are qualitatively *complete,* that is, if we have not overlooked any physical quantity which may appreciably influence the phenomena, we can apply to them the principle of dimensional homogeneity. This principle states that all the terms of a physical equation have the same dimensions. The best way of utilizing the principle here is by starting off with Buckingham's Π-*theorem,* according to which[1] any correct and complete physical equation must be of the general form

$$f\,(\Pi_1, \Pi_2, \ .\ .\ .\ \Pi_{n-k}, r\text{'s}) = 0 \qquad [13]$$

Here the Π's are all the independent dimensionless *products* of the form $Q^x{}_1\, Q^y{}_2\, Q^z{}_3$ . . . , which can be built up by combining in any way the n different kinds of physical quantity, Q_1, Q_2, . . . Q_n involved in the relation; and the r's are the ratios of each remaining quantity to that of its own kind used in forming the Π's. There will be $n-k$ such products if k is the number of fundamental units needed in measuring the n kinds of quantity. In order to apply the theorem to our present problem, that is, in order to express the laws of lubrication in the form of equation [13], we must now find the appropriate dimensionless products.

48 From equations [11] and [12], it appears that we have to deal with $n=6$ different kinds of physical quantities, namely, μ, n, L, V, Q, D. And, since we are concerned with forces and motions, the number of fundamental units needed is $k=3$. Hence the number of independent dimensionless products which can be made up from the six quantities μ, n, L, V, Q, D is $n-k=3$. On the force, length, and time system, the dimensions of the five kinds of quantity are:

$$\begin{aligned} \mu &= f\, l^{-2}\, t \qquad \text{(See Eq. [10])} \\ n &= t^{-1} \\ L &= f \\ V &= l^3 \\ Q &= l^3\, t^{-1} \\ D &= l \end{aligned}$$

in which f denotes force, l length, t time. By inspection of this list, the products $\Pi_1 = \mu\, n\, D^2\, L^{-1}$, $\Pi_2 = D^3\, n\, Q^{-1}$, and $\Pi_3 = V\, D^{-3}$ are seen to be dimensionless; hence equations [11] and [12] may at once be thrown into the forms

$$\Phi_1\left(\frac{F}{L}, \frac{\mu\, n\, D^2}{L}, \frac{D^3\, n}{Q}, \frac{V}{D^3}, \frac{c}{D}; \frac{l}{D}, r\right) = 0 \qquad [14]$$

[1] See Phys. Rev. 4, pp. 345-376, Oct. 1914, for derivation and discussion.

and

$$\Psi\left(\frac{x}{D}, \frac{\mu n D^2}{L}, \frac{D^3 n}{Q}, \frac{V}{D^3}, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [15]$$

But

$$\frac{F}{L} = f, \text{ and } \frac{\mu n D^2}{L} = \frac{\mu n}{\frac{L}{lD}\cdot\frac{l}{D}} = \frac{\mu n}{p}\cdot\frac{D}{l},$$

$$\frac{V}{D^3} = \frac{V}{D l c}\cdot\frac{l}{D}\cdot\frac{c}{D}, \textit{ and } \frac{x}{D} = \frac{x}{c}\cdot\frac{c}{D};$$ so that [14] and [15]

are identical with the equations

$$\Phi_1\left(f, \frac{\mu n}{p}\cdot\frac{D}{l}, \frac{D^3 n}{Q}, S.\frac{l}{D}\cdot\frac{c}{D}, \frac{c}{D}\cdot\frac{l}{D}, r\right) = 0 \qquad [16]$$

and

$$\Psi_1\left(\frac{x}{c}\cdot\frac{c}{D}, \frac{\mu n}{p}\cdot\frac{D}{l}, \frac{D^3 n}{Q}, S.\frac{l}{D}\cdot\frac{c}{D}, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [17]$$

Here S has been written for $\frac{V}{\pi D l c}$, the ratio of the volume of lubricant in the bearing to the whole volume of the clearance space; and which may be termed the *relative supply*. But there is no need of duplicating our symbols for the variables entering the relations, hence [16] and [17] may further be simplified by writing them in the equivalent forms

$$\Phi_2\left(f, \frac{\mu n}{p}, \frac{D^3 n}{Q}, S, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [18]$$

and

$$\Psi_2\left(\frac{x}{c}, \frac{\mu n}{p}, \frac{D^3 n}{Q}, S, \frac{c}{D}, \frac{l}{D}, r\right) = 0 \qquad [19]$$

respectively. Solving [18] for the coefficient of friction now gives

$$f = \varphi\left(\frac{\mu n}{p}, \frac{D^3 n}{Q}, S, \frac{c}{D}, \frac{l}{D}, r\right) \qquad [20]$$

Likewise solving [19] for $\frac{\mu n}{p}$, inverting, transposing, and remembering that by definition $p = p_0$ when $\frac{x}{c} = \left(\frac{x}{c}\right)_0$, gives for the carrying power

$$p_0 = \mu\, n \cdot \theta \left[\left(\frac{x}{c}\right)_0, \frac{D^3 n}{Q}, S, \frac{c}{D}, \frac{l}{D}, r \right] \qquad [21]$$

The functions φ and θ are of course unknown, and remain to be determined by experiment.

49 Equations [20] and [21] correspond to equations [5] and [6] respectively, and contain the two laws of lubrication in the most general form, consistent with the restrictions of Par. 17.

50 **Discussion and simplification.** Consider the forms to which equations [20] and [21] reduce when some of the arguments of φ and θ are held constant. If a bearing is sensibly cylindrical, (though not necessarily circular in cross-section), and free from end-effects, and uniformly loaded along its length, we can imagine it cut in two without changing either f or $\frac{x}{c}$; for both F and L will be altered in the same ratio as the length, and the wedging force of the film is cut down in the same ratio as the load to be supported. For such a bearing $\frac{l}{D}$ cannot enter the equations. If there is no cavitation, $S = 1$, and may be treated as a constant instead of a variable. If the cross-section of the bearing-surface remains geometrically similar to itself, the r's also become constant. Hence for all cylindrical bearings which are free from cavitation or end effects; centrally or uniformly loaded; and which have bearing surfaces that are geometrically similar in cross-section, (though not necessarily having the same relative clearance),

$$f = \varphi_1 \left(\frac{\mu\, n}{p}, \frac{D^3 n}{Q}, \frac{c}{D} \right) \qquad [22]$$

and

$$p_0 = \mu\, n \cdot \theta_1 \left[\left(\frac{x}{c}\right)_0, \frac{D^3 n}{Q}, \frac{c}{D} \right] \qquad [23]$$

51 For all bearings in which S, $\frac{l}{D}$, and r are the same, even though not cylindrical and centrally loaded; and in which there is no appreciable thrust due to forced lubrication, Q cannot enter the equations; hence the whole term $\frac{D^3 n}{Q}$ drops out and the equations reduce to

$$f = \varphi_2\left(\frac{\mu\, n}{p}, \frac{c}{D}\right) \tag{24}$$

and

$$p_0 = \mu\, n \cdot \theta_2\left[\left(\frac{x}{c}\right)_0, \frac{c}{D}\right] \tag{25}$$

An important particular case to which **[24]** and **[25]** are applicable is that of the *Ideal Bearing* discussed in Section V, where we shall see that the more specific results which Sommerfeld has deduced for this case by integrating the equations of hydrodynamics are, in fact, reducible to the form of **[24]** and **[25]**.

52 For bearings free from cavitation ($S = 1$), similarly loaded and geometrically similar throughout (r, $\frac{l}{D}$, and $\frac{c}{D}$ constant),

$$f = \varphi_3\left(\frac{\mu\, n}{p}, \frac{D^3\, n}{Q}\right) \tag{26}$$

and

$$p_0 = \mu\, n \cdot \theta_3\left[\left(\frac{x}{c}\right)_0, \frac{D^3\, n}{Q}\right] \tag{27}$$

The laws of lubrication for all such bearings of whatever size can be established experimentally by varying the two quantities $\frac{\mu\, n}{p}$ and $\frac{D^3\, n}{Q}$; and this can be done on a single bearing.

53 If, in the same bearings to which **[26]** and **[27]** apply we now diminish the degree of lubrication until there is no appreciable thrust due to any flow of lubricant into the bearing, still keeping S constant, though not necessarily equal to 1, the argument $\frac{D^3\, n}{Q}$ drops out and we have

$$f = \varphi_4\left(\frac{\mu\, n}{p}\right) \tag{28}$$

$$p_0 = \mu\, n \cdot \theta_4\left[\left(\frac{x}{c}\right)_0\right] \tag{29}$$

Evidently, however, a like result would have been reached if $\frac{D^3\, n}{Q}$, instead of being made to vanish, had simply been kept constant; e.g., if the quantity of oil pumped through any bearing in unit time were made to bear a constant ratio to the volume of the bearing and to its speed of rotation. Any two bearings in which $\frac{D^3 n}{Q}$ has the same value, and in which S also has the same value, may be called *similarly lubricated.*

54 Comparing general equations [**20**] and [**21**] with [**28**] and [**29**] we now see that the former are equivalent respectively to the following statements:

a. In geometrically similar bearings which are similarly loaded and lubricated, *the coefficient of friction depends only on the single variable* $\frac{\mu n}{p}$.

b. The carrying power of any bearing is directly proportional to the product of viscosity by revolutions per unit time; the constant of proportionality being the same for all geometrically similar bearings which are similarly loaded and lubricated and which are equally safe, i.e., $\left(\frac{x}{c}\right)_0$ constant.

55 The writer made several years ago some experiments on journal friction and carrying power the results of which are entirely consistent with the above conclusions.[1] These experiments are of less interest now than at the time they were performed, for it was not then so commonly recognized that bearings may safely carry heavier loads at the higher speeds.

56 **Dynamically similar bearings.** Any two geometrically similar bearings B and B' similarly loaded and lubricated, and which are running at the *corresponding* speeds, pressures, and viscosities defined by the equation

$$\frac{\mu n}{p} = \frac{\mu' n'}{p'} \qquad [30]$$

must, by equations [**18**] and [**19**], have the same coefficient of friction and the same relative film thickness. Such bearings may be termed *dynamically similar.* The power dissipated in either of them may be calculated from a test made on the other, for by [**4**], if $f = f'$,

$$\frac{P}{P'} = \frac{D}{D'} \cdot \frac{n}{n'} \cdot \frac{L}{L'} \qquad [31]$$

Now, equation [**21**] can be written

$$p_0 = \theta_0 \cdot \mu n \qquad [32]$$

[1]For a brief account of the experiments see Journ. Wash. Acad. Sci., v. IV, p. 549, 1914.

in which θ_0 denotes the value of the function θ when $\frac{x}{c}$ has the particular value $\left(\frac{x}{c}\right)_0$. As dynamically similar bearings have the same relative film thickness, $\theta_0 = \theta_0'$ and [32] gives for the two bearings B and B'

$$\frac{p_0}{p'_0} = \frac{\mu}{\mu'} \cdot \frac{n}{n'} \qquad [33]$$

Thus if the carrying power of one bearing has been found experimentally, that of the other can at once be calculated.

IV. A GENERAL METHOD FOR DETERMINING THERMAL EFFECTS

57 The principal effect of temperature, and the only effect we need analyze, is to decrease the viscosity of the lubricant as the bearing heats up. Consequently both friction and carrying power increase with speed less rapidly, under working conditions, than

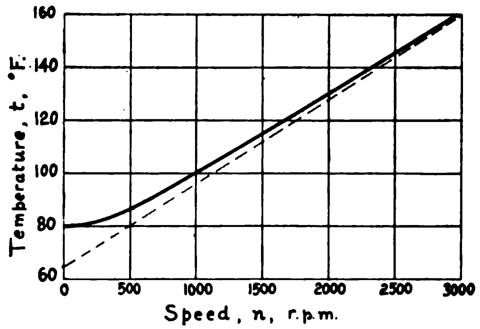

FIG. 5 PERMANENT RUNNING TEMPERATURES, NUMERICAL EXAMPLE 2

they would at constant temperature. The general dynamical equations [20] and [21] are true regardless of temperature, because they have been expressed in terms of the actual viscosity of the lubricant in the film at the moment in question; but it is desirable to go a step further. Equations like [20] and [21], which describe the behavior of a bearing when the viscosity is given, may be termed *characteristic equations* for that bearing. On the other hand, an equation describing the behavior of a bearing, not in terms of the instantaneous viscosity, but entirely in terms of known constants or controllable conditions like speed and load, may be termed a *working* *equation* for that bearing. Such an equation does not characterize the bearing in itself, but depends also on the nature of the lubricant and on the cooling system. We shall now outline a general method for determining the working equations for the friction and carrying power of any bearing. The method consists in eliminating viscosity from the characteristic equations by utilizing information about the lubricant and the cooling system.

58 Suppose the function θ of the characteristic carrying power equation [**21**] has been completely determined; let it now be required to eliminate the viscosity from this equation. If the viscosity-temperature curve for the oil to be used has been empirically determined, we can evidently replace the μ of equation [**21**] by some function of the temperature, t. But this too is unknown, so we must not stop here. Now the number of heat units, H, carried off

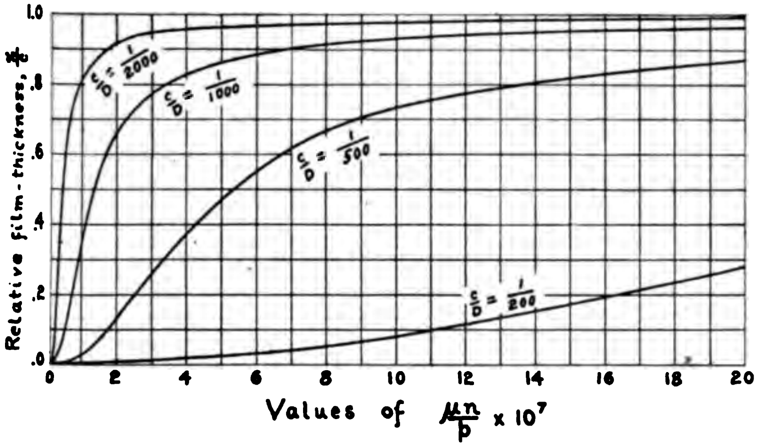

FIG. 6 CHARACTERISTIC CURVES FOR RELATIVE FILM-THICKNESS OF IDEAL BEARING

in unit time by the air, the oil itself, the jacket-water, or otherwise, will evidently depend on the temperature; so that if this relation between H and t has once been determined empirically, we can replace the t of our carrying power equation by some function of H. Now when the bearing has been running long enough to have reached a steady state, the heat carried off in unit time must be equal to the heat generated in unit time, which can be calculated in terms of the coefficient of friction; so that H can be replaced by a function of f. Finally, this function of f can be replaced by some function of μ, provided the function φ of equation [20] is known; so that we now have five independent relations from which to eliminate the four quantities μ, t, H, and f. To recapitulate, the five relations available are those connecting the quantities

$$\begin{array}{lcl} p_0 & \text{and} & \mu \\ \mu & \text{and} & t \\ t & \text{and} & H \\ H & \text{and} & f \\ f & \text{and} & \mu \end{array}$$

respectively. To eliminate viscosity from the carrying power equation, all five must be solved simultaneously; to eliminate it from the friction equation, it is sufficient to work with the last four of these. Let us now examine the form of these relations.

59 The first is the characteristic equation for carrying power, equation [21].

60 Any empirical equation for viscosity of an oil in terms of temperature must involve certain constants a', a'', etc., which are different for different oils: if we denote all these by a, the second relation can be written

$$\mu = F_1 (t, a) \qquad [34]$$

in which the function F_1 is supposed to have been determined empirically.

61 Likewise the heat carried away in unit time at any temperature t may be expressed by some empirical equation

$$H = F_2 (t, b) \qquad [35]$$

in which b denotes all of the constants b', b'', etc., entering the function F_2: they will of course depend on the room temperature, the size and shape of the bearing and adjacent parts, the conductivity of the metal, the rate of circulation of the cooling agent, etc.; but we need not know the character of this dependence as long as we do not attempt to extend equation [35] to other conditions than those under which the constants b were determined. A familiar approximate form of equation [35], known as Newton's law of cooling, states that H is proportional to the excess of the temperature, t, over that of the surroundings.

62 The fourth relation is obtained by equating $J\,H$, the mechanical equivalent of the heat carried off in unit time, to the power dissipated according to equation [4]. If we replace the L of equation [4] by its equivalent $p\,D\,l$, the fourth relation becomes

$$H = \frac{\pi}{J} D^2 \, l \, n \, p \, f \qquad [36]$$

63 The fifth relation is the characteristic equation [20] for the coefficient of friction.

64 Eliminating μ and H from the five relations leads at once to the three general equations

$$p_0 = F_1(t, a) \cdot n \cdot \theta \left[\left(\frac{x}{c} \right)_0, \frac{D^3 n}{Q}, R \right] \qquad [37]$$

$$F_2(t, b) = \frac{\pi}{J} D^2 \, l \, n \, p \cdot \varphi \left[\frac{F_1(t, a) \cdot n}{p}, \frac{D^3 n}{Q}, R \right] \qquad [38]$$

$$f = \varphi \left[\frac{F_1(t, a) \cdot n}{p}, \frac{D^3 n}{Q}, R \right] \qquad [39]$$

in which R has been written for all the ratios $S, \frac{c}{D}, \frac{l}{D}$, and r. These three relations may be regarded as a formal statement of the proposed method for determining thermal effects. The functions θ and φ are to be found by dynamical experiments; F_1 and F_2 by thermal experiments. After they have been determined, we can deduce a working equation for the carrying power of the desired form

$$p_0 = \psi_0 \left[\left(\frac{x}{c} \right)_0, n, D, l, Q, R, a, b \right] \qquad [40]$$

by eliminating t from [37] and [38]. Likewise, by eliminating t from [38] and [39], we can deduce a working equation for the coefficient of friction of the desired form

$$f = \varphi_0 \, (p, n, D, l, Q, R, a, b) \qquad [41]$$

Incidentally from [38] alone we may get an equation of the form

$$t = \zeta \, (p, n, D, l, Q, R, a, b) \qquad [42]$$

for the temperature of equilibrium or permanent running temperature of the bearing.

65 If any of the empirical functions θ, φ, F_1, or F_2 prove too complex to represent analytically, we can still accomplish the desired eliminations graphically.

66 The relations [37], [38], and [39], are perfectly general and not limited to any particular type of bearing or any particular lubri-

cant or cooling system. In the following pages we shall illustrate the use which can be made of such relations by treating special cases in which simple expressions may be taken for the empirical functions.

V. PROPERTIES OF THE IDEAL BEARING

67 **Definition of the Ideal Bearing.** Consider a bearing which is subject not only to the six fundamental restrictions set forth in Section II, but to the following special conditions as well:

a Let the bearing surfaces be perfectly smooth circular cylinders.

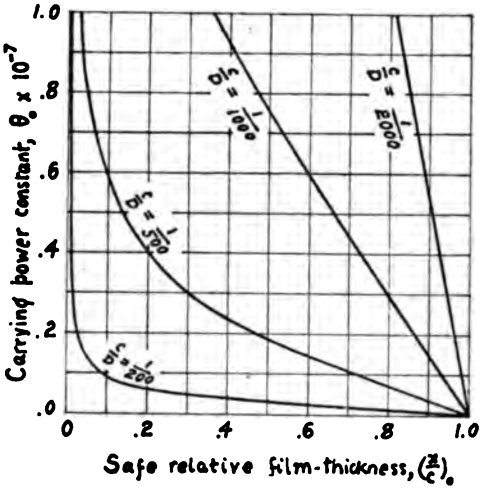

FIG. 7 CARRYING POWER CONSTANTS OF IDEAL BEARING

b Let the resultant load be applied exactly at the center of the journal.

c Let the bearing be long enough so that end-effects are negligible.

d Let the lubricant be stationary ($Q = 0$) except for the circumferential motion, and free from cavitation ($S = 1$). This implies no pressure drop at the open ends.

68 Such a bearing, which of course can never be absolutely realized in material form, may be termed the *Ideal Bearing*.

69 **Characteristic equation for high speed bearing.** Consider for the present an ideal bearing so lightly loaded or running at

such a high speed that the journal and bearing are *approximately* concentric. Such a bearing may for short be designated by the term *high speed bearing.*

70 By reference to equations [1], [2], and [10], the characteristic equation for the coefficient of friction of a high speed bearing is readily seen to be

$$f = \pi^2 \frac{D}{c} \frac{\mu n}{p} \qquad [43]$$

for we may take for Δa of equation [10] the whole surface of the bearing $\pi D l$; and for v, the surface speed of the journal $\pi D n$; and for X, the clearance c; ΔF then becoming F. As a check we may note that [43] agrees with [24].

71 From [25] we may write for the characteristic equation of carrying power

$$p_0 = A \mu n \qquad [44]$$

in which the constant of proportionality A is supposed to have been determined empirically. As an example of one way A might thus be determined without attempting to deduce the expression $\theta_2 \left[\left(\frac{x}{c} \right)_0, \frac{c}{D} \right]$ to which it is equivalent, let p_0', μ', and n' denote respectively the bearing pressure, actual viscosity, and speed of some high speed bearing already in service, having the same relative clearance though it need not be of the same size as the proposed bearing, and running with a satisfactory degree of safety which we now desire to reproduce. Under these circumstances we may use equation [33]. Writing it in the form

$$p_0 = \left(\frac{p'_0}{\mu' n'} \right) \mu n \qquad [45]$$

and comparing with [44] we see that

$$A = \frac{p'_0}{\mu' n'} \qquad [46]$$

72 **Convenient approximations for viscosity and cooling functions.** The viscosity-temperature curves of most lubricating oils can be roughly fitted by an empirical equation of the type

$$\mu = F_1 (t, a) = \frac{a'}{t - a''} \qquad [47]$$

that is, by treating them as rectangular hyperbolas relative to an axis located at some temperature a'' not greatly different from the true solidifying temperature. Fig. 2 shows the viscosity temperature relation for the four oils used by the writer in his own experiments, the curves being plotted from equation [**47**]. While these data are by no means recommended as standard values, they are sufficient for illustrating the principles discussed.

73 As an approximation for the cooling function we may conveniently adopt Newton's law of cooling, writing it

$$H = F_2\ (t, b) = b'\ (t - b'') \qquad [48]$$

74 For the purpose of utilizing the three relations [**37**], [**38**] and [**39**], in deducing the working equations of an ideal bearing, equations [**47**] and [**48**] may be used as they stand. But for the purpose of afterward giving the results a simple physical interpretation, they may be written

$$\mu = \frac{\mu_o\ (t_o - \tau)}{t - \tau} \qquad [49]$$

and

$$H = h\ (t - t_0) \qquad [50]$$

respectively. Here μ_o denotes the original viscosity of the oil at the room temperature t_o; τ is the apparent solidifying temperature; and h the heat carried away from the bearing in unit time per unit temperature elevation above room temperature. Thus the relations between the two sets of constants are

$$a' = \mu_o\ (t_o - \tau) \qquad [51]$$

$$a'' = \tau \qquad [52]$$

$$b' = h \qquad [53]$$

$$b'' = t_o \qquad [54]$$

75 **Working equations for permanent running temperature, friction, and carrying power of a high speed bearing.** If in [**37**], [**38**], and [**39**] we now substitute the expressions for θ, φ, F_1, and F_2, given by [**44**], [**43**], [**47**], and [**48**], respectively, the three general relations reduce to the following specific forms:

$$p_o = \frac{a'}{t - a''} \cdot n \cdot A \qquad [55]$$

$$b'(t - b'') = \frac{\pi}{J}\ D^2\ l\ n\ p \cdot \pi^2\ \frac{D}{c} \left(\frac{\frac{a'}{t - a''} \cdot n}{p} \right) \qquad [56]$$

$$f = \pi^2 \frac{D}{c} \left(\frac{\frac{a'}{t-a''} \cdot n}{p} \right) \qquad [57]$$

76 Solving [56] as a quadratic equation in t gives

$$t = \tfrac{1}{2}(b''+a'') + \tfrac{1}{2}(b''-a'')\sqrt{1 + \frac{4\pi^3}{J} D^2 l \frac{D}{c} \cdot \frac{a'}{b'(b''-a'')^2} \cdot n^2} \qquad [58]$$

an equation for permanent running temperature of the form [42]. If in place of the a's and b's we now substitute the constants given by equations [51] to [54] we arrive at the more intelligible form

$$t = \tfrac{1}{2}(t_o + \tau) + \tfrac{1}{2}(t_o - \tau)\sqrt{1 + k n^2} \qquad [59]$$

in which

$$k = \frac{4\pi^3}{J} \cdot \frac{\mu_o}{h(t_o - \tau)} \cdot D^3 \cdot \frac{\left(\frac{l}{D}\right)}{\left(\frac{c}{D}\right)} \qquad [60]$$

The quantity k may be termed the *heating constant*; the greater it is, the hotter the bearing will run at any given speed. Evidently k may be determined from an observation of the permanent running temperature at any one speed, for by solving [59] for k we see that

$$k = \frac{\left(\frac{2\,t - t_o - \tau}{t_o - \tau}\right)^2 - 1}{n^2} \qquad [61]$$

77 Eliminating t between equations [56] and [57] affords a working equation for coefficient of friction of the form [41]; which, if we again change over from the a's and b's to μ_o, τ, t_o and h becomes

$$f = \pi^2 \frac{D}{c} \frac{\mu_o n}{p} \left[\frac{2}{1 + \sqrt{1 + k n^2}} \right] \qquad [62]$$

78 From equations [55] and [56], in the same manner, we deduce the working equation for carrying power

$$p_o = A \mu_o n \left[\frac{2}{1 + \sqrt{1 + k n^2}} \right] \qquad [63]$$

79 Equation [62] shows that as the speed is indefinitely increased the coefficient of friction approaches asymptotically the limiting value

$$f_{max} = \pi^2 \frac{D}{c} \frac{\mu_o}{p} \frac{2}{\sqrt{k}} \qquad [64]$$

From **[63]** it is seen that the carrying power likewise approaches a limiting value

$$p_{o\,max} = A\,\mu_o \frac{2}{\sqrt{k}} \tag{65}$$

80 *Example* 2. To obtain an illustration of the working equations of a high speed bearing, let us substitute successively in equations **[61]**, **[62]**, and **[63]**, the following numerical values:

$t = 100$ deg. fahr. at $n = 1000$ r.p.m.
$t_o = 80$ deg. fahr.
$\tau = 50$ deg. fahr.
$\mu_o = 2 \times 10^{-7}$ lb. per in.^{2} $\times$ min.
$\frac{c}{D} = \frac{1}{1000}$
$p_o = 100$ lb. per in.^{2} when $n = 1000$ r.p.m.
$p = 100$ lb. per in.^{2}

From the above seven values we find that

$$k = 4.44 \times 10^{-6} \text{ sec}^2$$

and

$$A = 8.46 \times 10^5$$

hence the working equations for this bearing become

$$f = \frac{1.97}{10^5} n \left[\frac{2}{1 + \sqrt{1 + \frac{4.44}{10^6} n^2}} \right] \tag{66}$$

and

$$p_o = \frac{1.69}{10^3} n \left[\frac{2}{1 + \sqrt{1 + \frac{4.44}{10^6} n^2}} \right] \tag{67}$$

n being in r.p.m. and p_o in lb. per in.^{2} These results are plotted in Figs. 3 and 4 respectively.

81 The corresponding equation for the permanent running temperature, **[59]**, becomes in fahr. degrees

$$t = 65 + 15 \sqrt{1 + \frac{4.44}{10^6} n^2} \tag{68}$$

This equation is plotted in Fig. 5.

82 If we further adopt the values

$J = 778$ ft-lb. per B.t.u. $= 9330$ in-lb. per B.t.u.
$D = 1$ in.

and $\frac{l}{D} = 3$

equation [60] when solved for the cooling constant h gives

$$h = 5.88 \text{ B.t.u. per min. per deg. fahr.}$$

83 The numerical values which have been used would apply to the first of the two bearings treated in the numerical example at the beginning of the paper, and to an oil whose viscosity-temperature curve is approximately that of the mineral machine oil in Fig. 2.

84 **Characteristic equations for bearing with eccentric journal, based on Sommerfeld's theory.** Sommerfeld's investigation of the theory of lubrication[1] led to equations[2] which in our notation may be written

$$\frac{\mu n}{p} = \frac{1}{3\pi^2}\left(\frac{c}{D}\right)^2 \cdot \frac{\left[3-2\frac{x}{c}+\left(\frac{x}{c}\right)^2\right]\sqrt{\frac{x}{c}\left(2-\frac{x}{c}\right)}}{1-\frac{x}{c}} \qquad [69]$$

and

$$f = 2\frac{c}{D} \cdot \frac{1-\frac{4}{3}\frac{x}{c}+\frac{2}{3}\left(\frac{x}{c}\right)^2}{1-\frac{x}{c}} \qquad [70]$$

The assumptions underlying his deduction are such that equations [69] and [70] are, in effect, characteristic equations for an ideal bearing. Due account having been taken of the eccentricity, [69] and [70] are not confined to the high speed bearing.

85 In Fig. 6, equation [69] has been plotted in such a way as to show how the relative film thickness $\frac{x}{c}$ varies with $\frac{\mu n}{p}$ for a number of different values of the relative clearance. From [69], we see that the characteristic equation for carrying power may be written, as before,

$$p_0 = \theta_0 \cdot \mu n \qquad [32]$$

in which

$$\theta_0 = \frac{3\pi^2}{\left(\frac{c}{D}\right)^2} \cdot \frac{1-\left(\frac{x}{c}\right)_0}{\left[3-2\left(\frac{x}{c}\right)_0+\left(\frac{x}{c}\right)_0^2\right]\sqrt{\left(\frac{x}{c}\right)_0\left[2-\left(\frac{x}{c}\right)_0\right]}} \qquad [71]$$

[1] Zs. f. Math. u. Phys., v. 50, pp. 97-155, 1904. (See also his shorter account in the Archiv. f. Elektrotech., v. III, pp. 1-5, 1914).

[2] loc. cit., Equations [46] and [48], pp. 124 and 125.

Values of θ_0 for different relative clearances have been plotted against $\left(\frac{x}{c}\right)_0$ in Fig. 7.

86 Characteristic curves for coefficient of friction against $\frac{\mu n}{p}$ for different relative clearances are shown in Fig. 8; this diagram

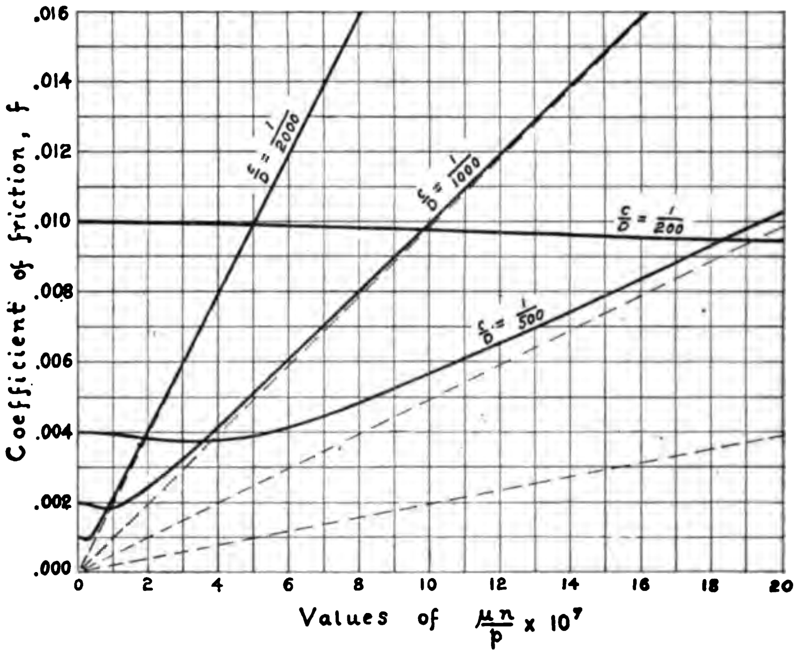

FIG. 8 CHARACTERISTIC CURVES FOR COEFFICIENT OF FRICTION OF IDEAL BEARING

having been constructed by eliminating $\frac{x}{c}$ between equations **[69]** and **[70]** by a graphical method.

87 Inspection of these diagrams serves to verify the fact, deduced analytically by Sommerfeld, that the minimum coefficient of friction,

$$f_{\min} = \frac{4}{3} \sqrt{2} \frac{c}{D} \qquad \textbf{[72]}$$

occurs at a *transition value*

$$\left(\frac{\mu\, n}{p}\right)_{\min} = \frac{5}{6\,\pi^2}\left(\frac{c}{D}\right)^2 \qquad [73]$$

and at a film thickness equal to about 0.29 of the clearance. As $\frac{\mu\, n}{p}$ approaches zero, the coefficient of friction is seen to approach the *initial value*

$$f_0 = 2\,\frac{c}{D} \qquad [74]$$

As $\frac{\mu\, n}{p}$ becomes indefinitely great, the coefficient of friction approaches asymptotically the straight line defined by the characteristic equation [43] of the high speed bearing.

88 It is useful to remember that actual viscosities of lubricating oils under ordinary running conditions are of the order of one ten-millionth of a lb-wt., inch, minute unit; such, of course, being the unit we must employ, if n is to be expressed in r.p.m. and p in lb. per sq. in.

89 **Working equations for bearing with eccentric journal.** It will be sufficient for the present to determine the shift in the location of the minimum point of the friction equation. If the transition speed shifts from n_0 to n while the temperature changes from t_0 to t, we see from [73] that

$$\frac{n}{n_0} = \frac{\mu_0}{\mu} \qquad [75]$$

But from [49]

$$\frac{\mu_0}{\mu} = \frac{t-\tau}{t_0-\tau} \qquad [76]$$

$$\therefore \frac{n}{n_0} = \frac{t-\tau}{t_0-\tau} \qquad [77]$$

Substituting from [50] and [72] into [38] gives

$$h(t-t_0) = h\left[(t-\tau)-(t_0-\tau)\right] = \frac{\pi}{J}\,D^2\,l\,n\,p \cdot \frac{4}{3}\sqrt{2}\,\frac{c}{D} \qquad [78]$$

Solving for $t-\tau$ and substituting in [77] gives

$$\frac{n}{n_0} = 1 + \frac{4\,\pi\,\sqrt{2}\,D\,c\,l\,p}{3\,J\,h\,(t_0-\tau)}\,n \qquad [79]$$

90 Substituting the numerical values used in previous example, and noting from [73] that $n = 42$ r.p.m., it appears that $\frac{n}{n_0}$ will

exceed unity by only about one part in twenty thousand, which is negligible. Hence from [79] we may conclude that the effect of temperature is sensibly the same, with an eccentric journal, as it would be with a concentric journal.

91 If, then, the relations between f or p_0 and n obtained by putting $\mu = \mu_0$ in the characteristic equations of any bearing be termed the *isothermal equations* we see from the foregoing, in conjunc-

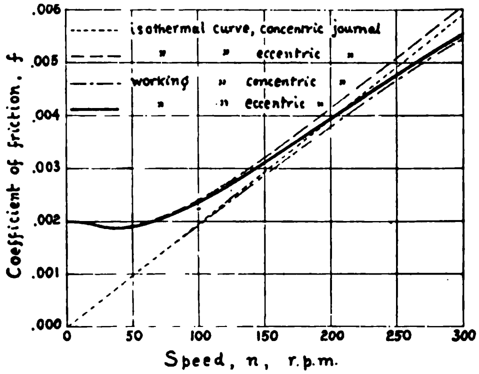

FIG. 9 WORKING CURVE FOR COEFFICIENT OF FRICTION, NUMERICAL EXAMPLE 3

tion with equations [62] and [63], that the working equations of an ideal bearing, whether running fast or slow, may ordinarily, as a first approximation, be constructed from its isothermal equations simply by multiplying the isothermal values of f or p_0 by the reduction factor

$$\frac{2}{1 + \sqrt{1 + k n^2}} .$$

92 *Numerical Example* 3. Fig. 9 gives the working curve for coefficient of friction at low speeds for the same bearing which has been treated in our previous numerical example. This was obtained from Fig. 8 by first constructing the corresponding isothermal curve, namely the uppermost curve shown in Fig. 9, and then multiplying all the ordinates of the isothermal curve by the reduction factor

$$\frac{2}{1 + \sqrt{1 + \frac{4.44}{10^6} n^2}}$$

93 The carrying power curve for this example has been given in Fig. 4. For, since carrying power curves are curves of constant film thickness, any such curve which is correct for a given bearing at high speeds, holds equally well at the lowest speeds.

VI. SUMMARY AND CONCLUSION

94 By way of summarizing the developments of the paper attention may be called to the following points:

1. The general form of the laws of lubrication which must be conformed with by any equations which are to be generally valid, whether obtained mathematically or empirically, has been given by equations [**20**] and [**21**].

2. These equations are, however, applicable only when the six fundamental restrictions discussed in Section II are sensibly realized; and then only when the term *carrying power* is defined in a particular way.

3. The distinction between the characteristic equation and the working equation for a bearing has been emphasized. If we are to bring any order at all out of the chaos of variables involved in the problem of completely mapping out the laws of lubrication, it is necessary to take advantage of every opportunity for dividing the problem into independent parts which may be separately investigated. The dynamical characteristics of the bearing, and the thermal characteristics of the lubricant, and of the cooling system, are three such parts.

4. A general method has been outlined for combining the results of these three investigations, after they have been separately carried out, and thus deducing the desired working equations.

5. In order to illustrate concretely the meaning of these various general principles, and also in order to afford at once a first approximation to the data needed in design, the properties of the *ideal bearing* have been defined and worked out. In doing this advantage has been taken of Sommerfeld's mathematical results, which, in fact, merit wider attention than they have yet received.

95 In conclusion, it is clear that what is needed next is a minute examination of existing experimental data in the light of the foregoing general principles, with a view to coördinating them; and, if possible, piecing together the results in such a way as to make our present knowledge of the laws of lubrication less fragmentary. If it then appears that further experiments are needed in order to establish a

rational basis for design, we may be guided as to the most economical way to plan the experiments, by reference to the general equations [**20**] and [**21**], and by adopting the principle of the separation of dynamical from thermal effects. For example, the cooling function F_2 might in some cases be determined with a stationary dummy bearing, in which the heat is generated by a heating coil, instead of by friction, and thus much more accurately measured. Finally it is to be observed that the conception of dynamically similar bearings may make it possible to evaluate some of the constants needed in design by the use of models, without waiting for a complete determination of the laws of lubrication.

APPENDIX

INDEX TO NOTATION

Symbol	Meaning	Equation in which it first occurs
f	coefficient of friction, F/L	[**1**]
F	frictional resistance	[**1**]
L	normal component of load	[**1**]
l	length of bearing	[**2**]
D	diameter of bearing	[**2**]
p	bearing pressure, $L/l\,D$	[**2**]
p_0	carrying power, (greatest permissible bearing pressure)	[**3**]
l_0	shortest permissible length	[**3**]
n	speed, in revolutions per unit time	[**4**]
P	power dissipated in friction	[**4**]
ΔF	frictional resistance of an element of the lubricant	[**10**]
Δa	area of element in plane of shear	[**10**]
X	thickness normal to plane of shear	[**10**]
v	surface speed of element and of journal	[**10**]
μ	viscosity of the lubricant	[**10**]
V	volume of lubricant in bearing	[**11**]
Q	quantity of flow, (volume of lubricant flowing through bearing in unit time)	[**11**]
c	mean radial clearance	[**11**]
r	abbreviation for the ratios r', r'', r''', etc., fixing line of action of load and shape of bearing	[**11**]
x	minimum film thickness (see Fig. 1)	[**12**]
S	relative supply, $V/D\,l\,c$	[**16**]
$\left(\frac{x}{c}\right)_0$	assumed value of safe relative film thickness	[**21**]
θ_0	carrying power constant, (giving the carrying power when multiplied by the product of viscosity and revolutions per unit time)	[**32**]
t	temperature of lubricant	[**34**]
F_1	empirical function in viscosity-temperature equation	[**34**]
a	abbreviation for the constants a', a'', etc., in viscosity-temperature equation	[**34**]
H	heat carried off from bearing in unit time	[**35**]
F_2	empirical function in equation for heat carried off	[**35**]
b	abbreviation for the constants b', b'', etc., in equation for heat carried off	[**35**]

J	mechanical equivalent of heat	[36]
R	abbreviation for S, $\frac{c}{D}$, $\frac{l}{D}$, and r	[37]
A	value of carrying power constant θ_0 for ideal bearing	[44]
μ_0	viscosity at room temperature	[49]
t_0	room temperature	[49]
τ	fictitious solidifying temperature of the lubricant, (one of the a's)	[49]
h	cooling constant, giving the heat carried off in unit time per unit temperature elevation above room temperature	[50]
k	heating constant, defined by equation [60]	[59]

The symbols ϕ, Φ, ψ, Ψ, θ, ζ, etc., denote undetermined functions of the arguments immediately following in parentheses, and of no others.

DISCUSSION

H. F. Moore (written). The author's results on carrying power are qualitatively confirmed by the writer's experience. Experiments made by the latter in 1903 with a bearing of babbitt metal on a steel journal showed that before the film of oil broke down the carrying power increased with some function of the speed; and so far as the tests went the bearing power was found to vary approximately with the square root of the speed. Later tests made at the University of Wisconsin on very carefully ground hardened steel journals rotating in bronze bearings showed a carrying power two or three times as great as did the babbitt metal bearing with unhardened journal. It might be expected that surface finish of journal and bearing would play a very important part in determining the breaking strength of the oil film.

W. H. Herschel. This work is a notable step in bringing order out of the chaos of experiments on the friction of journals. If it can be demonstrated that viscosity is the only property of the lubricant which influences the coefficient of friction and the maximum permissible load, lubricating problems will be greatly simplified. Thanks to Sommerfeld, we feel no longer the necessity of considering "adhesion," introduced into equations by earlier investigators, but there is a widespread belief that lubricants vary in regard to "oiliness," "lubricating value" or "body."[1]

Alford[2] says that "body" has been defined as that property of an oil that influences the change in viscosity when the oil is under pressure. Kapff[3] likewise claims to have proved experimentally that two oils may have the same viscosity at a certain temperature,

[1] A. Kingsbury, Trans. Am. Soc. M. E., vol. 24, p. 143.

[2] L. P. Alford, Bearings and Their Lubrication, p. 112.

[3] S. Kapff, Zeitschrift des Vereines deutscher Ingenieure, 1898, p. 554.

and yet give a different coefficient of friction at that temperature. If this is true, there is a necessary factor of "oiliness" which has been omitted from the author's equations. On the other hand, if it is not the case, and friction depends upon no other property of the lubricant except the viscosity, the importance of viscosimeters is greatly increased, and the value of friction machines for tests of lubricants is correspondingly diminished.

Kapff's contention, however, leads to a contradiction in terms, for since viscosity is a measure of the resistance to motion, if it is a true unit of measurement, it must have the same value whether determined by the efflux method, as in the Saybolt viscosimeter, or by resistance to turning, as in the Doolittle viscosimeter or in friction testing machines. Couette[1] has in fact determined viscosity by both methods with close agreement. We believe, therefore, that there is at present a preponderance of evidence in favor of the author's view.

The author assumes, without other evidence than is furnished by Fig. 2, that temperature-viscosity curves have the form of equation [49]. It would have been preferable, in this connection, to have indicated what degree of approximation might be expected, or the range of temperatures to which the formula applies. It is known that with many mineral oils there is a break in the temperature-viscosity curve at a point not far below 130 deg. fahr., due to the form of paraffine crystals, so that the same equation will not apply both above and below this point.

F. zur Nedden contributed a written discussion in which he compared the author's results with experiments by Professor Guembel described in the Monatsblätter des Berliner Bezirks-Vereines Deutscher Ingenieure, May and June, 1914. He continued:

When gradually reducing (at constant pressure p) the speed of any bearing, the coefficient of friction in the neighborhood of zero, after reaching a minimum, abruptly increases to a high figure, because with very low speeds liquid friction is giving way to semi-dry friction. The laws as set forth in the paper do not then hold good. The point where this change occurs depends partly on the smoothness of the gliding surfaces. This consideration is by no means of only theoretical interest. Toothed wheels running in oil will show lesser friction losses if designed so that the flanks of the teeth glide on each other with a speed exceeding the upper limit of semi-dry friction.

[1] Annal. de chemie et de Phys., 1890, vol. 21, p. 433.

The all-important question whether equations [11] and [12] are qualitatively complete should be further investigated in connection with a long series of very important tests by Professor Schlesinger in contention that it is not correct to assume that the only factor that matters in a lubricant (besides price) is its viscosity.

The writer has come to the conclusion that two properties of the lubricant, both of which are not covered by the author's formulæ, probably have something to do with the process of lubrication. The first and less important one is the specific heat of the oil itself. The second and very important one is the degree of amorphity. Lubrication depends entirely on the wedgelike action of the lubricant film. If the resistance of the lubricant to the mutual dislocation of its molecules is not absolutely the same regardless of the direction of the outer forces which tend to dislocate them—in other words, if the lubricant can be regarded no longer as a real amorphous liquid, the application of Newton's law of viscosity must be modified.

Several very important conclusions can be drawn from the paper. One of the stipulations to be found in almost every specification for turbo-machinery is that length of bearings should equal at least three times diameter. This rule of thumb is probably the outcome of practical experience gained with bearings of the comparatively low speed machinery exclusively used before the advent of the steam turbine. From the paper, it follows the carrying power is different at different speeds.

Viscosity of the oil and heat carrying capacity of the bearing are the essential standards which should be determined, but which are scarcely ever mentioned in any specification. Of these two factors one depends on the buyer who purchases the lubricant. Only the second, the heat carrying capacity, depends on the make of bearing. The author shows a way to determine this quality in a manner which will permit of standardization.

The Author. Professor Moore's own formula, the square root law of carrying power, has been unwarrantably extended by others, and applied to circumstances physically different from those under which it was established. Nevertheless, the historical significance of his experiments of 1903, the first aiming toward a direct determination of carrying power, can hardly be overestimated.

While experimenting with bearings in 1909 at the Massachusetts Institute of Technology, the writer tried to see whether this square root law or any similar relation held under more nearly practical conditions, that is, with a whole bearing instead of a half bearing, and at higher speeds. Under these circumstances complete film rupture never took place, but the load producing some specified electrical resistance other than zero did increase with speed.

As noted by Professor Moore it is to be expected that (except at high speeds) bearings with differently finished surfaces will behave differently; they cannot be considered geometrically similar unless either sensibly smooth or similarly rough—not equally rough, but twice as rough if twice as large.

The viscosity-temperature equation [49] is offered as a particular example of the general equation [34] to show as simply as possible how the operation of deriving the working equations of a bearing from [37] to [39] works out in a concrete case. Equation [49] is, as Mr. Herschel implies, only a first approximation of the facts.

Mr. Herschel alludes to the widespread belief that oils of the same viscosity may still differ in lubricating value. This belief is doubtless correct as regards bearings running at low enough values of $\mu\eta \div p$.

One fact may be recorded for the interest of those who consult the reference Mr. Nedden gives: If the curves on which Professor Guembel's final equations are based be fitted at three points instead of two, a more general type of equation results, which reduces to Guembel's at low values of $\mu\eta \div p$ and to Sommerfeld's at high values.

It is true that the present paper is inadequate to cope with semi-dry friction, and it is likely that Professor Schlesinger's experiments, dealing as they do with the aggregate friction losses of the engine lathe, are largely concerned with semi-dry friction.

The specific heat of the oil has to be regarded as one of the *b's* of equation [35] for heat carried off.

The author does not believe there is, as yet, any need for considering the effect of incomplete amorphity.

Mr. zur Nedden's suggestions are exceedingly interesting, and must certainly be taken account of in the subsequent development of the subject. But for the moment we are so much at sea, that it is well to follow the rule of navigation and keep to deep water, in doing which our six fundamental restrictions will serve as the channel buoys. Any exact charting out of the shoals has, to be sure, been postponed.

No. 1484

THE SURFACE CONDENSER

MODERN THEORY AND PRACTICE

By C. F. Braun, San Francisco, Cal.

Associate-Member of the Society

It is the purpose of this paper to analyze the functions of the surface condenser, present briefly the fundamental principles governing design, discuss rational ratings, and compare typical commercial designs with the principles which are presented.

2 The primary functions of a surface condenser are to reduce the back pressure on the exhaust side of a steam prime mover; to conserve and return to the steam generator the water of condensation, which is a chemically pure feed; to conserve and return to the steam generator as many heat units as possible; and to remove from the feed water air in solution, thus reducing pitting of boilers.

3 In accomplishing these results the condenser must handle four separate fluids: steam, air (including other non-condensable vapors), water of condensation, and cooling or circulating water. These must be considered separately in order to reach a clear understanding of the subject and to make a logical analysis. As the desirable condition or state of these several fluids is not the same, the problem at once becomes a complicated one. Briefly, the conditions which should be approached are as follows:

4 *Steam* should enter the condenser and be conducted freely to all parts thereof with least possible resistance; it should be reduced to the lowest practicable temperature (and consequently pressure) and should be converted into water for easy removal.

5 *Air*, an excellent non-conductor of heat, should be rapidly cleared from the heat-transmitting surfaces, collected at suitable places, practically freed from entrained water and water vapor, and cooled to a low temperature for removal at minimum volume, with consequent least expenditure of mechanical energy.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

6 *Condensate* should also be rapidly cleared from the heat transmitting surfaces, freed from air, collected at suitable points for removal, and returned to the steam generator at the maximum practical temperature.

7 *Circulating water* should pass through the condenser with least friction, deposit a minimum amount of precipitated chemicals or debris, and absorb a maximum amount of heat.

PRINCIPLES OF DESIGN

8 The main factor in accomplishing the desired conditions in the condenser is the transference of heat from the steam through the dividing surface to the cooling water. The transfer of heat per unit of area or of size is a measure of the efficiency of the apparatus and is directly proportional to the temperature difference or head. In a condenser the temperatures of these fluids are different at different parts of the surface. The temperature of the water increases during its passage because of the absorption of heat, and that of the steam decreases because of the frictional drop in pressure. It is therefore necessary to obtain mean values for temperature differences.

9 A simple arithmetic mean is not correct, but the following formula, developed mathematically by Grashof, has repeatedly been proven experimentally as very accurate and is almost universally adopted.

$$M = \frac{D_1 - D_2}{\log_e \dfrac{D_1}{D_2}} \quad \ldots\ldots [1]$$

where M = mean temperature difference
D_1 = temperature difference between fluids at beginning $= T\,S_1 - T\,W_1$
D_2 = temperature difference between fluids at end $= T\,S_2 - T\,W_2$
$T\,S_1$ = initial temperature of steam
$T\,S_2$ = final temperature of steam
$T\,W_1$ = initial temperature of circulating water
$T\,W_2$ = final temperature of circulating water

10 It is commonly assumed that $T\,S$ is constant throughout the condenser, by which [1] reduces to

$$M = \frac{T\,W_2 - T\,W_1}{\log_e \dfrac{T\,S - T\,W_1}{T\,S - T\,W_2}} \quad \ldots\ldots [2]$$

11 Since the frictional drop through the steam space of a condenser is usually 0.5 in. or more, representing with high vacuums a temperature difference of say 10 deg. fahr., it is evident that the use of formula [2] for applying to large condensers data obtained on smaller ones, or for analyzing the performance of a condenser or various sections of a condenser, will lead to serious errors. Further than this, the temperature at the end of the steam space is less than that for saturated steam corresponding to the pressure observed, by an amount depending on the partial pressure of the air present.

12 Since the total heat to be abstracted in condensing 1 lb. of exhaust steam is nearly constant within practical ranges of vacuum, it is apparent that the maintenance of high vacuum with temperatures rapidly approaching the temperature of the entering circulating water requires apparatus of much larger size, proportionately, than indicated by the vacuum, due to the decreasing value of the mean temperature difference, which figures for the first case approximately 27 deg. fahr., and for the second 13.5 deg. fahr.

13 With any given set of temperature values this mean temperature difference can be varied in only one way, namely by arrangement of heating surfaces. These must be such as to produce counter-current flow, the circulating water entering where the steam is coolest (away from exhaust inlet) and leaving where it is hottest (at exhaust inlet).

14 *Transfer of heat* through a unit of condenser tube area per unit of mean temperature difference was early recognized as varying greatly under different conditions, the most apparent variation being an increase with increase of water velocity. Many experimenters have carried out extensive and careful tests to determine values of this heat transfer, the most practical of which are probably those of Orrok, and nearly all have developed formulas purporting to express a relation between heat transfer and water velocity, a common one being $H = K V^{1/2}$.

15 That such a formula is fundamentally incorrect and misleading is at once apparent when it is considered that certain resistances to heat flow, namely that of the tube and that on the steam side of the tube, are practically constant and entirely independent of the water velocity.

16 *Resistance.* The transfer of heat produced by the temperature head is opposed by a total resistance R which for analysis divides conveniently into the resistance R_v on the vapor or steam

side of the surface, the resistance R_m of the metal walls of the surface, and the resistance R_w on the cooling water side of the surface.

17 A simple equation expressing heat transfer in useful terms may be written as follows:

$$R = \frac{M}{H} \qquad [3]$$

in which

H = number of heat units transferred per unit time
M = mean temperature difference
R = total resistance = $R_v + R_m + R_w$

18 Even with high steam pressures and with superheat, the total B.t.u. to be extracted by the condenser may safely be assumed as 1000, and it is convenient to adopt an arbitrary resistance unit such that

$$H = \frac{1000 \times M}{R} \qquad [4]$$

or

$$W = \frac{M}{R} \qquad [5]$$

in which

H = B.t.u. per sq. ft. per hr.
M = mean temperature difference in deg. fahr.
R = resistance per sq. ft. of surface
W = pounds steam condensed per sq. ft. per hour.

The symbol U will be used when M is unity, so that U = B.t.u. per sq. ft. per hour per deg. mean temperature difference. This resistance R may also be expressed in terms of equivalent conductivity by the equation

$$R = \frac{1000 \times L}{C \times 4290} \qquad [6]$$

in which

L = thickness of substance in inches
C = conductivity in c.g.s. units.

We will now consider separately each of these resistances.

19 R_m, *the resistance of the metallic walls* of the tube, is simple to determine, for the passage of heat is solely by conduction. The conductivity of brass may be taken as 0.26 (c.g.s. units) so that we may at once write from equation [6]

$$R_m = \frac{1000 \times L}{0.26 \times 4290}$$

which, for a No. 18 S.W.G. tube having a thickness of 0.049, becomes 0.044. This R_m, as will appear later, is only a very small part of the total resistance R which commercially ranges between 2 and 3. If, however, we consider as part of the tube any solids adhering to it, then this R_m may become relatively very large. Ordinary lime or magnesium scale has a conductivity of not greater than 0.004 (c.g.s. units) so that a deposit of this 0.01 in. thick will by equation [4] increase R_m by $\frac{1000 \times 0.01}{0.004 \times 4290} = 0.582$, or more than 1000 per cent.

20 Oil has even a lower conductivity, say conservatively 0.002, so that a deposit 0.005 in. thick will increase R_m by 0.582, or more than 1000 per cent. A slight coating due to oxidization on apparently clean old tubes many increase R_m several hundred per cent. These figures, although difficult to believe, are approximately correct and indicate the absolute necessity of incorporating in condenser design features which will insure the maintenance of clean cooling surfaces.

21 R_v, *the resistance on the vapor side* of the surface, has been determined by Prof. Callendar[1] as 0.333 for air-free steam and is probably not much greater in commercial condensers properly designed so as to produce a uniform flow of steam.

22 Since the particles of steam are rapidly moving in toward the tube where they are condensed and are replaced by other particles, it follows that the conductivity of steam is not an important factor in the problem and that R_v for air-free steam is due mostly to the film of condensate adhering to the tube.

23 The conductivity of water is very low, being only 0.0014, so that by equation [6] for a film of water 0.01 in. thick

$$R_v = \frac{1000 \times 0.01}{0.0014 \times 4290} = 1.67$$

Actually the particles of the film of water are in rapid motion, due to the movement of the steam, so that, if a fairly high steam velocity is maintained, the film itself is very thin.

24 The real factor which increases the value of R_v in most commercial condensers is the considerable quantity of air which is present and which is carried to the tube surface by the movement of the steam toward the tube during the process of condensation. As the conductivity of air is only about 0.00005 it is apparent that unless the design of the steamway provides for the rapid clearing of

[1] Encyclopædia Britannica.

the air from the surfaces and its movement to the point of removal from the condenser, R_v will be enormously increased. In poor designs it may be as much as from two to four times greater than necessary. To secure rapid clearing of the surfaces, the steamway should provide for a uniformly high steam velocity throughout the condenser and should be of such a shape that all the air will be swept ahead by the steam flow.

25 R_w, *the resistance on the water side of the tube*, comprises, within commercial limits of water velocity, the largest part of the total resistance R and its values can be readily determined by deducting R_v and R_m from R.

26 The writer believes that the results of the tests by Orrok[1]

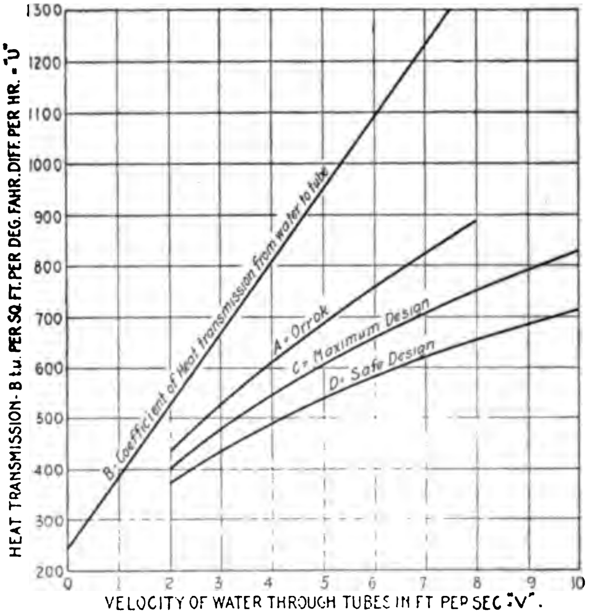

FIG. 1 HEAT TRANSMISSION—VELOCITY CURVES

provide the most reliable data available on heat transfer through condenser tubes. In Fig. 1 (curve A) is shown the curve given by Orrok to represent these results. In Fig. 2 (curve A) are plotted total resistances R obtained by applying the values from Orrok's

[1]Transmission of Heat in Surface Condensation, Geo. A. Orrok, Trans. Am. Soc. M. E., Vol. 32, page 1139.

curve in Fig. 1 to equation [4], in which M is taken as unity. For convenience, these resistances are plotted against reciprocal velocity instead of against velocity.

27 It has previously been shown that for ideal conditions $R_v = 0.333$ and that for a No. 18 gage brass tube $R_m = 0.044$, making $R_v + R_m = 0.333 + 0.044 = 0.377$, and it is reasonable to assume that Orrok's tests approach these conditions sufficiently closely so that a reasonable value to accept for $R_v + R_m$ for his tests is 0.4. On this assumption curve B, Fig. 2, is plotted showing the relation of R_w to the reciprocal velocity.

28 A curve, Fig. 1 (B), plotted from the values on curve B,

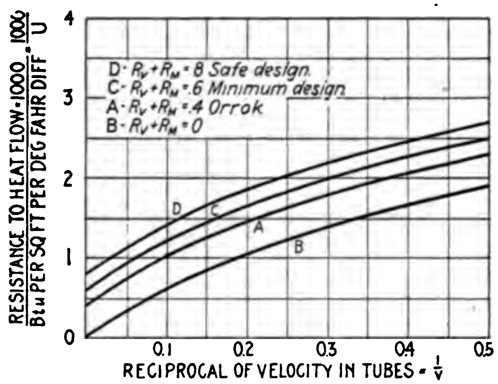

FIG. 2 RESISTANCE—RECIPROCAL VELOCITY CURVES

Fig. 2, represents the relation of heat transfer from the surface of a condenser tube to velocity of the water in contact with that surface. It is very pleasing to find that as was expected this is a straight line for all values of V between 1 and the upper limit 7. From this curve U_w varies directly with V according to the equation

$$U_w = 245 + 141\ V \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

29 This variation of resistance, inversely with velocity, is due to the fact that the particles of water in contact with the surface at any instant form a non-conductor which prevents the flow of heat from particles in the body of the water to the surface of the tube. The transfer of heat is really by convection, and the more rapid the removal of the heated particles and their replacement by cooler ones, the greater the heat transfer.

30 With the same velocity this transfer of particles is much

more rapid in a small tube than in a large one, where, so to speak, a cold core of water exists. This indicates the desirability of small tubes and experience dictates 3/4 in. to 7/8 in. inside diameter as a maximum. Probably the straight line representing the relation of

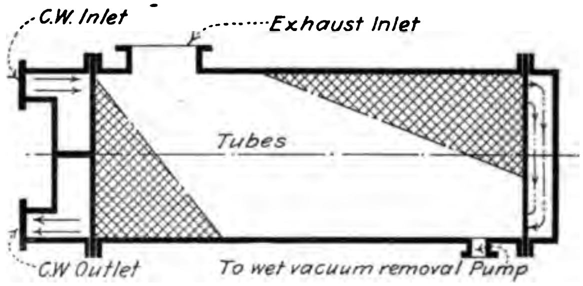

FIG. 3 CONDENSER ILLUSTRATING INCORRECT PARALLEL FLOW, INCORRECT WATER CONNECTIONS, AND NARROW WATER CHANNELS

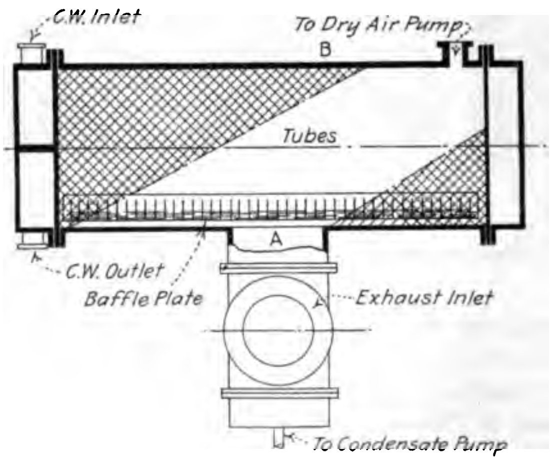

FIG. 4 CONDENSER SHOWING EFFECT OF EXCESSIVE BAFFLING, AND IMPROPERLY LOCATED AIR CONNECTIONS

U_w to V will be higher up and steeper for very small tubes than that shown in Fig. 1.

31 It is apparent that any material increase in the viscosity of the cooling water would retard the rapid exchange of particles at the surface, thereby increasing R_w, and some investigators have developed elaborate formulas purporting to express this relationship. The viscosity of water varies so slightly within the ordinary

range of cooling water temperatures for which condensers are designed, say 60 deg. to 80 deg. fahr., that it appears unnecessary to give this factor consideration.

PRACTICE

32 *Mean temperature difference*, as previously proven, requires for a maximum that the surfaces be arranged for counter-current flow, the water entering farthest away from the steam and passing consecutively through groups of tubes so as finally to pass out through the entering steam. This calls for multi-pass construction.

33 Among metals commercially available for use in condenser tubes, copper has the highest conductivity and furthermore, when

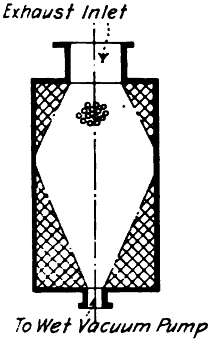

Fig. 5

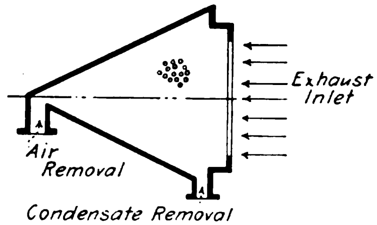

Fig. 6

Fig. 5 High Rectangular Condenser
Fig. 6 Theoretically-Correct Condenser Shape

properly alloyed, is less subject to corrosion than most others, thus permitting the use of thinner surfaces. Hence practically all condenser tubes are copper or high percentage copper alloy.

34 The size of tube is a determining factor in the thickness, larger tubes requiring greater thickness for mechanical strength, and from this viewpoint also small tubes are desirable.

35 Since a deposit of scale 0.01 in. thick may increase R_m by, say, 0.582 or 30 per cent of the total R, the minimizing of possibility of scaling is very important. High circulating water velocity will accomplish this and is a more important reason for using small tubes and several passes than is generally recognized. The arrangement of heating surfaces for easy cleaning and the construction of water channel covers independent of pipe connections is important, although frequently neglected.

36 R_v, *the resistance on the steam side* of the tube, appears to vary but little from, say 0.333, when the steam is practically air-free, probably due mainly to the film of condensate. There will be more variation, however, if the condenser is exceedingly high, as in Fig. 5, so that a large quantity of condensate falls over the lower tubes. Attempts to drain groups of tubes by trays (so-called dry-tube condensers, Figs. 7, 8, and 13) introduce resistances to steam flow which more than offset any gain from draining.

37 Only recently has the enormous effect of air on R_v been recognized. Investigation of condensers of the original round type proves that only those tubes which are near the exhaust are fully effective, owing to the stagnating of the air around the tubes and the average value for R_v is as high as unity or greater.

38 An exhaust opening of liberal size with a dome extending the length of the shell, Figs. 11 and 14, will cause the steam to be distributed to the ends of the tubes and prevent stagnant corners such as represented by the shaded portions in Figs. 3, 4, and 5.

39 Baffle plates for directing the steam to remote parts of the condenser introduce resistance to steam flow and should be avoided, except for the small plate directly in front of the exhaust inlet to protect the tubes from entrained water in the exhaust. The condenser, Fig. 4 (later rebuilt by the writer), with a baffle plate having only 13 per cent opening showed the differences in vacuum of 2½ in. between points A and B.

40 The steam passing over the tubes condenses and diminishes in volume as it progresses, and hence in ordinary condensers the steam flow velocity decreases and becomes practically nil in the portion away from the inlet, permitting air to stagnate and render ineffective the shaded portions, Figs. 3, 4, 5, the conditions being worst in Figs. 3 and 4 where the shaded portions are entirely out of the steam stream and remote from the air removal connection.

41 This steam flow velocity may be maintained by constructing a gradually reducing steamway, a triangle with steam entering over one entire side, Fig. 6, being theoretically correct, or by gradually reducing the pitch of the tubes or by making lanes or passages to various parts of the steam space by omitting tubes, Figs. 11 and 14. Any one of these methods properly applied should be effective and result in good steam distribution at uniform velocities, prevent the stagnation of air at any point, and minimize the frictional drop. When a condenser is incorporated in the base of a turbine, these lanes should start from opposite the buckets as nearly as possible.

42 Fig. 10 shows a commercial form of taper passage condenser of English type, which, however, has a shell of a shape that is somewhat inconvenient to construct. Fig. 9 shows similar taper passages embodied in a round shell, but the large heavy baffle plates which occupy the available tube space are objectionable. Fig. 7

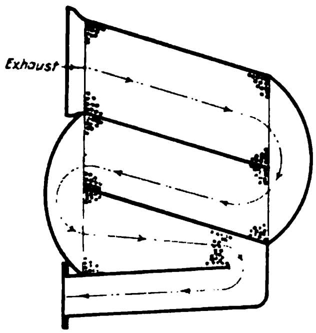

FIG. 7

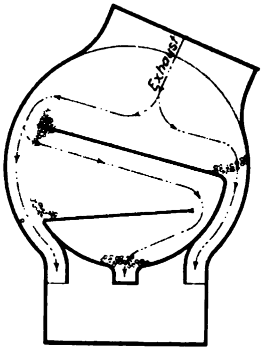

FIG. 8

FIG. 7 TAPER PASSAGE CONDENSER, ILLUSTRATING RESTRICTED STEAMWAY AND UNCOMMERCIAL SHAPED SHELL

FIG. 8 TAPER PASSAGE CONDENSER, ILLUSTRATING RESTRICTED STEAMWAY

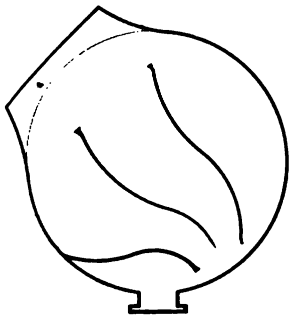

FIG. 9

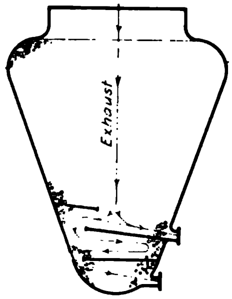

FIG. 10

FIG. 9 ROUND SHELL CONDENSER BAFFLED TO HAVE THREE TAPER PASSAGES

FIG. 10 MODERN ENGLISH TAPER SHELL CONDENSER

shows this feature also, but the shape of the shell and the resistance to steam flow make it an unpractical construction. The design in Fig. 8 likewise has too many baffles which increase the weight and obstruct the steam flow. The design in Fig. 14 approaches that of the others in principle, but the shape of the water passages is objectionable.

43 If a liberal pitch be employed for the tubes, and ample lanes be provided, the frictional drop, even through a large condenser need not exceed 0.4 in. and less in smaller ones. Figs. 11, 13 and 14 show proper distribution, but Figs. 4, 7, 8, and 12, having long steamways and closely pitched tubes, may have frictional drops as great as 2 in.

44 It is important that a sufficient number of air removal connections be located at points where air accumulates (Fig. 11), or the air will stagnate and render certain portions ineffective, as shown by the shaded areas in Figs. 3 and 4.

45 The quantity of air allowed to enter a condenser should at all times be minimized and the importance of tight joints and pipe connections should be impressed upon operators.

46 As only a very small quantity of air can enter with the feed water, it is evident that proper operating attention to the tightness of condenser shell, low-pressure stages of the turbine, piping and valves, will reduce the quantity of air in the condenser to a very low figure.

47 In order to determine the completeness of the distribution of the steam to all parts of the steam space, it is very instructive to obtain circulating water temperature readings at two or three points in the water box between the first and second passes, and corresponding steam temperature readings. From these it is simple to compute the coefficient of heat transmission U for each pass. The writer has computed these for two large condensers and has found U for the first pass only about 10 per cent less than for the second, indicating that with proper steam distribution and effective air removal the value of U will be about the same for all parts of the surface. If, however, steam distribution is poor and the clearing away from the surfaces and the removal of the air unsatisfactory, the value of U for the first pass may be only 50 per cent of that in the second. Likewise, if the tubes away from the center are not doing their proportion of work, this will be indicated by the temperature readings taken on the water flowing from them.

48 R_w, the resistance on the water side of the tube, would be negligibly small if we could use very high water velocities through exceedingly small tubes. This, however, would cause a rapid increase of frictional resistance and consequent cost of pumping, erosion of the tubes if the water contained sand, and an undesirable number of passes or a very long condenser. For these reasons a velocity

flow is ordinarily limited to about 4 to 6 ft. per second, and R_w under favorable conditions is, say 1.0, or the largest of the three resistances. Frictional loss may be minimized by using long tubes and fewer passes, reducing water passage and tube entrance loss.

49 Even distribution of water through all tubes is important and narrow channels causing high velocities, Figs. 3 and 4, or inlets

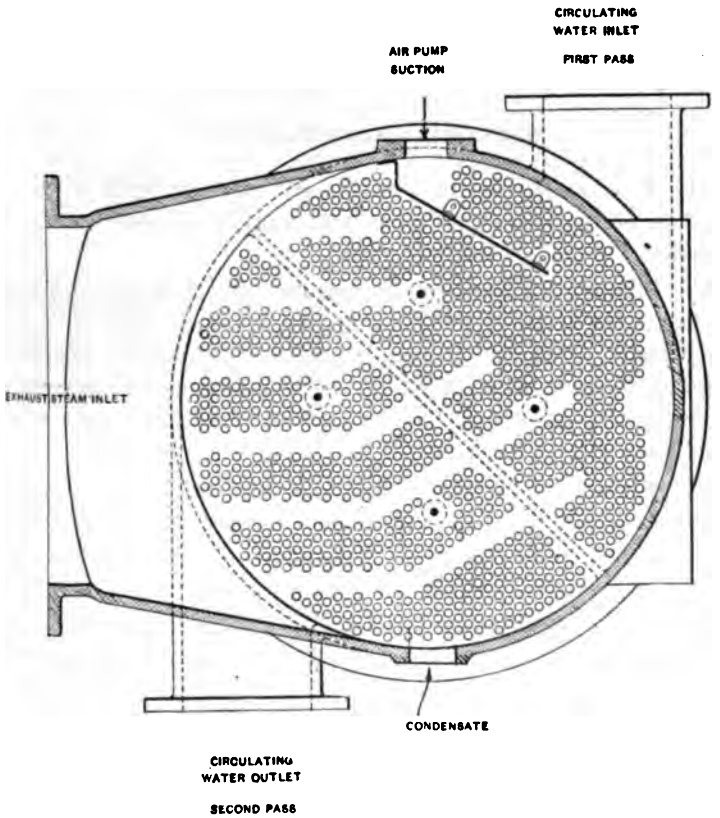

FIG. 11 WELL-DESIGNED CYLINDRICAL CONDENSER

directing water onto the tubes, Fig. 3, must be avoided, since uneven distribution is sure to result, those tubes not in the stream line receiving little water and being therefore largely ineffective.

50 R, the total resistance which includes R_v, R_m and R_w is applicable to commercial designs, as shown by curve C, Fig. 2, arrived at by accepting a coëfficient of heat transmission U, of say

550, for a water velocity of 4 ft. per sec. This coefficient is about the best obtainable in practice and there are considerable data available for this water velocity. By comparing this with the value on curve B, Fig. 1, the value of $R_v + R_m$ is determined as say 0.6, which is used in plotting the points on curve C, Fig. 2. Perhaps in most cases a value of 0.8 for $R_v + R_m$, giving curve D, would be safer to use for design. If we accept four as a desirable water velocity we obtain from curve D, Fig. 2, $R = 2$, corresponding to an allowable value for U of say 500 B.t.u.

FINAL FLUID TEMPERATURES

51 It is instructive to consider our problem with reference to desirable temperature conditions and method of removal from the condenser for each of the four fluids.

52 Steam must be maintained at the lowest practicable pressure, and hence the temperature must approach closely that of the circulating water discharge. There must be a difference, however, in order to produce heat flow, but in well-designed counter-current apparatus this difference may be kept within 10 deg. fahr. without objectionably large apparatus.

53 Air must be removed from the condenser by a mechanical pump, the energy required for operation being directly proportional to the volume of air moved. This volume should be minimized by causing the air finally to pass over the coldest tubes, Figs. 4, 11, 13 and 14, and thus to approach the circulation water temperature.

54 Since air pumps must have small clearances to operate efficiently, it is desirable that the air leave the condenser free from entrained water. This can be accomplished by causing it to flow over tubes protected from falling condensate, Fig. 11, and not mingling with the warm condensate as in Figs. 3, 5, 9 and 10.

55 With these points observed, air pump sizes may be quite small, and the energy required for operation only, say one per cent, of the main engine. Manifestly the old, and even now common marine practice of removing air and condensate with a single pump fulfills not one of the desired conditions, and often requires as much as 10 per cent of the steam required for the main engine.

56 Furthermore, actual tests have proven that for ordinary wet vacuum pumps to handle the mixture of air, vapor, and water and maintain even moderately high vacuums it is necessary to cool the

condensate 10 to 15 deg. below that due to the vacuum, which of course requires more circulating water and wastes more heat from the system.

57 Another serious objection to the wet vacuum system is that compressing an emulsion of air and water is a most effective method of mixing the air with the condensate to return to boilers.

58 *Condensate should be removed at a high temperature,* thus minimizing the work of the feed heater and boiler. This temperature can be maintained within 1 deg. of the vacuum temperature,

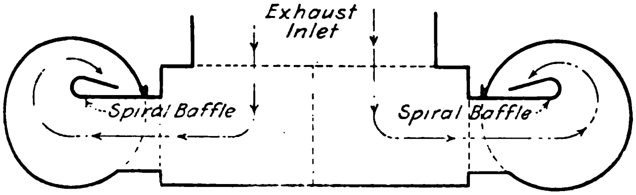

FIG. 12 BASE AND DOUBLE WING CONDENSER, ILLUSTRATING EXCESSIVELY LONG STEAMWAY

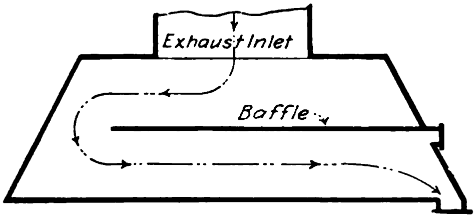

FIG. 13 BASE CONDENSER WITH TAPER PASSAGE

by arranging so that the condensate will fall finally over those tubes containing the hottest circulating water and through the entering steam, Figs. 4, 11 and 14. It will be 20 to 40 deg. below the vacuum temperature if the condensate be permitted to fall finally over tubes containing the coolest circulating water, Figs. 5, 8, 9, and 10. Especially in large condensers, a number of condensate removal connections should be provided on the shell to insure free and quick flow to the removal pump, generally a centrifugal which, unlike a plunger pump, will handle varying quantities without speed changes, and which if properly vented never becomes vapor-bound. With the wet vacuum system the air pump cannot operate without vapor binding unless the condensate temperature is 15 to 30 deg. fahr. below the steam temperature, depending on the vacuum.

59 *Circulating water*, to reach minimum quantity, must have an exit temperature closely approaching the steam temperature. The great effect of a comparatively small variation in this temperature may be appreciated by considering the maintenance of a 29 in. vacuum (79 deg. fahr.) with circulating water at 60 deg. fahr., the quantity required being twice as much if heated to within 14 deg. of the steam temperature, than if heated to within 9 deg., and since friction head increases approximately as the square of the velocity, the energy required to pump the circulating water increases as the cube of the quantity, or in this case 8 times.

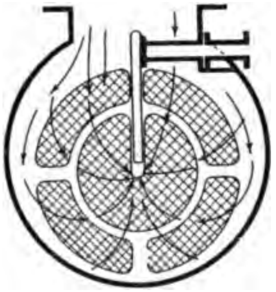

FIG. 14 CONDENSER GIVING TAPER PASSAGE EFFECT WITHOUT BAFFLES

60 The optimum value for this minimum temperature difference is always a compromise between condenser cost and pump and pumping cost but with well designed apparatus should not exceed 10 deg. fahr. With poor designs, especially those having parallel flow, this difference is sure to be 15 to 20 deg. fahr. For service with circulating water obtained from cooling towers or other expensive source it might be warrantable to have this difference as low as 5 deg. fahr.

PIPING

61 The connections between and relative location of condensers and auxiliaries are important factors in condenser efficiency, but in most cases are beyond the control of the manufacturer and consequently are neglected and are a common source of condenser trouble.

62 The exhaust pipe must be large enough to carry without appreciable pressure loss the steam from engine to condenser, and experience has determined a velocity of say 500 ft. per sec. as a

maximum. To reduce air leak possibilities it should be short and without joints, consisting preferably only of an expansion piece, the condenser being designed so that the exhaust inlet can be placed opposite the engine exhaust. Nearly all marine installations have small and long exhaust pipes and show, on examination, pressure losses up to 3 in. between condenser and engine. An inspection of the pressure volume curve for steam will show the rapid increase of volume at the higher vacuums, the volume at 28½ in. vacuum being 17 times that at atmospheric pressure.

63 Condensate piping must be large, figured say for 1 ft. per sec. velocity, and must be straight down to the pump with no chance for vapor pockets. The pump should be, say 4 ft. below the condenser, and have a pressure equalizing pipe between the suction at pump and the condenser, otherwise vapor binding is apt to occur.

64 The air pump piping should be amply large and should be arranged to eliminate the possibility of condensate passing over into the air pump which has small clearances. Ordinarily, however, air piping is much larger than necessary.

RATINGS

65 The surface condenser, like most other apparatus, is subject to irrational, meaningless, and misleading ratings, the most objectionable being square feet per engine horse power, on account of the wide variation in the amount of steam required per engine horse power, say 9 to 25 lb. per hour.

66 A comprehensive rating must include the following:

a Quantity of steam condensed.
b Vacuum obtainable (corrected to 30 in. barometer)
c Temperature of available cooling water.
d Cooling water exit temperature
e Condition of air at point of removal.
f Friction head on cooling water.
g Temperature of condensate at point of removal.

The first four items express the heat transmitting efficiency and can for purposes of comparison be reduced to B.t.u. per sq. ft. per deg. difference per hour.

67 Since a higher vacuum at the air pump than at the exhaust inlet is of no value, the mean temperature difference used for comparing results on condensers should be computed on the assumption that the steam temperature throughout the condenser is that due

to the vacuum at the exhaust inlet and the equation [2] should be used.

68 Fig. 15 is a diagram for the rapid determination of mean temperature difference values. It is generally assumed that the transmission of heat by steam, gas or liquid through metal divisions

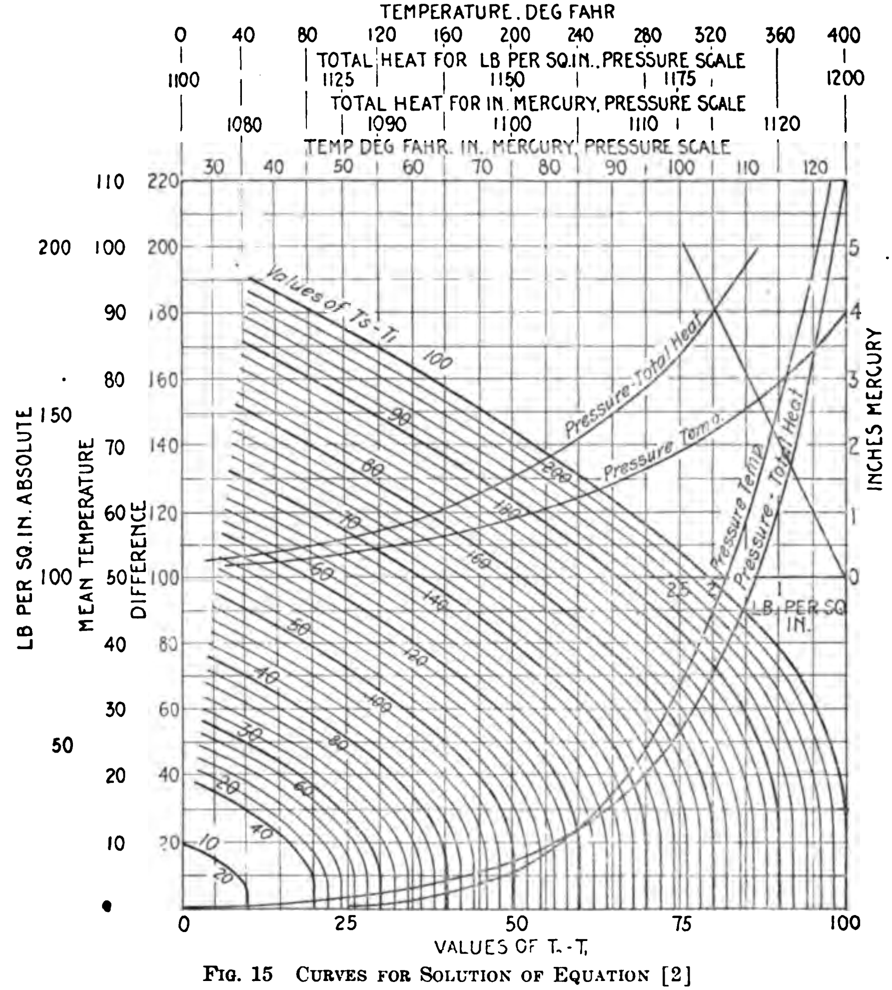

FIG. 15 CURVES FOR SOLUTION OF EQUATION [2]

is proportional to the difference in temperature between the substances. But generally the temperature of each of the substances is not the same at all parts of the surface. In the case of feed water heaters, condensers, etc., the temperature of the steam may be taken as constant, but the temperature of the water decreases or increases as the case may be in its passage over the surface. For purposes of computation a mean temperature difference value must be determined. Probably the most reliable formula for determining this value is that of Grashof which when the temperature of one of the mediums is constant reduces to

$$D = \frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}}$$

where D equals the mean temperature difference, T_1 equals the lowest temperature of the fluid, T_2 equals the highest temperature of the fluid, T_s equals the temperature of the gas. The result will not be changed if any constant be deducted from all of the T's in the equation. By making this constant T the equation is simplified to one of three variables, which are plotted on this diagram, Fig. 15. To increase the accuracy or range of the diagram two scales are given. For temperatures between 200 deg. and 1000 deg. use the 100 scale. Five curves showing pressure-heat-temperature relations of steam are produced on the same set for convenient reference.

69 The complete equation for the condenser is

$$S = \frac{W \times Q}{M \times U} \quad \text{or} \quad U = \frac{W \times Q}{M \times S} \qquad [8]$$

in which

U = B.t.u. per sq. ft. per deg. fahr. difference per hour = coefficient of heat transmission
M = Mean temperature difference in deg. fahr.
W = Pounds steam condensed per hour.
S = Square feet of cooling surface.
Q = Total heat removed by circulating water per pound steam condensed (usually taken as 1000 in all cases for simplicity).

Thus for example, assuming results as follows from two condensers, we can say that A is 50 per cent more efficient than B.

	A	B
Square feet surface	2000	5500
Pounds steam condensed per hour	18500	57200
Vacuum	28½ in. (1½ in. abs.)	27
Temperature entering cooling water	65	60
Temperature exit cooling water	80	100
Mean temperature difference per equation [1]	18.3	30.9
B.t.u. per sq. ft. per deg. difference per hour	505	337

70 Items 3 to 6 determine the mechanical energy required by

air condensate and circulating pumps, which must for best plant economy be kept low so that the exhaust steam from all auxiliaries will not exceed the amount which can be condensed in the feed heater. Even for high vacuum the steam required by air, condensate, and circulating pump need not exceed say 7 per cent of that of the main engine.

71 Item 7 indicates the heat efficiency being a measure of the heat returned to the system in the condensate.

The writer acknowledges with thanks the valuable assistance of Mr. P. E. Reynolds who obtained some of the necessary test data, checked computations, and offered suggestions.

DISCUSSION

P. E. Reynolds (written). A few years back it appeared to be the main idea of the designer to crowd as much cooling surface as possible into the least space, regardless of accessibility of the surface to the steam, or the friction loss entailed in bringing the steam to the surface.

There is no doubt that large exhaust openings, combined with steam distributing domes of ample dimensions and steam lanes or passages through the tubes, are some of the main features of efficient surface condensers designed to maintain high vacua.

Regarding the variations of heat transfer with water velocity, although Mr. Braun may be theoretically correct in stating that Mr. Orrok's exponential formula is fundamentally wrong, since the resistance to heat transfer of the tube itself and that on the steam side of the tube are constant, yet it seems that since these resistances are constant it is a useless complication for all practical conditions of condenser designs to take them into consideration.

Regarding parallel flow and counter-current condensers, I agree with Mr. Braun that the latter give the best results; however, Mr. Braun's proof of the fact by means of formula [1]. I do not consider correct. It is my understanding that in the mathematics involved in the derivation of this formula, the assumption is made that the heat absorbed by the cooler fluid results in a corresponding decrease in temperature of the hotter fluid. As these conditions do not prevail in a steam condenser, since the steam temperature is not decreased by the abstraction of heat at constant pressure, I would not consider that this formula could be correctly applied.

Formula [2], based on the assumption that the steam temperature $T\ S$ is constant throughout the condenser, is the only one that can be correctly used. However, it appears that too much weight should not be attached to the mean temperature as given by this formula, since the assumptions on which its mathematics is based are not fulfilled in actual surface condensers, and if a true mean temperature difference between the cooling water and steam is desired, it is probable that the arithmetical mean is as nearly correct as any.

H. Wade Hibbard mentioned that in the author's reference to the exhaust pipe between the engine and condenser he would suggest adding that in some installations it is desirable to use a steam separator to remove the water from the exhaust steam before it goes to the condenser.

The Author. Referring to the remarks of Professor Hibbard, who suggests that it may be desirable to have a steam separator between the prime mover and the condenser, I will assume that he thinks it advisable to remove the water from the exhaust steam so as to reduce the coating of condensate which will adhere to the condenser tubes. This has been tried many times, and very elaborately by one or two manufacturers, that is, by the installing of drain plates, which might be called separators, in the condenser; and it has been invariably found that these offer resistance to the flow of steam, besides complicating and increasing the cost of the design, which more than offsets any possible value that they might have in increasing the heat transfer by reason of decreasing the resistance on the steam side of the tube, and I feel sure that this is now an established fact.

Replying to Mr. Reynolds' remarks, I must point out some obvious fallacies.

His statement that the resistance R_v on the steam side of the tube, and the resistance R_m of the tube are, in condensers, nearly constant, does not agree with facts.

Actually, R_v varies greatly with varying amounts of air present, as is plainly apparent when one considers the marked effect upon vacuum produced by even the most minute of air leaks. Furthermore, repeated tests on two-pass condensers, from which the performances of each half the condenser has been computed separately, have invariably shown that the coefficient of heat transfer is less in that half which contains the air outlets.

R_m, which properly includes the resistance of *any solids adhering to the tube*, increases greatly, as we all well know, when the condenser becomes foul with scale or oil, and the value to be given it should depend upon the quality of the circulating water, the presence or absence of oil in the exhaust and the continuity of service required.

Mr. Reynolds' limitation of the logarithmic mean temperature difference formula is also incorrect. The only assumption involved in the mathematical derivation is the proportionality of heat transmitted to the first power of the temperature difference. This proportionality is not absolutely true in a commercial condenser, due to the presence of air, but it is certainly more desirable to start with rational and the theoretically correct formula, making allowances in practical design for known influencing factors, than to revert to rule of thumb methods and accept formulæ such as the arithmetical mean for temperature difference which we know to be fundamentally wrong. To fulfill Mr. Reynolds' conditions for the correctness of formula [1] it is only necessary to consider the steam as a fluid having an infinitely large specific heat.

No. 1485

DESIGN OF RECTANGULAR CONCRETE BEAMS

By Howard Harding, Rochester, N. Y.

Associate-Member of the Society

The resisting moment of a reinforced concrete beam, in inch-pounds, may be represented by the formula

$$\text{Resisting moment} = R\,b\,d^2$$

where

b is the breadth of the beam in inches

d is the effective depth in inches

R is a numerical coefficient

The value of R (for a given ratio of the modulus of elasticity of steel to that of concrete) depends upon the percentage of steel reinforcement used and the safe working stresses for steel and concrete. These values of R for different working stresses and percentages of steel have been plotted in very convenient curve form in Turneaure and Maurer's "Principles of Reinforced Concrete." For the purposes of this article it will be necessary to cite only a few values of R corresponding to some of the more common working stresses. The ratio of E_s to E_c is taken as 15.

For $f_c = 700$ and $f_s = 16{,}000$	$R = 120$	% steel = 0.87
For $f_c = 600$ and $f_s = 16{,}000$	$R = 94$	% steel = 0.68
For $f_c = 500$ and $f_s = 16{,}000$	$R = 71$	% steel = 0.50
For $f_c = 400$ and $f_s = 16{,}000$	$R = 49$	% steel = 0.34

2 In case of reinforcement in excess of the amount required to divide the working stresses as shown there is a slight gain in strength due to the shifting of the neutral axis toward the steel. When the concrete has reached its working stress the steel will still be understressed. In such cases the value of R used must be the one corre-

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

sponding to the working stress of concrete and the percentage of reinforcement. Reference to the diagrams given in Turneaure and Maurer's book will make the meaning of the foregoing evident (Fig. 1).

3 The ordinary method of using the data thus given is that of "cut and try." Suppose it is desired to design a beam to withstand a bending moment of 120,000 in-lb., by use of the formula Resisting moment $= R\,b\,d^2$. If $f_s = 16{,}000$; $f_c = 600$; and the reinforcing steel $= 0.68$ per cent, then

$$120{,}000 = 94\ b\ d^2$$

Try $b = 4$ in.
Then

$$d^2 = \frac{120{,}000}{94 \times 4} = 319$$

$$d = \sqrt{319} = 17.8 \text{ in.}$$

The ratio of breadth to depth is not satisfactory, so
Try $b = 8$ in.
Then

$$d^2 = \frac{120{,}000}{94 \times 8} = 159$$

$$d = \sqrt{159} = 12.6 \text{ in.}$$

The ratio of b to d is now satisfactory, but to arrive at the final dimensions some allowance must be made for the dead weight of the beam and such allowance will necessarily depend largely upon the span for which the beam is to be used.

4 The solution as indicated is very indirect and roundabout. After quite a bit of experience with that method the growing dissatisfaction prompted the plotting of the equations involved so as to give a direct method of arriving at the result desired. After some study a logarithmic form was adopted which seems to serve the purpose.

5 On a sheet of logarithmic cross-section paper the values of b were denoted by the horizontal lines intersecting the left-hand ordinate. The left-hand ordinate was called the b scale. Through the origin (that is, through the point 1, 1 of the diagram) draw a line to the right at 45 deg. to the horizontal. By inspection of Fig. 2 it can readily be seen that the point of intersection of any value b with the oblique line, if projected vertically upward to the top horizontal scale of the paper, will give a logarithmic reading exactly equivalent to that of the b scale. Now let the oblique line correspond to $d = 10$ and multiply the readings of the top horizontal scale by 100. Then again

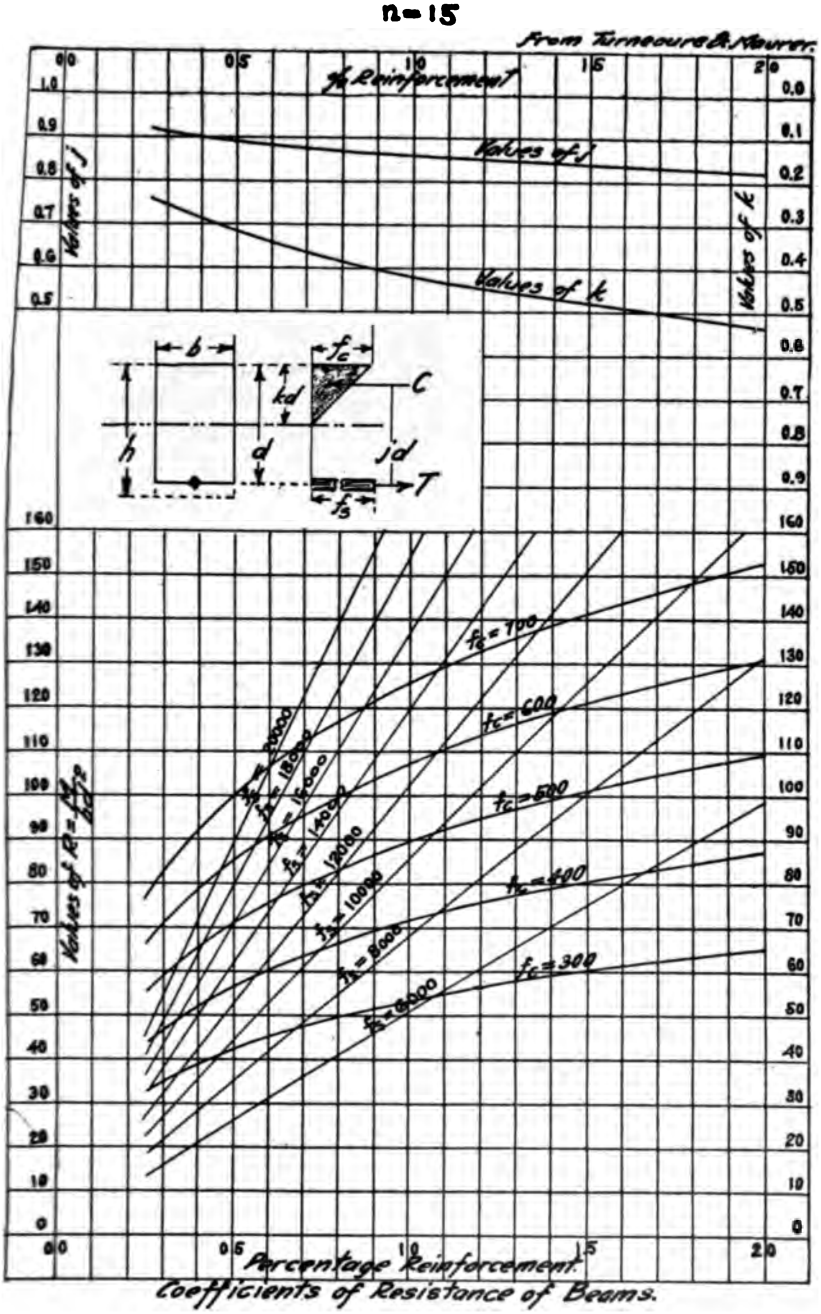

Fig. 1 Diagram from Turneaure and Maurer's Book Showing Relation of Working Stress in Concrete and Percentage of Reinforcement

by inspection, it will be seen that by projecting vertically upward the intersection of any value b and the line $d = 10$, we may (on the multiplied horizontal scale) read directly the corresponding value of $b\,d^2$.

6 For values of d other than 10 there is a family of lines parallel to the line $d = 10$. If we let b_y be the intersection of any d line with the b scale, then we may write the following equation

$$b_y\,d^2 = 100, \text{ or } b_y = \frac{100}{d^2}$$

For $d = 5$

$$b_y = \frac{100}{5^2} = 4$$

The relations indicated by the equation may be easily checked by reference to Fig. 2. By solving the equation for b_y with different values of d we are able to plot on the diagram the corresponding

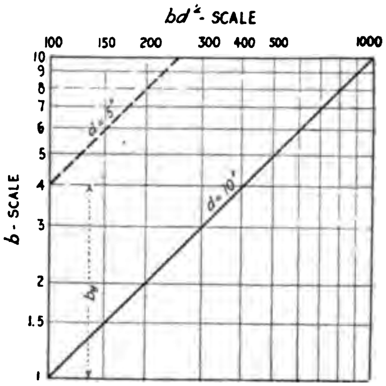

FIG. 2 DIAGRAM FOR GRAPHICALLY DETERMINING bd^2

family of d lines. At this stage we have a diagram from which we can read directly any value of $b\,d^2$ corresponding to given values of b and d; or given $b\,d^2$ and either b or d we may determine the corresponding d or b.

7 It now remains to introduce the factor R into the diagram. A range of from 40 to 120 for this coefficient will be found to cover about all cases likely to arise. Since the bending moment is ordinarily computed in foot pounds we may write our fundamental formula to correspond to those units.

Then

$$\text{Resisting moment} = R\frac{b\,d^2}{12}$$

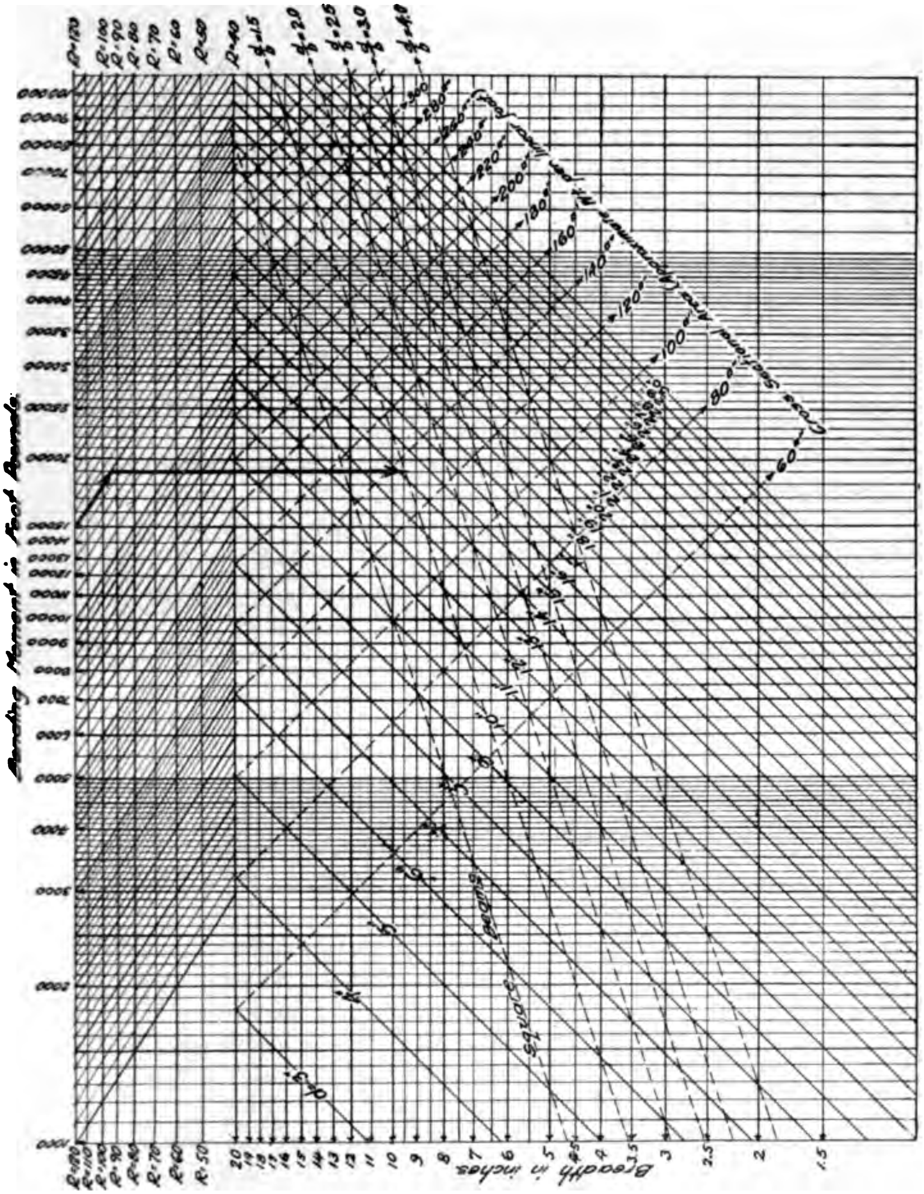

FIG. 3 DIAGRAM FOR GRAPHICALLY DETERMINING THE RESISTING MOMENTS OF REINFORCED CONCRETE BEAMS

When $R = 120$

$$\text{Resisting moment} = 10\ b\ d^2$$

By projecting the values of $b\ d^2$ upward to a horizontal scale, Fig. 3, whose readings are 10 times that of the $b\ d^2$ scale, we obtain directly the resisting moment in foot-pounds corresponding to any value of $b\ d^2$ and $R = 120$. The diagram is arranged to care for other values of R by taking the proper fractional part of the resisting moment of the beam at $R = 120$. Thus, for $R = 60$, one-half of the resisting moment of the beam at $R = 120$ is taken. This is done graphically by proceeding upward to the horizontal line representing the desired value of R and thence proceeding along the oblique line to the top scale where the resisting moment in foot-pounds is read.

8 In the lower part of the diagram, Fig. 3, it will be noticed that there are two other families of oblique lines. One of these shows the ratio of d to b and the other the cross-sectional area or the product of d times b.

9 The method of using the diagram is ordinarily just the reverse of the order in which it has been developed. First decide upon the working stresses and determine the corresponding value of R and the percentage of reinforcement. Then compute the bending moment in foot-pounds which it is desired that the beam should withstand. Locate the bending moment thus computed at the top of the diagram, Fig. 3. Thence proceed parallel to the oblique lines until the value of R previously determined is intersected. From this intersection proceed vertically downward into the lower part of the diagram intersecting the horizontal lines representing breadth, and the oblique lines representing depth. At any one of these intersections the required breadth and depth may be read. Any of the corresponding values will satisfy the condition for strength, but the values chosen will depend upon the shape of beam desired. If we wish a certain ratio of depth to breadth we can obtain it by stopping at the dotted line representing that ratio. If a certain depth or a certain breadth is desired it is necessary to proceed vertically downward until said depth or breadth is intersected and the point of intersection automatically determines the value of the other dimension.

10 In a beam of dimensions as determined by the use of the diagram in Fig. 3 no allowance has been made for the bending moment on the beam due to its own dead weight. To take care of this additional bending moment either one or both of the dimensions must be increased. To determine the amount of increase necessary for different

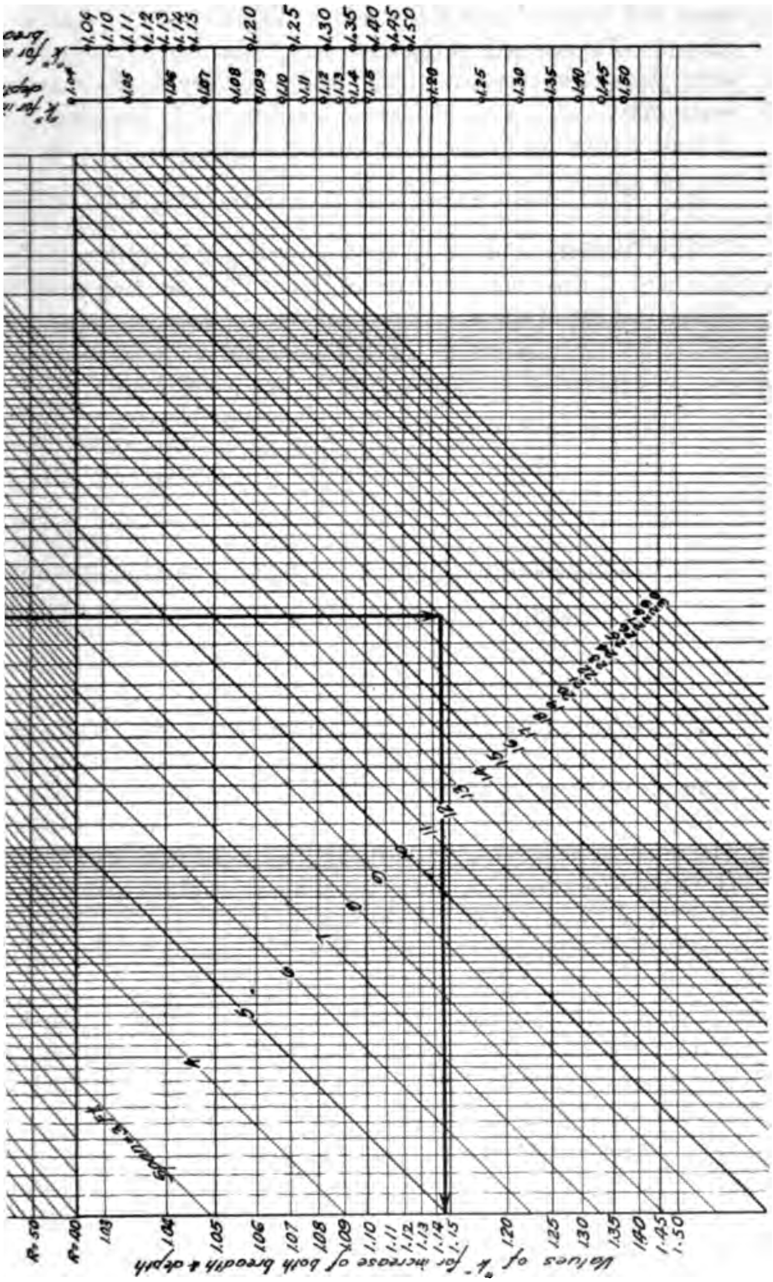

FIG. 4 DIAGRAM TO INDICATE ALLOWANCES TO BE MADE IN BENDING MOMENTS OF BEAMS DUE TO THEIR DEAD WEIGHT

spans and types of beams the second diagram, Fig. 4, has been developed. The strength of the beam may be increased in three different ways, viz; (*a*) by increasing both breadth and depth, (*b*) by increasing depth only, and (*c*) by increasing breadth only. The determination of multiplying factors for these three cases will now be taken up.

(*a*) INCREASING STRENGTH BY INCREASING BREADTH AND DEPTH

11 Determination of factor for increasing breadth and depth to take care of dead load bending moment. The bending moment due to dead weight of the beam is

$$\frac{w\,l^2}{8} = \frac{150\ b\ d\ l^2}{8\times 144}\text{ft-lb.}$$

$$\frac{w\,l^2}{8} = \frac{150\times 12\ \ b\ d\ l^2}{8\times 144}\text{ in-lb.}$$

$$= 1.56\ b\ d\ l^2 \text{ in-lb.}$$

where

$b =$ breadth, inches

$d =$ depth, inches

$l =$ span, feet

$150 =$ weight of concrete

12 If we increase each dimension (b and d) by a certain per cent and let $k = 1 +$ ratio of increase of each dimension, then the bending moment due to the dead weight of this new beam is

$$1.56\ (k\,b)(k\,d)\ l^2 = 1.56\ k^2b\ d\ l^2 \textit{ in-lb.}$$

The resisting moment of the beam of dimensions b and d is $R\,b\,d^2$ in-lb. If we increase each dimension (b and d) by a certain per cent and let $k = 1 +$ ratio of increase of each dimension then the resisting moment of the new section becomes

$$R\ (k\,b)(k\,d)^2 = R\ k^3b\ d^2$$

13 If $R\ b\ d^2 =$ resisting moment just sufficient to support the load on the beam, then we may consider the excess resisting moment of $R\ k^3b\ d^2$ as resisting the dead load bending moment of the enlarged section. This condition obtains if we write

$$R\ k^3b\ d^2 - R\ b\ d^2 = 1.56\ k^2b\ d\ l^2$$

$$R\ k^3d - R\ d = 1.56\ k^2l^2$$

$$R\ d\ (k^3 - 1) = 1.56\ k^2l^2$$

$$\left(\frac{1.56\ k^2}{k^3-1}\right) l^2 = R\ d$$

(b) INCREASING STRENGTH BY INCREASING DEPTH ONLY

14 Determination of factor for increasing depth only to take care of bending moment due to dead weight of beam. As before, dead weight bending moment $= 1.56\ b\ d\ l^2$ in-lb. If we multiply depth by $k = (1 +$ ratio of increase of depth) then the dead weight bending moment of the enlarged beam $= 1.56\ k\ b\ d\ l^2$. The resisting moment of a beam $= R\ b\ d^2$. If we multiply the depth by $k = (1 +$ ratio of increase of depth) then the new resisting moment of the enlarged beam $= R\ b\ (k\ d)(k\ d) = R\ k^2 b\ d^2$.

15 As before, equating the increase in strength to the dead weight bending moment of the enlarged section we have

$$R\ k^2 b\ d^2 - R\ b\ d^2 = 1.56\ k\ b\ d\ l^2$$
$$R\ k^2 d - R\ d = 1.56\ k\ l^2$$
$$R\ d(k^2 - 1) = 1.56\ k\ l^2$$
$$\left(\frac{1.56\ k}{k^2 - 1}\right) l^2 = R\ d$$

(c) INCREASING STRENGTH BY INCREASING BREADTH ONLY

16 Determination of factor for increasing breadth only to take care of bending moment due to dead weight of beam. Dead weight bending moment $= 1.56\ b\ d\ l^2$. If we multiply the depth by $k = (1 +$ ratio of increase of breadth) then the dead weight bending moment of the enlarged section $= 1.56\ k\ b\ d\ l^2$.

17 The resisting moment of a beam $= R\ b\ d^2$. If we multiply the breadth by $k = (1 +$ ratio of increase) then the resisting moment of the enlarged section $= R\ k\ b\ d^2$.

As before, equating the increase in strength to the dead weight bending moment of the enlarged section we obtain

$$R\ k\ b\ d^2 - R\ b\ d^2 = 1.56\ k\ b\ d\ l^2$$
$$R\ k\ d - R\ d = 1.56\ k\ l^2$$
$$R\ d\ (k - 1) = 1.56\ k\ l^2$$
$$\left(\frac{1.56\ k}{k - 1}\right) l^2 = R\ d$$

It will be noted that the form of the three expressions above developed is the same, the only variation being in the exponent of the term k enclosed in the parenthesis. A tabulation of the value of the parenthesis corresponding to different values of k is given in Table 1.

18 The second diagram in Fig. 4 has been prepared from the preceding data. In general form it will be noted that it is quite

similar to the other diagram. The values of the parenthesis in the first equation are represented by the left hand logarithmic scale but instead of writing the values down the corresponding values of k are noted opposite the proper lines. The lines representing the span were plotted in much the same way as the lines representing d in the other diagram. To avoid making extra diagrams for the other equations two more logarithmic scales were added at the right and the proper values of the parenthesis in each case corresponding to the assigned values of k were plotted. Here too, the corresponding values of k were

TABLE 1 TABULATION OF VALUE OF THE PARENTHESIS CORRESPONDING TO DIFFERENT VALUES OF k

Value of k	$\frac{1.56\,k^2}{k^2-1}$	$\frac{1.56\,k}{k^2-1}$	$\frac{1.56\,k}{k-1}$
1.01	53.00	78.00	157.00
1.03	17.80	26.80	53.50
1.04	13.60	20.20	40.60
1.05	11.00	16.00	37.80
1.06	9.17	13.40	27.60
1.07	7.94	11.50	23.80
1.08	7.03	10.10	21.10
1.09	6.30	9.04	18.90
1.10	5.70	8.17	17.20
1.11	5.24	7.46	15.70
1.12	4.84	6.87	14.60
1.13	4.50	6.37	13.60
1.14	4.22	5.92	12.70
1.15	3.96	5.67	12.00
1.20	3.12	4.25	9.36
1.25	2.56	3.47	7.80
1.30	2.24	2.94	6.76
1.35	1.95	2.56	6.02
1.40	1.75	2.28	5.46
1.45	1.60	2.06	5.03
1.50	1.48	1.87	4.68

noted rather than the value of the parenthesis. At the top of the diagram the values of d are plotted on the line $R = 100$. Through the values of d as indicated there is a family of diagonal lines which intersect the horizontal lines representing other values of R. It may be seen by inspection that the product of any R times any d is equal to 100 times the logarithmic reading obtained by projecting the point of intersection of the horizontal line through R and the oblique line through d vertically upward or downward to the horizontal scale representing depth of beam.

19 Having determined by means of the first diagram, the dimensions of a beam suitable to support the desired load, we are now

ready to find the correction factor by which one or both of the said beam dimensions must be multiplied in order to give the beam enough additional strength to support its own dead weight. Using diagram (Fig. 4) and starting on the depth scale at a point representing the depth first determined follow oblique line up or down to intersect value of R that is being used. Thence follow vertically downward to intersect the line representing the span of the beam. If both dimensions are to be increased the value of k may be obtained by proceeding thence horizontally to the left hand scale. If one dimension only is to be increased (the other being kept at the value first determined) proceed horizontally to the right to intersect the proper scale.

20 To summarize the foregoing it may be well to work out a problem in detail. The writer has found the following to be a very convenient and useful form of computation:

Girder G_7

Span = 18 ft. Load = 370 lb. per linear ft.

Bending moment $= \frac{w\,l^2}{8} = \frac{370 \times 18^2}{8} = 15{,}000$ ft-lb.

Shear = 3330 lb. + ½ weight of beam

$f_s = 16{,}000$ $f_c = 600$ $R = 94$ % steel = 0.68

Square steel rods to be used

$b_1 \times d_1$	12.5×12.5	9.4×14.10	7.5×16 (See Par. 9)
Corrector	1.17	1.145	1.136 (See Par. 12)
$b \times d$	14.6×14.6	10.8×16.1	8.5×18.2
$b\,d$	213	174	155
Unit shear	25	29	31
Steel	Four ⅝-in. bars, or Three 11/16-in. bars	Four ⅝-in. bars, or Three 11/16-in. bars	Four ⅝-in. bars, or Three 11/16-in. bars

21 The above tabulation shows three beams which would be suitable. The one which best suits the rest of the structural design may be used. The correction factor used was the one corresponding to the increase in both dimensions. It should be noted that d represents the effective depth, and the distance from the steel reinforcing to the bottom of the beam must be added to get the total depth. This "cover" portion of the beam has been neglected in all of the computations for strength. The determination of the second beam in the tabulated form is plainly indicated on the diagrams so that the method of using them ought to be readily understood.

No. 1486

SOME MECHANICAL FEATURES OF THE HYDRATION OF PORTLAND CEMENT AND THE MAKING OF CONCRETE

By Nathan C. Johnson, New York, N. Y.

Member of the Society

The data embodied in this paper have been obtained in the course of a research conducted in the Sibley College laboratories at Cornell University.

2 At the beginning of this research, it was believed that the obtaining of more effective hydration was a prime factor in the production of durable concrete. It has long been known experimentally that a set cement or concrete could be reground and a new set obtained on gaging with water;[1] but the extent to which unhydrated particles were present in the mass was first made visually evident, so far as the author is aware, by experiments conducted in 1911 in the Sibley College Laboratories at Cornell.[2] In these experiments, neat cement briquettes were surfaced and polished, as in the metallography of steel; and the degree of hydration obtained in different ways was more or less accurately judged by the appearance of these polished sections, viewed through the microscope. Later this same procedure was extended to the examination of 1:3 standard sand mortars and still later to the examination of field concretes.

3 A section of such 1:3 mortars photographed through the microscope, is shown in Fig. 1. The grains of sand show as large as boulders and each has a well-defined shadow, just as would a true boulder in a field when the sun is at the zenith. Further, since only

[1]H. Sager & E. Cramer. Touind Ziet, 1908, 32, 1746.

[2]Masters Thesis by N. C. Johnson, Cornell University, 1913.

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

cement and a sand of known size were used in the briquette, the "rocks" between the boulders can be nothing other than cement; and this cement must be unhydrated, since its color by direct vision, varying from light green to dark brown, and its resistance to the cutting of the polishing powders used in preparing the surface to remove the portion known to be hydrated, together with petrographic examination, all testify to the same fact.

4 To ensure a thorough understanding of the evidences and effects of hydration of cement, it may be well to consider here the nature of the processes involved. As is well known, Portland cement contains in varying amounts and combined in various ways, a number of chemical radicals—CaO, MgO, SiO_2, Al_2O_3, Fe_2O_3, SO_3, H_2O, etc. Some of these compounds are due to the cement rocks themselves and others, notably a certain proportion of the CaO, SO_3 and H_2O present, to the secondary addition of gypsum to control the setting. For the present this product will be assumed to be satisfactory, judged from the standpoint of the average user.

5 The processes of pulverizing the raw materials, sintering, and regrinding are so well known as to require no review. The completed cement derives its usefulness from the hardening produced shortly after its admixture with water.

6 There are two stages in the passage from the semi-fluid state of gaged cement to that of the hardened. In the first stage the mass loses its plasticity and becomes more or less friable. In the second stage consolidation takes place, the mass increasing in hardness until a stony texture is obtained. These two stages are respectively distinguished as "setting" and "hardening."

7 It follows from the complex character of its composition, that the reactions involved in the setting and hardening of Portland cement are themselves complex. It seems probable, however, that the reactions of setting involve the formation of super-saturated solutions and the deposition therefrom of close-knitted, interlacing crystals of various substances, while the slower reactions of hardening consist partly in the formation of similar interlacing crystalline products, but more especially in the production of a colloidal "gel" or glue-like substance, probably colloidal calcium hydrosilicate, and its gradual desiccation.

8 This colloidal interpretation of the hardening of cements is due to Dr. Wm. Michaelis, Sr., who reasoned that a pure crystalliza-

tion process, as in the case of plaster of paris, can never cause hydraulic hardening, for a conglomerate of crystals, however insoluble, cannot produce impermeability, since crystals have plane surfaces with voids between them which would admit water. On the other hand, hardened cement mortars are properly impermeable, so that crystallization alone could not confer on them their valuable properties. The medium responsible Dr. Michaelis found to be the colloidal silicates formed in the later stages of hydration.

9 The essential characteristics of the hydration of cement are therefore the production of initial set by the deposition of interlacing crystals from super-saturated solution, and the production of hardening by the slower formation of a colloid, with its subsequent desiccation.

10 Illustrative of such structures and possibly of the primary crystalline formations which have been indicated as being responsible for initial set, in Fig. 2 is shown a surface found in a concrete 34 years old, taken from a dock wall in New York harbor. The production of these crystals was quite unlooked for, being due to an unusually delicate etching of the piece of clinker after polishing. It should be remembered that this magnification is high (150 diameters) and that these crystals are formed on a single particle. Multiply such interlacing formations a few million times for each cu. yd. of concrete, and it might be expected that a very considerable degree of mechanical strength, quite comparable to the strength of initial set, would be developed. This particular crystalline formation is shown tentatively, however, in this connection, as its identity with those chemical compounds known to be responsible for initial set is as yet conjectural.

11 But since this attack on the problems of hydration is primarily engineering rather than chemical, and must therefore necessarily be identified with a study of actual concretes, it becomes necessary to make sure that there is a proper understanding of the internal structure of concretes and of the relative size, position in the mass and importance of the several constituents.

12 In Fig. 3 is shown a sectioned concrete specimen, weighing about ten lb. and about 8-in. $\times$ 6-in. $\times$ 1.5-in. size. It should be noted that none of the aggregate, whether large or small, so far as they are here visible, are touching at any point; rather, they are widely dispersed. But void determinations on this stone were made

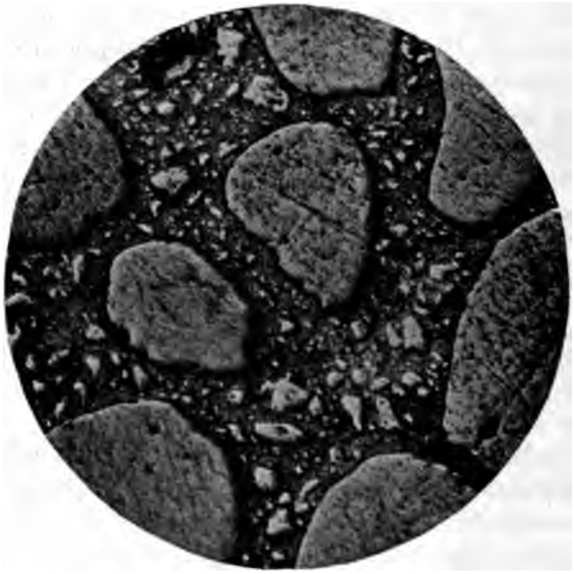

FIG. 1 MICRO-PHOTOGRAPH OF 1:3 MORTAR

FIG. 2 MICRO-PHOTOGRAPH OF 34-YR. OLD CONCRETE

Fig. 3 Sectioned Concrete Specimen, Magnified 2 Diam.

Fig. 4 Portion of Field of Fig. 3, Magnified 150 Diam.

when the pieces were in at least point contact. Evidently, the voids must have been increased enormously by this dispersion; and if similar dispersion obtains for the finer materials as well, it is no wonder that proportioning concretes on the basis of void determinations has become discredited.

13 Magnifying a portion of this same field 150 times, as in Fig. 4, this honeycomb structure is revealed more definitely. Since elimination of stone and sand from the known composition of the concrete indicates this honeycombed particle to be unhydrated cement, and as similar formations occur profusely throughout all concretes, it becomes important to identify them beyond question and to study their functions in the rigid concrete mass, for, if they are unhydrated cement, lying inert, they have a most important bearing on the strengths of concrete.

14 The easiest identification is offered by comparison between the known and unknown material. In Fig. 5 is shown the structure of a neat cement briquette as viewed through the microscope at a magnification of 60 diameters. In Fig. 6 is shown the central portion of Fig. 5 but at a magnification of 200 diameters. Comparing this with Fig. 4, the general characteristics are seen to be identical.

15 In Fig. 7 is shown at a magnification of 20 diameters the appearance of dry cement as it lies on a microscope slide. This cement film is very thin, a mere surface dust, but it will be seen that the distribution is very uniform. But if water is added, quite different conditions obtain. In Fig. 8, the water may be seen progressing across the field. Where the water is, there is not, as might be expected, a uniform wetting of the dry powder, but instead, the fine material is grouped in masses of varying size, and it is plain that some extraneous force would be required to break up these masses, since they evidently are in their present positions in obedience to the action of certain forces which have moved the particles from their original place.

16 The truth of this is easily demonstrable. Surface tension—that peculiar chain of action and interaction which causes rain drops to assume the spherical form—gathers the particles together in groups; and the finer they are, the more easily are they affected. The fineness of cement particles is, in this respect, a detriment to efficient hydration.

17 But in the process of this grouping, the particles have be-

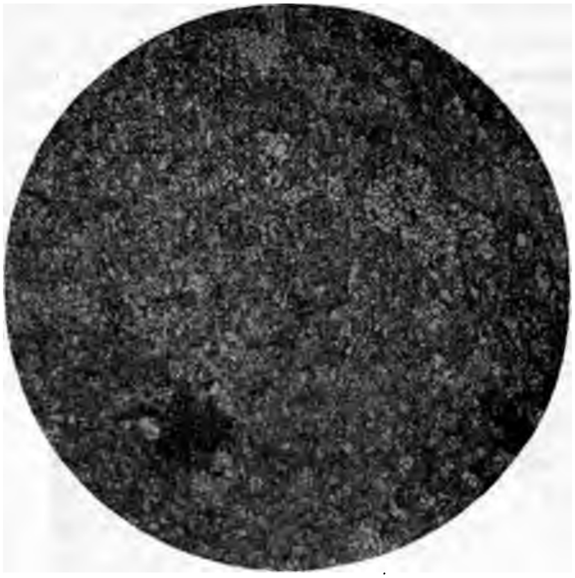

FIG. 5 STRUCTURE OF NEAT CEMENT BRIQUETTE, 60 DIAM.

FIG. 6 SAME AS FIG. 5, BUT 200 DIAM.

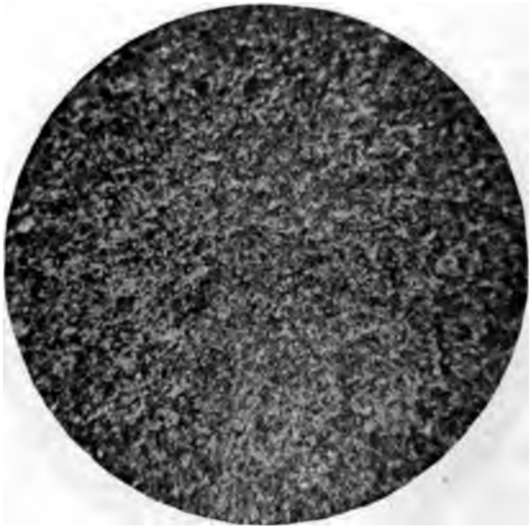

Fig. 7 Dry Cement, Magnified 20 Diam.

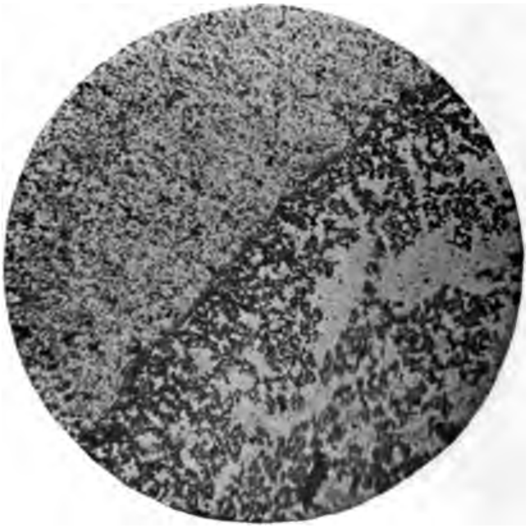

Fig. 8 Effect of Adding Water to Dry Cement

come slightly wetted. This has immediately resulted in reactions with the cement; and as the quantity of water is necessarily limited by the close grouping of these fine particles, a saturated solution between them, with deposition therefrom of the interlacing crystals before noted, has speedily resulted. In far shorter time, therefore, than is required for initial set under usual conditions, these little groups of particles have become consolidated; and unless the mechanical agitation of mixing is sufficient to break down this crystal-

FIG. 9 METHOD USED IN COLLECTING SAMPLES OF FIELD CONCRETE

line bond, they will remain unhydrated, save at their outer surfaces, so long as the concrete endures. This condition of affairs is difficult to overcome, even by the mechanical action of mixing, as the very small size of the groups renders unlikely their physical dispersion by the agitation of relatively large materials such as sand or stone, which by reason of the water present, are out of contact with one another. This is partially true regardless of the extent to which the process is prolonged. Further, the bond between these particles is speedily rendered more secure by scondary colloidal formations, with their desiccation by absorption of water by the interior of the mass. These colloidal boundary and bonding masses form the honeycomb structure seen in the photograph. Their appearance as white boundaries in some slides and dark boundaries in others is due to the method of polish-attack used in developing the surface prior to its examination.

18 This grouping and quick-cementing action, therefore, is the reason that experimental apparatus developed for the mechanical hydration of cement by finely dividing the cement by a current of air, so that the particles were widely separated, and by discharging this cloud of cement into an atmosphere of finely divided water, so that each particle of cement should meet a mating water particle, proved inadequate for the task. This is particularly true of later forms, for in them the number of atomizing jets was increased to such an extent that the fine drops, though initially of a size suited to an individual cement particle, were in such numbers as to coalesce; and the larger drops thus formed caused, by surface-tension, grouping of cement particles.

19 With a view to ascertaining the results produced by field conditions, as distinguished from the laboratory-made specimens, sections of a very large number of concretes obtained from actual commercial constructions were examined. To aid in the collection of these field concretes, bags such as that shown in Fig. 9 were employed.

20 Representative surfaces from the samples thus collected are shown in Figs. 10 and 11.

Fig. 10 shows at a magnification of 150 diameters a large group of unhydrated cement particles from concrete in Pier 2, in Halifax, N. S., eight months after the concrete was placed. The black spots dotting this group are voids of indefinite depth, and it is probable that because of these, hydration will increase considerably with progress of time, as through their agency, water will be brought to the interior of the group, instead of being confined in its action to outside layers.

Fig. 11 shows at a magnification of 150 diameters two groups of unhydrated cement particles from pile caps, Staten Island Ferry Pier, Battery Park, New York, eight years after placing the concrete. It will be observed that hydration has not progressed appreciably.

20 Examination of these specimens shows that similar conditions exist throughout the entire mass of each and since none of the

FIG. 10 CONCRETE FROM PIER 2, HALIFAX, N. S.

FIG. 11 CEMENT FROM PILE CAPS, BATTERY PARK, NEW YORK

specimens is over 1 cu. in. in volume, the quantity of cement remaining unhydrated through such grouping in each cu yd. of concrete is a matter to ponder upon. It is easy to extend like illustrations to a far greater length with the data already at hand, but the series herein examined covers a wide range of classes of concrete of varying ages and conditions of service, and made under widely different conditions. *Grouping of particles in the manner indicated is a characteristic of all concretes so far examined;* and because of this grouping and of actions next to be indicated, it seems beyond question that only a small percentage of the cement added to concrete is effectively used.

21 Obviously, it is of the greatest importance that this grouping and isolation of large masses should be prevented. But even if this were quite perfectly done, there remain other actions which militate against thoroughly efficient hydration.

22 In microscopic examination of concretes, one of the most notable features of the hydrated matrix, aside from the large masses remaining unhydrated, is a ''mealy'' or speckled appearance. By careful examination, it can be determined that each of these spots is a cement particle, covered over with a colloidal, or perhaps a colloidal-and-crystalline skin, but remaining quite unhydrated at its center.

23 The reasons for these conditions are made clear by a consideration of the phenomena of hydration as before given, together with the absorption of water by the particle as hydration proceeds and the formation of an impermeable covering by the desiccation of this colloid through abstraction of water from the outer layers by the inner. Further, even with free access of water to the outside of a particle these conditions obtain, for in an agitated mass of cement in spite of the profuse formation of flocculent colloids an unhydrated center is found in each of the particles examined.

24 It is evident therefore that the finer the grinding, the better will be the hydration, as the finer the particle, the greater its surface in proportion to its bulk and in consequence, the more ready the penetration of reactive water. On the other hand, the finer the particles, the more ready and the closer will be their grouping by surface tension of the mixing water unless means can be found to lower this surface tension, without harmfully affecting the quality or increasing the cost of the concrete.

FIG. 12 DISPERSION OF CEMENT PARTICLES BY ALCOHOL

FIG. 13 "SHEAR PLANES" IN CEMENT MATRIX

25 The importance of increasing hydration cannot be exaggerated. Even a slight increase in thickness of hydrated layer results in notable increase in strength. In the earlier, though limited, investigations of this research when cement was delivered by an air-boil, to be met by finely atomized water under the best conditions for union, tensile and compressive strengths rose to high values, with a further peculiarity that no retrogression showed with passage of time. Micro-sections also confirmed physical tests as to the progress of hydration, so that high hopes were entertained for commercial success along similar lines.

26 But since the conditions revealed by this research have become known and their causes have been ascertained, the next step is the overcoming of objectionable features by removing the causes of their origin. In other words, if high surface tension is at fault, to promote hydration, lower that factor to a value which will not be disadvantageous. There are several substances which will lower the surface tension of water. One of these is alcohol, and the effect of additions of alcohol to the mixing water of cement may be readily investigated with the microscope.

27 In Fig. 7 is shown the appearance of dry cement on a glass microscope slide. In Fig. 8 is shown the effect of adding water to this cement. But if to this water is added a small percentage of alcohol, dispersion immediately results, as in Fig. 12. Although the dispersion here is not complete, the cement is in vastly better situation to be effectively hydrated than it was when closely grouped, before the deflocculant (alcohol) was added.

28 In commercial work it is of course impracticable to use alcohol in sufficient quantities to properly promote hydration, nor would its use be beneficial to the concrete, because of the excessive liquid content thus made necessary to allow for the evaporation of the alcohol. But there are other agents even more efficient, which are cheap and easy to use and have no objectionable chemical action. So far as experiments have been carried out, these substances promise to produce concrete of greater density and strength than has heretofore been possible.

29 There have been in the past certain cases of failure of concretes exposed to sea-water, which have become very widely known. It is probable that to no one phase of the concrete industry has so much thought and labor been devoted as to the study of the behavior

FIG. 14 PARALLEL SHEAR PLANES, 200 DIAM.

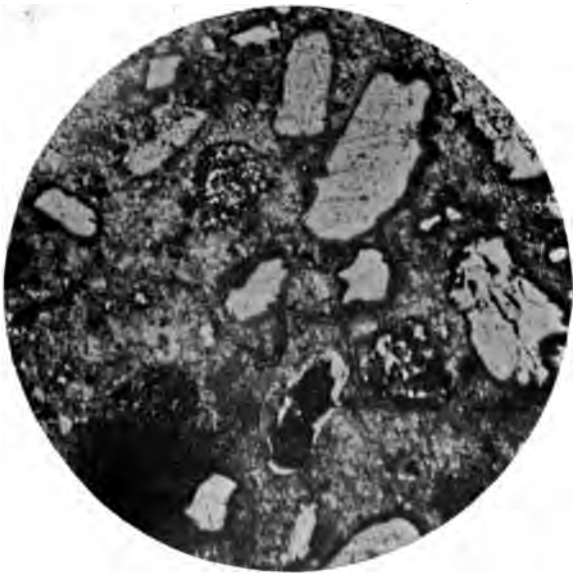

FIG. 15 CRYSTALLINE DISRUPTIVE ACTION OVER CONSIDERABLE AREA

of concretes in sea-water and the production of a cement capable of withstanding this action. There is considerable disagreement among authorities as to the causes of such disintegration, but there seems to be concurrence in the opinion that the formation of calcium sulpho-aluminate by interaction between the sulphates of sea-water and the aluminates of the cement, is in large measure responsible. This salt (calcium sulpho-aluminate) increases largely in bulk by crystallization and disrupts the concrete by the physical actions attendant thereon.

Fig. 16 Appearance of Disruptive Dikes in Natural Stone similar to Crystalline Action Shown in Fig. 15

30 If we were to picture mentally the formation of such an expansive material in the cement matrix of a set concrete, we should have to imagine a gradual straining of the confining material until rupture occurred. It is significant as to the correctness of such an hypothesis, that in the matrix of concretes which show outward signs of disintegration are found by microscopic study interior evidences of such strain.

31 In babbitts and the softer bronzes, incipient fracture is indicated by "shear planes" or "slip bands." In Fig. 13 at 150 diameters are shown similar "shear planes" in the cement matrix of a sea-water concrete taken from New York Harbor. It will be observed that these shear planes are at right angles to the polish scratches on the specimen, so that their lack of identity cannot be questioned.

32 Various stages of such crystalline disruptions, indicative

at least of the possibility of such action, are detected by further micro-examination. In Fig. 14 at 200 diameters is seen a pair of parallel shear planes which have evidently progressed beyond the stage of strain and become actual fractures filled with crystalline material, as may be determined by their having withstood the abrasive action of the polishing powders, which cut away the softer matrix surrounding, leaving these fillings in high relief.

33 Under certain conditions also this crystalline disruptive action may affect a considerable area. In Fig. 15 is shown at a magnification of 60 diameters part of such an extended action, where the crystalline formation is a long, disruptive dike. It is greatly to be regretted that these dikes are so minute as to render their isolation and chemical analysis almost impossible. Such an analysis would be of great value in deciding many questions which at present seem to defy solution. And while this subject of crystalline disruption is under discussion, it may be well to point out that natural stones, either themselves containing soluble matter or lying in a locality where seepage water is charged with dissolved salts of various kinds, show similar dike formations. The appearance of such a natural ledge, with disruptive dikes of various calcium salts extruding ½ in. or more from the surface, is shown in Fig. 16.

34 But whether or not calcium sulpho-aluminate is the major factor in disruptive formations in concrete, there seems to be agreement among cement experts that advantageous changes could be made in the chemical composition of cements, especially where they are to be used for sea-water concrete. A reduction in the alumina content would be beneficial in this respect, though it would entail serious difficulties in manufacture due to the higher temperatures required in burning the clinker. The addition in grinding of pozzolaine material, in order to obtain silica in a condition chemically available for union with the calcium hydrate liberated in the setting reactions, seems also to have much to recommend it. This latter change we should expect to prove very beneficial, for the leaching of calcium hydrate from cement structures which are subjected to the percolation of water is well known. Any such removal of material must necessarily leave a corresponding void suited to the reception of perhaps deleterious crystalline material, such as forms the dikes above noted.

35 There has recently been published a very exhaustive research, by Rankin and Shepherd[1] of the Geophysical Laboratory at Wash-

[1]Am. Journal of Science, Jan. 1915.

ington, on the products formed by fusing various proportions of the three radicals, CaO, Al_2O_3 and SiO_2, which are the essential constituents of Portland cement. A portion of this three-component diagram, taken from the Rankin & Shepherd diagram is shown in Fig. 17.

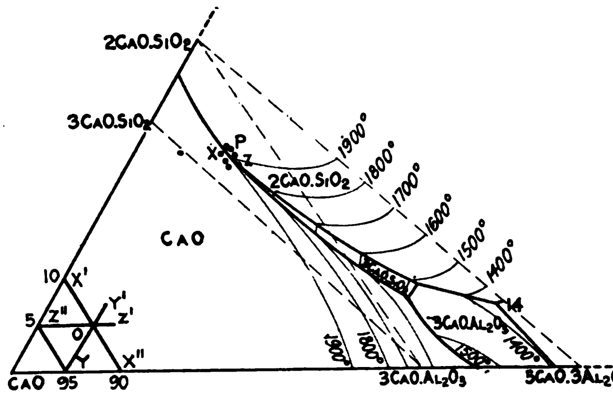

FIG. 17 PROJECTION OF A PORTION OF CONCENTRATION—TEMPERATURE DIAGRAM OF THE SYSTEM $CaO\text{-}Al_2O_3\text{-}SiO_2$ WITH ISOTHERMS. THE DOTS REPRESENT MIXTURES OF CaO, Al_2O_3, SiO_2 FROM WHICH PORTLAND CEMENT OF GOOD QUALITY CAN BE MADE

36 This diagram is a projection on the horizontal plane of a portion of the solid concentration-temperature diagram for the system CaO, Al_2O_3, SiO_2. Temperatures are represented here by isothermal lines, which are completely analogous to the contour lines on an ordinary map. The heavy full lines represent the boundary curves which separate the fields of the different compounds of CaO, Al_2O_3, SiO_2 which occur in Portland cement. The dotted lines which divide the diagram into a number of triangular areas are compound composition lines. Points within each triangle represent all possible mixtures of the three compounds whose compositions are given by the apices of the triangle. The CaO, Al_2O_3, SiO_2 content of points within the diagram are given as percentage weights. As the location of points which represent compositions in a triangular diagram may be puzzling to those not familiar with its use, it may be well to illustrate with examples. The apices of the triangle, of course, represent the components, each 100 per

cent pure; thus the apex for CaO, the only one shown here, represents 100 per cent CaO. The sides of the triangle represent binary mixtures; thus point x'' represents 90 per cent CaO and 10 per cent Al_2O_3. Within the triangle the percentage of CaO is read on lines parallel to the Al_2O_3 SiO_2 side, the percentage of Al_2 O_3 on lines parallel to the CaO—SiO_2 side and the percentage of SiO_2 on lines parallel to the CaO—Al_2O_3 side of the triangle; thus the line $x''x'$ represents compositions containing 90 per cent lime; the line yy', compositions containing 5 per cent Al_2O_3; and the line $z''z'$ compositions containing 5 per cent SiO_2. The point O, where these three lines intersect, has the composition 90 per cent CaO, 5 per cent SiO_2, 5 per cent Al_2O_3. The dots within this diagram represent compositions from which Portland cement of good quality can be made. It will be noticed that the melting temperature of these compositions is about 1900 deg. cent. The temperature at which melting begins in any one of these compositions is about 1335 deg. cent., the eutectic temperature for the compounds $5CaO.3Al_2O_3$, $3CaO.Al_2O_3$, and $2CaO.SiO_2$ (point 14 in the diagram).

37 The temperature required for burning the clinker composed only of CaO, Al_2O_3, SiO_2, whose composition is represented by point P, is 1650 deg. cent. This temperature is below that required for complete melting (1900 deg. cent.) but above that at which melting begins (1335 deg. cent.). From this it is evident that during the burning the clinker is only partially fused. If about 30 per cent of the clinker is melted (the clinker represented by point P would be 30 per cent melted at 1650 deg. cent.), the necessary chemical reactions will go to completion within a reasonable time. In commercial practice the clinker is burned at a temperature of about 1425 deg. cent. That it is possible to produce good clinker at this temperature, which is 225 deg. cent. below that required for burning clinker made up only of CaO, Al_2O_3, and SiO_2, is due to the fact that commercial clinker always contains small percentages of MgO, Fe_2O, alkali, etc. These minor components materially lower the temperature at which melting begins, and thus lower the temperature at which the necessary chemical reactions will go to completion during the burning of the clinker. If these reactions are completed in the burning of a clinker of average CaO, Al_2O_3, SiO_2 content (point z, Fig. 17), the compounds formed will be $3CaO.SiO_2$, $2CaO.SiO_2$, and $3CaO.Al_2O_3$, since point z is within the triangle formed by lines joining the compositions of these three

compounds. Also microscopical examination of commercial Portland cement clinker shows it to be made up largely of these three compounds. It sometimes happens, however, that commercial clinker is not sufficiently well burned so that all chemical reactions are completed. That being the case, there will be present, besides the three compounds mentioned, free CaO and the compound $5CaO.3Al_2O_3$. This compound, $5CaO.3Al_2O_3$, is undoubtedly of little value as a constituent of Portland cement, and may even be detrimental when present in cements used in sea-water construction. This is a phase of the sea-water question in the solving of which the user can do little except demand a material of the quality he desires. Cement manufacturers are, however, caring for this question as adequately as commercial conditions will permit.

38 The diagram, Fig. 17, and the accompanying discussion give a brief but accurate statement of the results obtained from an investigation of Portland cement at the Geophysical Laboratory in Washington.

39 In this discussion the endeavor has been made to bring out some of the salient features in regard to the hydration of Portland cement and its use in sea-water concrete, with particular reference to the little-considered mechanical actions involved. It is impossible in limited space to give the subject full scope, or to make detailed acknowledgments of the very great debt due to other investigators and authorities, both in this country and abroad. If, however, the matter herein set forth proves of some service in the advancement of the art, full payment for the intensive and extensive work involved in its preparation will have been received.

DISCUSSION

Harry Franklin Porter[1] wrote that the author has established, by laboratory methods, conclusions which the writer believed since 1908, that the true cement is a colloid produced by thorough hydration. A special article dealing with this important subject, based largely on the findings of Dr. Michaelis, was inserted in an appendix to a publication by the writer the following year.

Further, at the 1910 convention of the American Concrete Institute, the writer again devoted a considerable portion of a paper to a discussion of this subject, recounting an experience in Philadelphia several years previous, in which the value of thorough

[1]Management Engineer, 1118 Westminster Bldg., Chicago, Ill.

mixing of concrete as promoting colloidal formation was conclusively demonstrated.

This experience was that a certain batch of concrete, supposedly useless on account of having been churned in the mixer during the whole of the noon hour, was dumped on concrete poured the previous day. Instead of becoming a loose mass of rock and sand, as expected, it set up smooth and hard and adhered so firmly to the old concrete as to require picking to remove it. Moreover, it presented a different appearance than any concrete the writer had seen before, and was much more uniform in color and texture. The exceptionally thorough mixing, and the unintentional thorough hydration, instead of weakening it seemed to have vastly improved its quality.

In concrete work, there is necessity for, *first,* intelligent selection of the aggregates; *second,* their scientific proportioning; and *third,* their thorough admixture. If the proper care is taken in these matters, not only is a considerably less amount of cement sufficient, but a concrete of vastly superior quality and strength is secured.

John R. Freeman considered the paper, while excellently setting forth advanced methods in microscopic analysis of Portland cement mortar, gave simply a diagnosis of some of the most important diseases to which cement work is subject, without yet giving the remedy.

He looked upon the matter of applying the methods of micro-petrography and micro-metallurgy to concrete study as nothing less than epoch-making, and considered that the recent studies made by the author and others gave great promise of extremely important benefits. Yet so far as these studies were reported, he regarded them merely as an excellent beginning of a more precise, profound and intimate study of this most important building material which should be encouraged by engineers and by cement manufacturers and carried very much further.

This paper considers chiefly the methods of microscopic study of voids and lack of hydration; another investigator has made a good beginning on studies of a soluble content of the cement. But it is no less important that the study should be continued into the realms of microphysics and colloidal chemistry, if one would learn the cause of some of the occasional serious failures in concrete and its slow disintegration by water and frost.

He had no doubt that ten years from now, following studies of this kind, a better grade of concrete would be made, and that some cement made by recent rapid and cheap methods and acceptable under present standard specifications would come to be regarded as doubtful if not dangerous. For Portland cement is by no means a simple uniform material, and different brands and perhaps different batches of the same brand are far from being alike and perhaps differ in very important particulars from the cement made ten, twenty or fifty years ago, with which the reputation of Portland cement as a permanent water-resisting and frost-resisting material was established.

He recalled the great change within the past thirty or forty years in mixing and depositing Portland cement concrete; and that the methods of distributing and depositing concrete in a sloppy condition which are standard today would not have been tolerated in the days when standard practice was mixing concrete by shovels on a mortar bed, turning it over two or three times, and depositing it in a mealy, slightly moist condition in thin layers and carefully ramming it until water flushed to its surface. Today mixing by rotary machines is surely far more thorough and better than the old hand-mixing, and this improvement doubtless more than compensates for the possible injury from excess of water in the modern method as compared with the old.

While nearly everyone knows of this great change in methods of mixing and depositing concrete, there are not many who fully appreciate the great change within the past few years in methods of making cement in virtually a single operation under intense heat in rotary kilns; and we cannot yet be sure that the cement concrete put down today will all of it prove as durable as most of that made with the old-fashioned cement and deposited by the old-fashioned methods.

Microscopical studies like those presented will do much toward finding out how to improve the character of the cement and how to improve the quality of concrete, and today give more promise than any other line of investigation. But the studies must be carried far beyond the scope outlined, and the most refined processes of physical chemistry called in to aid.

He suggested two matters of special importance which needed further study:

(1) Possible solubility of some of the ultimate compounds, resulting from the setting of the modern cements, whereby structures having thin sections would in course of

time lose much of the strength on which their integrity depends.

(2) Effect of obscure chemical properties in good-looking sand, which might seriously affect the strength of the mortar.

To illustrate these matters he cited a few tests made, incidental to other work, in the laboratory of the New York Board of Water Supply, which suggested this danger of dissolving out and loss of the strength-giving quality in thin sections of concrete of somewhat pervious or lean quality. These tests were a few made primarily to determine whether or not aggregates containing certain varieties of limestone were soluble, but in the series were specimens made up for control purposes with aggregates essentially insoluble. Croton water under about 40 pounds pressure was made to percolate very slowly through disks some 4 or 5 in. in diameter by 10 or 12 in. broad for a number of months and the filtrate caught and subjected to chemical analysis, with the result that it appeared that more of the cement than of the aggregate was being dissolved. At the close of this test a few of the disks were tested for compressive strength with the surprising result that they possessed, as the speaker remembered it, less than one-fourth of the strength of similar disks which had not been subjected to this slow percolation. These experiments were not followed up because of lack of time and pressure of other work, but they give much food for thought, and it would seem that, after all this percolation, the cement in these specimens must have been hydrated and thus free from one important defect found by the author in most of his specimens.

He said he had recently been shown the results of an investigation of disintegration of concrete which seemed to clearly indicate that the cement was partially dissolved out near the surface after a few years exposure to water, and that the action of frost on the slightly porous mass remaining caused its breaking down.

Upon the second topic, of obscure chemical properties, and as illustrating the need of supplementing the microscopic studies by some researches in colloidal chemistry or by researches advanced into the realms of physical chemistry, he cited some troubles he encountered three or four years ago when planning a concrete dam. Close to the site he found what seemed to be a perfect mixture of gravel and sand, well graded and apparently clean and requiring simply to be mixed with cement and water in order to give excellent concrete, but on sending samples of this material to the laboratory for mechanical analysis and test of mortar, it was found that the mortar made from this sand would not stand up sufficiently to be held in the clamps of a testing machine after setting 7 days, and after setting for 28 days it gave a strength less than one-fifth of the normal when tested with four or five different brands of cement; yet one particular brand of cement was found which gave a fairly strong mortar with this same sand, thus proving that the trouble was to be looked for in the sand rather than in the cement, but no reason has yet been found why this one particular make of cement worked so very differently with this peculiar sand. After washing this sand in a way intended to cause mutual attrition and scrubbing of the individual grains, it was found to give tolerably strong mortar with all these brands of cement, and on analyzing the wash water by methods used for drinking water the reports showed a large percentage of albuminoid ammonia. An expert of the U. S. Department of Agriculture, who had spent much time on studies of the solubility of the mineral ingredients of different soils, suggested to him that the peculiar behavior of this sand in relation to Portland cement was probably due to a low, complex organic acid, akin to tannic acid, which was present in a condition of *adsorption* on the outside of the sand grains, and that to find out what was worth knowing about this action might keep a skilled research chemist busy for a year.[1]

It is certain, he said, that some concrete has withstood frost and the wear of rapid currents of water for many years marvelously well. On the other hand, it is certain that some concrete from cement made by rapid modern methods, under rigid specifications and conscientious inspections, has gone to pieces in a most distressing way. It is possible that some of the modern concrete put down with a super-abundance of water, and perhaps some that is made with cement containing ingredients leading to compounds that will ultimately become soluble, may bring disaster when concrete is used in thin sections under high water pressure and expected to retain its normal strength indefinitely.

He considered the subject extremely important, and that engin eers should be grateful for all contributions to it like the presen paper and should be pleased to note that while the author no simply gives his method of diagnosis, he implies that in a futu

[1] *Engineering News*, July, 1912, contribution by the discussor.

paper a remedy for some of the present troubles will be presented.

THE AUTHOR. The comment most frequently met with in regard to this microscopic study of concretes is that it does not indicate a remedy for the defects shown. Microscopic examination is primarily a diagnosis. It shows concretes as they actually are, not as they are supposed to be, and the faults that careless procedure and improper selection of materials induce. Further, the inter-relation and the inter-action of the several materials composing concrete are made plain, together with the penalties attendant upon any neglect of these inter-relations.

The remedy lies very largely in alterations of our present procedure. It is very true that there are certain beneficial changes that might be made in the cement, which is not by any means an ideal product, but it is true, also, that more care should be used in the selection and particularly in the quantities of the other constituents used. When the concrete maker realizes that the quantities and gradings of his stone, and especially of his sand, are vitally important, and when he also realizes that the quantity of water may not be varied beyond certain limits, we may hope to see a very important change in the quality of concrete.

It should also be realized by engineers that concrete is not an infinitely dense, hard, resistant substance, but that, on the contrary, it is a material which varies in durability, and one that varies widely in proportions, qualities and value, largely depending on the care and understanding used in its manufacture.

No. 1487

MODEL EXPERIMENTS AND THE FORMS OF EMPIRICAL EQUATIONS

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Member of the Society

It would often be very advantageous if the behavior under service conditions of projected structures or machines could be predicted from the results of experiments on small models; even roughly approximate information, if thoroughly reliable within its known limits of accuracy, would often prevent serious failures or enable the designer to avoid the use of exaggerated or badly co-ordinated factors of safety and allowances for uncertainty. Experiments on the resistance of ship models have been used in this way for many years with satisfactory results, and have been of great value to the naval architect. On the other hand, it is a familiar fact that in many instances, small scale models do not act at all like their full-sized originals under conditions which seem at first sight to be similar, and that hasty conclusions from model experiments are very unsafe indeed.

2 It is of interest to enquire into the general principles involved in the use of models, and to find, if possible, conditions which must govern such experiments in order that the model shall be similar to its original and its behavior give definite information about the behavior of the original. Just what is meant by *similar*, depends on the nature of the particular case in hand; but there are general rules which show us how to make a model similar to its original, when this is practicable, and which also show us how and why it is often not practicable to fulfill the required conditions. I shall not dilate on the origin of the rules, which are a simple and immediate consequence of familiar physical principles, but will illustrate their meaning and practical application by a few simple examples. There is nothing essentially new in what I have to

Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

present, but the subject seems to be rather unfamiliar to engineers in general and is worth discussing because of its frequent practical utility.

THE GENERAL THEOREM

3 To interpret model experiments we have to know how the behavior of the model or the original depends on size; i.e., we must have an equation which describes the behavior of the machine or structure under the conditions of service, and which contains as variables, the size of the machine and the quantities such as speed, applied forces, viscosity of the surrounding medium, etc., which suffice to specify all the essential circumstances of operation. Such an equation is called a physical equation.

4 Except in the most simple cases, the required equation can be found only by experiment, and to set up a satisfactory empirical equation may require very laborious and expensive experimentation if the problem is at all complicated. But the so-called "principle of dimensional homogeneity" is sometimes of great assistance in the following ways: First, it directs our attention to the things we need to measure and keeps our eyes open to the simplifying approximate assumptions we may have to adopt in our work. Second, it reduces the number of separate quantities that have to be varied and gives hints and suggestions as to the most economical way of getting the desired information. Third, it always gives us some information as to the possibilities, and by showing conclusively that certain empirical equations cannot possibly be generally valid, warns us against trusting them too far outside the range of the experiments from which they were deduced and which they may represent quite satisfactorily. Fourth, it sometimes enables us to put equations in such a form that we can use English and metric measurements indiscriminately without wasting time on conversion. And last, and most important for the present discussion, it often enables us to dispense with complete experimental investigations and shows us how very incomplete sets of experiments may give reliable information in particular cases.

5 The principle of dimensional homogeneity states that all the terms of any correct and complete physical equation must have the same dimensions. By this is meant merely that if the numerical value of any term in the equation depends on the size chosen for one of our fundamental units, all the other terms must depend on it in the same way, so that when the size of this unit is changed, the terms

will all be changed in the same ratio and the equation will remain valid, which it would not do otherwise. The necessity for this may be seen from the consideration that the relation, to be described by the equation, among the sizes of certain physical quantities, is a reality which subsists quite independently of any arbitrary choice of units on our part. Hence if the equation is to be complete and correct, the description of facts which it gives must not change when we arbitrarily change from pounds to kilograms or from inches to miles.

6 By means of this principle it may easily be shown[1] that any equation

$$F(Q_1, Q_2, \ldots Q_n) = 0 \qquad [1]$$

describing a relation among the n different kinds of quantity $Q_1, Q_2, \ldots Q_n$ is always reducible to the form

$$f(\Pi_1, \Pi_2, \ldots \Pi_{n-k}) = 0, \qquad [2]$$

in which each of the variables Π represents a dimensionless product of the form

$$\Pi = Q^a{}_1 Q^b{}_2 \ldots Q^n{}_n; \qquad [3]$$

k is the number of independent fundamental units needed in specifying the units of the n kinds of quantity; and f is some unknown function to be found by experiment.

7 A dimensionless quantity is one of which the numerical value does not change when the sizes of the fundamental units alter so long as the relations between the derived and the fundamental units are kept unchanged. The simplest example of such a quantity is the ratio of two quantities of the same kind; the ratio of two lengths, for instance, does not depend on the unit we adopt for measuring lengths. Another simple example is the expression $\frac{Dg}{S^2}$ or $D\,g\,S^{-2}$, in which D is the diameter of a fly wheel, S its peripheral speed, and g the acceleration of gravity. The numerical value of this product, in any particular case, will be the same whether measured by an American using feet and seconds or by a European using meters and seconds, and either of them might change from seconds to minutes without affecting his numerical result, if he changed his derived units of speed and of acceleration accordingly.

8 If there are n separate kinds of quantity but more than one quantity of each kind,—a number of lengths or a number of forces concerned in the relation to be described by the equation, all

[1]Physical Review, vol. 4, p. 345, October 1914.

the quantities of any one kind may be represented by specifying a single one of that kind and the ratios r', r'' . . . of the others to this one. Equation [1] then takes the form

$$F(Q_1, Q_2, \ldots Q_n, r', r'', \ldots) = 0 \qquad [4]$$

and this is always reducible to

$$f(\Pi_1, \Pi_2, \ldots \Pi_{n-k}, r', r'' \ldots) = 0 \qquad [5]$$

9 This theorem which, for short, I may call the Π theorem, is a convenient statement, for practical use, of the requirement of dimensional homogeneity. We may proceed at once to illustrate its meaning by application to the familiar problem of the flow of fluids through pipes.

THE FLOW OF LIQUIDS IN SMOOTH PIPES

10 When a liquid flows, at a constant rate, through a smooth straight pipe, the pressure gradient G may be expected to depend on diameter D, speed S, and density ρ and viscosity μ of the liquid. So long as the pipe is full and the liquid sensibly incompressible, we do not see anything else for G to depend on; and unless we have omitted some essential circumstance, these five quantities must be connected by some sort of relation which may be symbolized by writing

$$F(G, D, S, \rho, \mu) = 0 \qquad [6]$$

We shall proceed to apply the Π theorem to this equation and compare the results with observed facts.

11 There are 5 separate kinds of quantity involved in the relation, so that $n = 5$, but the units needed for measuring them can all be derived from $k = 3$ fundamental units, so that $n - k = 2$. Hence, whatever the nature of the relation may be, it must be reducible to the form

$$f(\Pi_1, \Pi_2) = 0 \qquad [7]$$

containing not five but only two independent variables. Furthermore, we know that any empirical form for [6] which cannot be expressed in the form [7] cannot be generally correct, no matter how good an approximation it may be over a limited range of experiments.

12 To find the two independent dimensionless products Π_1 and Π_2 we must first know the dimensions of the five kinds of quantity in terms of some three which we agree to regard as fundamental. We are not restricted to any particular three, because we are not now concerned with the question of preserving our units by con-

venient primary standards, but only with the interrelations of the five kinds of unit needed in our measurements. Any three mechanical units will serve our purpose if they are independent, i.e., if no one of them can be derived from the others. We adopt mass m, length l, and time t, because these are most commonly used. If we used any other three, such as force, length, and time, the results would be the same, only a little intermediate algebra would be different. We then have the following definitions and dimensional equations:

$$\left.\begin{array}{ll}
\text{Pressure gradient} = \dfrac{\text{force}}{\text{area}} \div \text{length} & G = m\, l^{-2}\, t^{-2} \\
\text{Diameter} = \text{a length} \ldots\ldots\ldots\ldots & D = l \\
\text{Speed} = \text{length} \div \text{time} \ldots\ldots\ldots\ldots & S = l\, t^{-1} \\
\text{Density} = \text{mass} \div \text{volume} \ldots\ldots\ldots & \rho = m\, l^{-3} \\
\text{Viscosity} = \dfrac{\text{force}}{\text{area}} \div \text{rate of shear.}\ . & \mu = m\, l^{-1}\, t^{-1}
\end{array}\right\} \quad . \qquad [8]$$

13 We select any three of these quantities, D, S, ρ, which are independent and might therefore, if we chose, be themselves used as fundamental units, and we write down the equation

$$\Pi_1 = D^x\, S^y\, \rho^z\, G \qquad [9]$$

We have then to determine the exponents x, y, z so that Π_1 shall have no dimensions, i.e., so that its unit shall be independent of the sizes of the arbitrary fundamental units m, l, t. This is very easy. Substituting in [9] from [8] we have for the dimensions of Π_1

$$\begin{aligned}\Pi_1 &= l^x . l^y\, t^{-y} . m^z\, l^{-3z} . m\, l^{-2}\, t^{-2} \\ &= l^{x+y-3z-2} . t^{-y-2} . m^{z+1}\end{aligned} \qquad [10]$$

For Π_1 to be dimensionless, the exponent of l which shows how the unit of Π_1 depends on the size of the length unit, must vanish; and similar conditions hold for time and mass. We therefore have the three equations

$$\left.\begin{array}{r} x+y-3z-2=0 \\ -y-2=0 \\ z+1=0 \end{array}\right\} \quad \text{whence} \quad \left\{\begin{array}{l} x=1 \\ y=-2 \\ z=-1 \end{array}\right.$$

and by substituting in [9] we have

$$\Pi_1 = \frac{D\, G}{\rho\, S^2} \qquad [11]$$

14 To find Π_2 we start with the equation

$$\Pi_2 = D^a S^b \rho^c \mu$$

and follow a similar procedure for determining the exponents a, b, c so that Π_2 shall be dimensionless. The result is

$$\Pi_2 = \frac{\mu}{D S \rho} \qquad [12]$$

and by equations [11] and [12], equation [7] finally takes the form

$$f\left(\frac{D G}{\rho S^2}, \frac{\mu}{D S \rho}\right) = 0 \qquad [13]$$

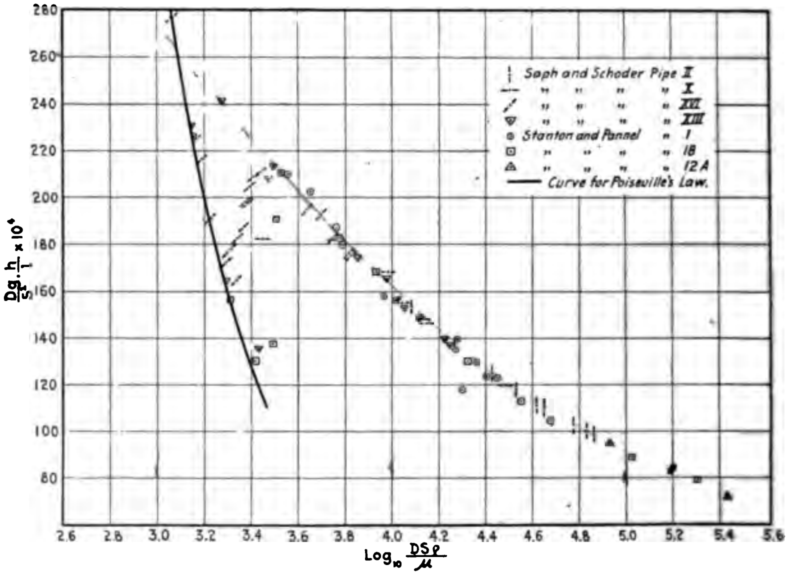

FIG. 1 RESULTS WITH SEAMLESS-DRAWN BRASS PIPE

15 The form of the function f remains to be found from experiment; but whatever it may turn out to be, we know that if there is any relation involving the five quantities of equation [6] and no others, it is such that the values of $\frac{D G}{\rho S^2}$ and $\frac{\mu}{D S \rho}$ are connected by a single relation, symbolized by [13], and that the size of either of them fixes that of the other,—as may be more conveniently expressed by imagining equation [13] to be solved for $\frac{D G}{\rho S^2}$ and written in the form

$$\frac{D\,G}{\rho\,S^2} = \varphi\left(\frac{D\,S\,\rho}{\mu}\right) \qquad [14]$$

16 We are now in a position to check results: for if observed values of $\frac{D\,G}{\rho\,S^2}$ are plotted against simultaneous values of $\frac{DS\rho}{\mu}$, the resulting points should lie along a single curve. If they do not do so, within the experimental errors, we shall know that some essential circumstance must have been overlooked in writing down our original equation [6].

17 Fig. 1 shows results obtained with seamless-drawn brass pipe of various diameters. The ordinate is $\frac{D\,G}{\rho\,S^2}\times 10^4$ or its equivalent $\frac{D\,g}{S^2}\cdot\frac{h}{l}\times 10^4$, where $\frac{h}{l}$ is the hydraulic gradient. The abscissa is $\log_{10}\frac{DS\rho}{\mu}$, which is better than $\frac{D\,S\,\rho}{\mu}$ itself which would make the diagram inconveniently long. Some of the points are from Saph and Schoder's[1] experiments on water. The others are from Stanton and Pannell's[2] experiments on both water and air. Out of the large number of series of experiments, a few were selected at random but so as to cover the whole range of diameters; and of these, only every fourth point was plotted except for $\log_{10}\frac{DS\rho}{\mu} < 3.5$ when all the points were plotted. If all the hundreds of points had been put in, the diagram would have had to be on a very large scale, but the conclusions to be drawn from it would have been unchanged.

18 It appears first, that the points of all three classes are mixed indiscriminately; and second, that the points are in fact distributed in a well defined band. Hence we conclude that nothing essential was omitted from equation [6].

19 The construction of this diagram illustrates one advantage of using dimensionless variables, namely, that it obviates the need of conversions from metric to English units, and vice versa. Saph and Schoder's data are published in English units and in reducing them, the values of $\frac{\mu}{\rho}$ were also expressed in English units. On the other hand, Stanton and Pannell's data are published in c.g.s. units:

[1] Am. Soc. Civ. Engineers; vol. 29, 1903, p. 419.
[2] Phil. Trans. Royal Soc. London: vol. A 214, 1914, p. 199.

but no conversion was needed before plotting, because the numerical value of a dimensionless quantity is independent of the size of the fundamental units so long as we keep the interrelations of the units unchanged, as we do in passing from c.g.s. units to normal English foot, second, pound mass units.

20 Since the object of this section is to illustrate the meaning of the Π theorem in a familiar problem, it would be out of place here to go into an extended discussion of the mathematical form of the function φ and the consequent form of the equation

$$\frac{h}{l}=\frac{S^2}{gD}\varphi\left(\frac{D\,S\,\rho}{\mu}\right)$$

and we must also pass over the more interesting subject of rough pipes and the modifications needed in the equations for treating them. We may, however, make three remarks before leaving the present subject:

First. The critical region, where stream line flow changes over to turbulent or hydraulic flow, is shown by the plot to occur at about $\log_{10}\frac{D\,S\,\rho}{\mu}=3.3$ to 3.4 or $\frac{D\,S\,\rho}{\mu}=2000$ to 2500, as found by Reynolds and others.

Second. If G is proportional to S, as we know experimentally that it is, below the critical speed, the unknown function φ must have the form $\varphi\left(\frac{D\,S\,\rho}{\mu}\right)=K\frac{\mu}{D\,S\,\rho}$, where K is a constant. Hence equation **[14]** reduces to

$$G=K\frac{S\,\mu}{D^2}$$

a form of Poiseuille's equation, showing that in this sort of motion the resistance is directly proportional to the viscosity and independent of the density.

Third. If, by reason of high speed, large diameter, or low viscosity, the motion becomes very turbulent, we know from observation that the resistance approaches proportionality to the square of the speed. The function $\varphi\left(\frac{D\,S\,\rho}{\mu}\right)$ must then degenerate into a mere constant K_1, and equation **[14]** gives us

$$G=K_1\frac{\rho\,S^2}{D}$$

the resistance being now sensibly proportional to the density and independent of the viscosity. This agrees perfectly with common sense. The tangential drag between two bodies of liquid moving past each other is merely an effect of cross transmission of momentum between them. When there are no eddies, this interchange of momentum occurs by mixing on a molecular scale, i.e., by diffusion. Hence in stream line motion the resistance depends directly on diffusion and the resulting viscosity. But if the motion is very turbulent, mixing occurs by eddies, and the amount of momentum carried by an eddy of given size depends on the density; the effects of diffusion and its consequence, viscosity, being of very minor importance. We also see why it is that temperature has so little effect on the resistance if the motion is turbulent. Temperature influences viscosity very much and density very little. Hence as turbulence increases and the resistance is more nearly proportional to S^2, the importance of temperature decreases because the importance of viscosity, the only thing sensibly dependent on temperature, decreases.

RESISTANCE OF IMMERSED BODIES AT MODERATE SPEEDS

21 We may next consider the motion of a completely immersed body such as an aeroplane, a dirigible balloon, or a submarine so deeply submerged as to cause no appreciable surface disturbance. Let us enquire how the forces between the fluid and the solid body depend on the various circumstances; and to be specific, let us consider the total head resistance R. The first question is : On what measurable quantities does R depend?

22 To start with, we have the relative speed S of the body and the undisturbed fluid at a distance; we shall suppose S to be constant so that there is no acceleration of the body. Next, we have the size and shape of the body and its orientation with regard to the line of motion: if D is a linear dimension of the body, the shape and orientation may be specified by the ratios r', r'', . . . etc., of a number of other lengths to the particular length D. . . . Finally, we have the mechanical properties of the fluid, its density ρ and viscosity μ. The effects of compressibility do not play any sensible part until the speed approaches that of sound in the medium, and at aeroplane speeds the air behaves very nearly as if incompressible. By moderate speeds we therefore mean speeds which are only a small fraction of the acoustic speed, and for such speeds, compressibility may be left out of account.

23 The condition of total immersion obviates the need to con-

sider either surface tension or the intensity of gravity. In problems of fluid motion, the ratio of viscosity to density appears very often, and it is convenient to represent it by a single symbol $\frac{\mu}{\rho} = \nu$. The quantity ν is known as the *kinematic viscosity* and we may specify the properties of the medium by ρ and ν instead of by ρ and μ as hitherto.

24 If we have not overlooked any important circumstance, there must be some definite quantitative relation connecting the resistance R with the other quantities enumerated, and it may be symbolized by the equation

$$F(R, D, S, \rho, \nu, r', r'', \ldots) = 0 \qquad [15]$$

which is analogous to equation [6] except that beside the $n = 5$ physical quantities R, D, S, ρ, ν, it contains also a number of dimensionless ratios r.

25 To this equation apply the Π theorem in its general form [5]. The quantities are all mechanical and $k = 3$ fundamental units are required for measuring them. Hence $n - k = 2$ and if such a relation as [15] subsists, it must be of or reducible to the form

$$f(\Pi_1, \Pi_2, r', r'', \ldots) = 0$$

As in the previous example, the Π's may be found by setting

$$\Pi_1 = D^x S^y \rho^z R; \quad \Pi_2 = D^a S^b \rho^c \nu$$

inserting the known dimensions of D, S, ρ, R, ν; and determining the unknown exponents x, y, z and a, b, c so as to make Π_1 and Π_2 dimensionless. It is unnecessary to give the very simple algebra of the solution: it suffices to note that R is a force and has the dimensions $R = m\, l\, t^{-2}$, while ν has the dimensions $\nu = l^2\, t^{-1}$. The result is

$$\Pi_1 = \frac{R}{\rho D^2 S^2}; \quad \Pi_2 = \frac{\nu}{DS}$$

and the Π theorem therefore tells us that if such a relation as [15] subsists, it must necessarily be reducible to the form

$$f\left(\frac{R}{\rho D^2 S^2}, \frac{\nu}{DS}, r', r'', \ldots\right) = 0 \qquad [16]$$

If this is solved for Π_1 it may be written

$$R = \rho D^2 S^2 \varphi\left(\frac{DS}{\nu}, r', r'', \ldots\right) \qquad [17]$$

in which the nature of the dependence expressed by the unknown function φ remains to be found by experiment if it has to be found at all.

26 In view of the infinite possibilities of varying a body's shape, there are, in the general case, an infinite number of the independent arguments r so that a general determination of the form of φ is impossible. Even if we restrict ourselves to comparatively simple shapes, there will usually be so many r's that the determination of the form of φ would be very laborious. We therefore cut the knot by limiting ourselves to the consideration of one shape and orientation at a time, in other words to studying a series of geometrically similar bodies, anyone of which may be regarded as an increased or diminished model of any other. The separate bodies now differ only in their size, specified by the value of D. The r's are the same for all, i.e., they are mere constants: hence they may be omitted from our equations, and [17] reduces, *for any series of geometrically similar bodies*, to the simpler form

$$R = \rho D^2 S^2 \Psi\left(\frac{DS}{\nu}\right) \qquad [18]$$

in which the unknown function Ψ has only the one argument $\frac{DS}{\nu}$.

27 We might now proceed to find the form of Ψ for bodies of the given shape by plotting observed values of $\frac{R}{\rho D^2 S^2}$ against simultaneous values of $\frac{DS}{\nu}$, drawing a curve, and representing it by an empirical equation. And it may be noted that while the results of experiments on bodies of any size and in any medium might all be utilized, no such variation of D and ν is at all necessary. For $\frac{DS}{\nu}$ may be given any value we please by varying S alone, so that the required information is obtainable from experiments on a single body of the series in a particular fluid.

28 But let us suppose that we are confronted, say, by the practical problem of finding, as economically and quickly as possible, the head resistance of a dirigible balloon of some new and untried shape but of given size and at some given speed. We must if possible avoid the labor of such a complete investigation as outlined above, and equation [18] shows us how this may be accomplished by means of model experiments.

29 The original being of length D_0 we may construct a small-

scale model of length D. Let S_0 be the speed of the original and let $S = S_0 \frac{D_0}{D}$. If the model is run at the speed S and the medium is air, as for the original, we shall then have

$$\frac{D\,S}{\nu} = \frac{D_0\,S_0}{\nu_0}, \qquad [19]$$

so that $\Psi\left(\frac{D\,S}{\nu}\right)$ may be treated as a mere constant. Hence dividing equation [18] for the original by the same equation for the model and setting $\rho = \rho_0$, we have

$$\frac{R_0}{R} = \left(\frac{D_0\,S_0}{D\,S}\right)^2 = 1 \qquad [20]$$

Speeds which are thus chosen so that the unknown function with which the dimensional reasoning leaves us degenerates into a constant, are called *corresponding speeds;* and at corresponding speeds the original and the model are said to be *dynamically similar.* In the present case corresponding speeds are inversely proportional to the linear dimensions, and at corresponding speeds the resistance of the original is equal to that of the model as shown by equation [20].

30 Upon reflexion we see that the foregoing result is of little or no value. For unless the speed S_0 were rather low, any great reduction of scale would require the corresponding speed S of the model to be impracticably high, perhaps even approaching the acoustic speed, so that our disregard of compressibility would no longer be legitimate. But we have still another possibility, namely that of using another medium and so changing $\frac{\nu}{\nu_0}$. The kinematic viscosity of water at ordinary temperatures is from 1/10 to 1/20 that of air according to the temperatures. If we take 1/15 and run the model not in air but in water, we have $\frac{\nu}{\nu_0} = \frac{1}{15}$ and equation [19] then gives us

$$\frac{S}{S_0} = \frac{1}{15}\,\frac{D_0}{D},$$

so that a model of given size need be run only 1/15 as fast as in air. If experiments are thus made in water, R_0 may be computed from the observed resistance R of the model by using the equation

$$\frac{R_0}{R} = \frac{\rho_0}{\rho}\left(\frac{D_0\,S_0}{D\,S}\right)^2 = 15^2\,\frac{\rho_0}{\rho}$$

or approximately

$$\frac{R_o}{R} = 0.28$$

31 While the foregoing method is strictly correct for speeds at which compressibility is negligible, the process may be much simplified when no high accuracy is required. For it has been established by experiment that with bodies of ordinary shapes at ordinary speeds, i.e., in cases where eddy resistance rather than skin friction is the important thing, the head resistance is very nearly proportional to the square of the speed. But if $R \propto S^2$, the function Ψ is no longer unknown: equation [18] shows that it is a constant, i.e., a mere shape factor for bodies of the given shape; and the influence of viscosity vanishes just as it does in the flow of liquids through pipes when there is so much turbulence that the resistance is proportional to the square of the speed. It then follows that

$$R = K \rho D^2 S^2 \qquad [21]$$

and all speeds of a body and its model are corresponding speeds, no matter what the scale ratio or the nature of the medium. All we need, therefore, is a determination of the shape factor K by an experiment on a model of any size, in any convenient medium, after which R may be found from equation [21] for the original. Experiments on very small models of dirigibles run in water have been used for determining the relative values of shape factors for various shapes.

32 It may be noted that although R in the equations has represented head resistance, it might equally well have represented the lift of an aeroplane or an inclined dirigible, or any other particular force on an immersed moving body, since we used in our reasoning only the dimensions of R, and all forces have the same dimensions. An equation of the form [18] is obtained in any case or, if $R \propto S^2$, an equation of the form [21], the shape factor K depending, of course, on what sort of force is under discussion.

RESISTANCE TO THE FLIGHT OF PROJECTILES

33 When the speed of an immersed body approaches or exceeds that of sound in the medium, as with modern projectiles, the compressibility may become a determining factor in the phenomenon, the energy lost by a high speed projectile being drained away principally in the head and base waves. Compressibility must therefore be recognized in any equations which are to describe what happens.

34 Compressibility and density, together, determine the acoustic speed in the medium, so that this speed C may be used, with the density ρ, in specifying the properties of the medium, instead of using the compressibility itself. Furthermore, since C is a quantity of the same kind as S, the speed of the body, we need not introduce both C and S into our equations but if S appears, we may represent C by its ratio to S, i.e., by $\frac{C}{S}$ or, if we prefer, by $\frac{S}{C}$. In place of equation [15] we now have

$$F\left(R, D, S, \rho, \nu, \frac{S}{C}, r', r'', \ldots\right)=0$$

35 The new variable $\frac{S}{C}$ being a mere ratio like the r's, the application of the Π theorem is precisely the same as before, and instead of equation [17] we have

$$R=\rho D^2 S^2 \varphi\left(\frac{DS}{\nu}, \frac{S}{C} r', r'', \ldots\right) \qquad [22]$$

To make any practical advance we must again confine our attention to projectiles of some one shape so as to get rid of the shape variables r, and we thus arrive at the equation

$$R=\rho D^2 S^2 \Psi\left(\frac{DS}{\nu}, \frac{S}{C}\right) \qquad [23]$$

36 We have here an instance in which model experiments in a single medium could not, a priori, be expected to furnish reliable information. In order to insure that a projectile and its model shall be dynamically similar, i.e., that the numerical value of the unknown function be the same for both, we must, until it is shown to be needless, make both $\frac{DS}{\nu}$ and $\frac{S}{C}$ constant. Now for a given medium under fixed conditions, the acoustic speed C is fixed; hence the speed S of the model must be the same as that of the original. But ν is also constant for the medium; and it follows that if S, ν, and $\frac{DS}{\nu}$ are to be constant, D also must be the same for the model as for the original. Therefore no change of scale is permissible and we cannot expect to get reliable information from a model. ✓

37 Even if we change the medium from air to water, the case is not much better. The speed of sound in water is about four times that in air, so that if the original were a 15-in. shell at a muzzle velocity of 2500 ft. per sec., a dynamically similar model would need to have a speed in water of about 10,000 ft. per sec.,—a rather formidable

requirement. Assuming this difficulty to be superable, and using water at a temperature such as to make its kinematic viscosity 1/15 of that of air under service conditions, we might then make the model dynamically similar to the original by reducing its diameter to 0.25 in. for we should then have $\frac{DS}{\nu} = \frac{D_o S_o}{\nu_o}$ as well as $\frac{S}{C} = \frac{S_o}{C_o}$. It is conceivable that if we could attain the required initial speed, experiments conducted in this way might furnish some interesting information, equation [23] reducing to [21] and the shape factor K being found from experiments on models to a scale of about 1/60.

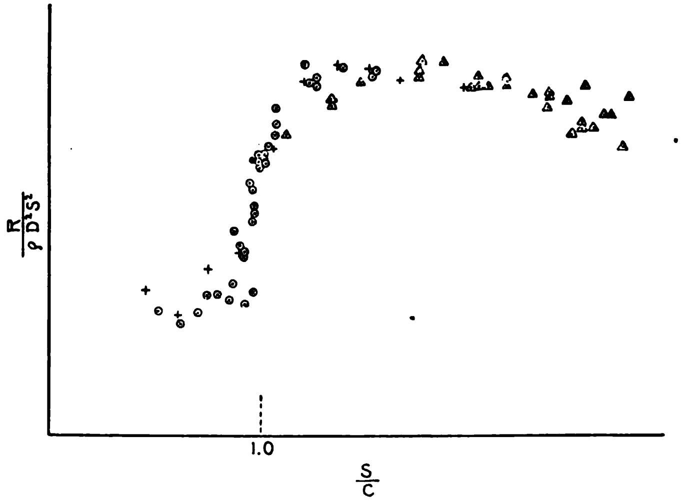

FIG. 2 RESISTANCE OF PROJECTILES

38 While the form of Ψ in [23] has not been accurately determined for any shape of projectile, we know that the influence of $\frac{S}{C}$ predominates over that of $\frac{DS}{\nu}$, as might have been expected in view of the turbulence of the motion. For if the observed values of the dimensionless variable $\frac{R}{\rho D^2 S^2}$ for projectiles of the older forms and of various calibers are plotted against values of $\frac{S}{C}$, the resulting points, while rather scattered, do lie more or less along a single curve, which proves that such variations of shape and of $\frac{DS}{\nu}$ as occurred, were of minor importance in comparison with the ratio of

the speed S to the acoustic speed C. Fig. 2 which is reproduced from Cranz's Text Book of Ballistics, exhibits a number of experimental results. The striking thing about the observations is their showing that the development of the wave system on passing the acoustic speed $\frac{S}{C}=1$ increases the resistance about three times, while both above and below this critical region $\frac{R}{\rho D^2 S^2}$ is fairly constant, i.e., the head pressure in air of given density is nearly proportional to the square of the speed.

SCREW PROPELLERS

39 The thrust T of a screw propeller of given shape may be supposed to depend on diameter D, rate of revolution n, and speed of advance S, and on the density ρ and kinematic viscosity ν of the water. If the screw were so deeply immersed that it caused no surface disturbance, these would seem to be all the quantities to be considered; but screws are not usually so deeply immersed as this and we must therefore bring in the weight of the water, i.e., the acceleration of gravity g. We may as well confine our attention, from the start, to propellers of a single shape, immersed to depths proportional to their diameters, and if we have not overlooked any essential factor in the action of the propeller we shall then have

$$F(T, D, n, S, \rho, \nu, g) = 0 \qquad [24]$$

40 For the measurement of these $n=7$ kinds of quantity, $k=3$ fundamental units are again required, so that $n-k=4$. The Π theorem therefore states that if equation [24] is complete, it must be reducible to the form

$$f(\Pi_1, \Pi_2, \Pi_3, \Pi_4) = 0 \qquad [25]$$

As usual, we select D, S, ρ as convenient independent quantities, and proceed as already illustrated, the only new kinds of quantity having the dimensions $n=t^{-1}$ and $g=l\,t^{-2}$

41 Writing

$$\Pi = D^x S^y \rho^z P,$$

where $P=T$, n, ν, and g successively, and finding the four sets of values which x, y, z must have in order to make Π dimensionless, we obtain the following expressions:

$$\left.\begin{aligned} \Pi_1 &= \frac{T}{\rho D^2 S^2}; \qquad & \Pi_3 &= \frac{\nu}{D S} \\ \Pi_2 &= \frac{D n}{S}; \qquad & \Pi_4 &= \frac{D g}{S^2} \end{aligned}\right\} \qquad [26]$$

Hence if an equation [24] subsists, it is certainly of such a nature that it is reducible to the form

$$f\left(\frac{T}{\rho D^2 S^2}, \frac{D n}{S}, \frac{\nu}{D S}, \frac{D g}{S^2}\right) = 0 \qquad [27]$$

Being interested more particularly in T, we may solve for Π_1 and write the equation

$$T = \rho\ D^2\ S^2\ \varphi\left(\frac{D n}{S}, \frac{D S}{\nu}, \frac{D g}{S^2}\right), \qquad [28]$$

in which φ is the usual unknown function which always remains after the application of dimensional reasoning, when the number of separate kinds of quantity n is greater than $k+1$.

42 We shall not discuss possibilities of determining the general form of φ, which in this instance has three independent arguments, but shall proceed at once to the question whether experiments on small-scale model propellers can be so carried out and interpreted as to furnish reliable information about full sized propellers of the same shape.

43 To ensure the model being dynamically similar to the original, or in other words, to make certain that φ shall have the same numerical value for both, we must make the three separate arguments of φ have the same values for both, unless it has already been proved that this is unnecessary. The first condition is very simple; for since $D\,n$ is proportional to the tip speed of the blades, $\frac{D n}{S}$ will be constant if the ratio of tip speed to speed of advance is constant. Since the two propellers are of the same shape, this means that corresponding elements of the blades must have the same angle of attack, or that the slip ratios must be the same. The first condition, therefore, for the dynamical similarity of two propellers of the same shape is that they shall be run at the same relative immersion and at the same slip ratio. When this condition is satisfied equation [28] reduces to

$$T = \rho\, D^2\, S^2\, \Psi\left(\frac{D S}{\nu}, \frac{D g}{S^2}\right) \qquad [29]$$

44 We now encounter a seemingly insuperable obstacle. In practice, we are limited to water so that ν remains nearly constant, and furthermore g is constant. Hence to attain exact

similarity we are directed to keep both $D\,S$ and $\frac{D}{S^2}$ constant, which means that neither D nor S can vary. It follows that no change of size is permissible, and two propellers of different diameters run in the same liquid cannot be made dynamically similar unless it turns out in practice that one of the two arguments of Ψ does not in fact have any sensible influence on the numerical value of Ψ.

45 We therefore cast about to see what may be done in making a justifiable approximation. Viscosity affects the motion of fluids mainly when the motion is quiet; if the motion is very turbulent, viscosity becomes of small or vanishing importance. Now the motion of the water about a screw propeller is excessively turbulent, and if we assume that it is therefore sensibly unaffected by viscosity, we may omit from equation [29] the argument $\frac{D\,S}{\nu}$ in which alone the viscosity appears. We thus get a still further simplification to the form

$$T = \rho\, D^2\, S^2\, \Psi_1\left(\frac{D\,g}{S^2}\right) \qquad [30]$$

46 The rest is easy: if the model of diameter D and original of diameter D_0 are run at *corresponding* speeds of advance, S and S_0, such that

$$\frac{D\,g}{S^2} = \frac{D_0\,g}{S^2_0},\ i.e.\ \frac{S}{S_0} = \left(\frac{D}{D_0}\right)^{\frac{1}{2}}, \qquad [31]$$

the numerical value of Ψ_1 will be the same for both. We then have, by equation [30],

$$\frac{T_0}{T} = \frac{\rho_0}{\rho}\left(\frac{D_0\,S_0}{D\,S}\right)^2;$$

or since $\rho = \rho_0$ we have, utilizing [31],

$$\frac{T_0}{T} = \left(\frac{D_0}{D}\right)^3 \qquad [32]$$

Equation [32] states that if geometrically similar propellers are run at the same relative immersion, at the same slip ratio, and at corresponding speeds as defined by [31], the thrusts will be proportional to the cubes of the diameters, if the effects of viscosity are unimportant.

47 The validity of this result evidently depends on how far we are justified in the approximation introduced by neglecting viscosity

and so stepping from equation [29] to equation [30]. It might appear, at first sight, that we were assuming the skin friction of the blades to be unimportant, but the assumption is by no means so violent as this. The only resistance to flow through straight pipes is skin friction, and we know that even for smooth pipes viscosity cuts very little figure in the resistance above the critical speed; while for rougher pipes or more turbulent motion, the influence of viscosity becomes quite negligible. The assumption made above amounts, therefore, only to assuming that if skin friction does play an important role in the operation of a screw propeller, the whole flow of water past the blades is so turbulent that skin friction is sensibly proportional to the square of the speed of the water over the blades. This assumption may not be quite accurate for slow speed propellers but will almost certainly be correct for higher speeds. The justification for the assumption must, of course, rest finally on experiment. So far as the writer knows—his information coming from Rear Admiral D. W. Taylor, U. S. N.—the application of equations [31] and [32], which amounts to using "Froude's law of comparison," does give correct results, though the available data are not very numerous.

48 For propellers so deeply immersed as to cause no surface disturbance, the weight of the water can have no effect, but only its inertia, so that the value of g can not influence the thrust. It follows that g cannot appear in the equations; $\Psi\left(\frac{D\,g}{S^2}\right)$ must be a mere constant shape factor; and [30] reduces to

$$T = K \rho D^2 S^2 \qquad [33]$$

Thus we conclude that for propellers of a given shape, running at the same slip ratio and sufficiently deeply immersed, there is no condition of corresponding speeds to be satisfied: the shape factor K may be found by a thrust measurement on the model, and the thrust of the original, running at any speed but with the same slip ratio, may then be computed from equation [33] and from the shape factor found for the model at the given slip ratio.

49 Numerous other points might be considered, in addition to the question how the foregoing reasoning, which evidently applies only to propellers advancing into still water, may have to be modified when the propeller is working behind a hull. One such point is cavitation. It is evident without any further algebra that the beginning of cavitation depends on the hydrostatic pressure, among

other things, so that even for deeply immersed propellers, the value of g must occur in the conditions for similarity as regards the cavitation point. When we come to the study of light-weight aero-propellers, we may have to run our propellers so that they are similarly distorted by thrust or by centrifugal force. Or the blade speeds may be so high that the compressibility of the air cannot be disregarded, and then we may have to run propellers of a given shape all at the same tip speed, in order to make them comparable. It is impossible to discuss these questions here but it is also needless because our object is not to discuss the theory of screw propellers but merely to illustrate the use of a method of reasoning.

50 In any case, we have first to ask what are all the physical quantities which may enter into the problem. If we are in doubt about any quantity, it is best to include it in our list and let it drop out later if it proves unimportant. The principle of dimensional homogeneity, in the convenient form of the Π theorem, tells us something about how these various quantities must appear in any correct equation connecting them. Upon examining the result we then see, by inspection, the conditions of dynamical similarity and can judge whether model experiments offer any prospect of advantage or whether they must involve the use of so many doubtful assumptions as to be untrustworthy until based on more complete and general experimental investigations of the phenomena in question.

HEAT TRANSMISSION

51 To avoid leaving one with the erroneous impression that the use of the principle of dimensional homogeneity is confined to the field of mechanics, it will be worth while to treat an illustrative problem which involves thermal quantities. In order not to run to intolerable lengths, we must introduce various simplifying restrictions and use very roughly approximate data as if they were exact; but if these limitations are clearly recognized they will not impair the illustrative value of the treatment.

52 Let us suppose that a homogeneous fluid such as air, water, or superheated steam, is flowing through some metallic apparatus which is hotter or colder than the fluid by an average amount Δ^0. The apparatus and the fluid might, for instance, be a surface condenser and the water in its tubes; or a nest of steam or brine coils and the air passing over them; or simply a straight pipe with hot water running through it. A certain amount of heat H will be transmitted per unit time between the metal and the fluid in ac-

cordance with the temperature head Δ, and we wish to examine the question what this rate of heat transmission depends on and how.

53 The first condition we impose is that the speed of flow S measured, say, at the inlet, shall be great enough that the fluid motion is everywhere very turbulent. We may then regard the motion of the fluid and its convective action as not dependent on viscosity. On the other hand, the speed shall nowhere be so great as to oblige us to introduce compressibility into our equations. We thus exclude from consideration the transmission of heat between a steam-turbine nozzle and its steam jet, but we do not exclude any ordinary practical case. Under the foregoing conditions, the density ρ of the fluid is the only one of its mechanical properties that concerns us.

54 The other properties of the fluid on which H may be supposed to depend are specific heat C, which determines the convective effect of the motion of a given volume, and the thermal conductivity λ, which determines the facility with which heat can pass through the nearly stagnant film which always sticks to the metal surfaces. The greater the speed, in a given apparatus, the more effective will convection be and the more important the specific heat; while simultaneously, the increased scouring action decreases the thickness of the quasi-stagnant film and so decreases the relative importance of the conductivity of the fluid. So long as we have clean metal surfaces we may usually disregard the resistance of the solid parts of the apparatus; but in any event the temperature head Δ now refers to the temperature difference between the fluid and the *surface* of the metal or other solid, either between two specified points or as an average for the whole apparatus, so that the nature and properties of the solid walls do not interest us.

55 We next agree to limit ourselves to consideration of moderate temperature heads such as 200 deg. fahr. or less, within ordinary temperature ranges,—say 0 deg. to 500 deg. fahr. This permits us to introduce several approximations. In the first place, we may disregard variations of the properties of the fluid with temperature, and treat ρ, C, and λ as constants, using average values at the mean temperature. In the second place, we may treat the heat transmission H as dependent only on the difference of temperatures and not directly on the temperatures themselves. And finally, we may without serious error, disregard direct radiation, which might not be legitimate if we were treating of flame in boiler tubes or other instances in which one of the temperatures was very high and the other was not.

56 In spite of these various limitations, it appears upon reflexion that we may still expect our treatment to be approximately correct for the great majority of heat transmitting devices under their ordinary working conditions.

57 The only things not yet mentioned which may be expected to influence H, are shape and size of the apparatus. If we confine our attention to an apparatus of one particular design, the only quantity needed for specifying it is some linear dimension D; and if we have not overlooked anything, there is an equation

$$F\,(H, \Delta, D, S, \rho, C, \lambda) = 0 \tag{34}$$

to which we may apply the Π theorem as soon as we know the dimensions of the separate kinds of quantity involved.

58 Three fundamental units are not now sufficient because, in addition to purely mechanical quantities, we now have also thermal quantities to deal with and require a thermal fundamental unit. The most convenient is temperature, which will be denoted by θ and the units for all the $n = 7$ kinds of quantity may be derived from the $k = 4$ fundamental units $[m, l, t, \theta]$. The Π theorem therefore tells us that equation [34] must be reducible to the form

$$f\,(\Pi_1, \Pi_2, \Pi_3) = 0 \tag{35}$$

59 To determine the forms of the Π's we need the new dimensional equations

$$H = (m\, l^2\, t^{-3})^1 \qquad C = l^2\, t^{-2}\, \theta^{-1},$$
$$\Delta = \theta, \qquad \lambda = m\, l\, t^{-3}\, \theta^{-1},$$

and if we select D, S, ρ, Δ as the quantities for which exponents are to be found, the result of proceeding in the usual manner is to give us, for equation [35], the more specific expression

$$f\left(\frac{H}{\rho\, D^2\, S^3}, \frac{C\,\Delta}{S^2}, \frac{\lambda\,\Delta}{D\,\rho\, S^3}\right) = 0,$$

or after solving for H, in which we are particularly interested,

$$H = \rho\, D^2\, S^3\, \varphi\left(\frac{C\,\Delta}{S^2}, \frac{\lambda\,\Delta}{D\,\rho\, S^3}\right) \tag{36}$$

60 This is as far as we can go by dimensional reasoning alone, and we must next refer to experimental data, if we can find any suitable, in order to get some information about the form of the unknown function φ. In a new experimental investigation we

[1]A rate of heat flow is merely an amount of power; e.g., 2543 B.t.u. per hour = 1 horse power.

should, of course wish to vary the two quantities $\frac{C\,\Delta}{S^2}$ and $\frac{\lambda\,\Delta}{D\,\rho\,S^3}$ separately. But the quantities S and Δ, which are the easiest to vary in practice, occur in both; while ρ, C and λ, being properties of the fluid, cannot be varied arbitrarily; and changing D requires the use of a larger or smaller model of the apparatus under investigation. What is true of future experiments is true of the past; we must expect to find that the most accurate and most numerous data available refer to the dependence of H on S and Δ.

61 Equation **[36]** is therefore not in proper form for our present purpose, but it may readily be transformed so that either S or Δ shall appear in only one of the arguments of φ. Starting with Δ, we proceed as follows: if

$$\frac{C\,\Delta}{S^2} = \Pi_1 \quad \text{and} \quad \frac{\lambda\,\Delta}{D\,\rho\,S^3} = \Pi_2,$$

we have

$$\Delta = \frac{S^2}{C}\Pi_1 \quad \text{and} \quad \Pi_2 = \frac{\lambda}{DS\rho C}\Pi_1,$$

whence we may write

$$\varphi\left(\Pi_1, \Pi_2\right) = \varphi\left(\Pi_1, \frac{\lambda}{DS\rho C}\Pi_1\right)$$

But if the value of a certain quantity is determined by the values of two other quantities x and y, it may equally well be specified by stating the values of x and $\frac{y}{x}$; for x and $\frac{y}{x}$ together fix both x and y. To put it in another way, any function $\varphi\ (x, y)$ may also be described as some other function $\Psi\ (x, \frac{y}{x})$. Accordingly we may replace the unknown function $\varphi\ (\Pi_1, \Pi_2)$ of equation **[36]** by $\Psi\left(\Pi_1, \frac{\lambda}{DS\rho C}\right)$ and so get the equivalent equation

$$H = \rho\, D^2\, S^3\, \Psi\left(\frac{C\,\Delta}{S^2}, \frac{\lambda}{DS\rho C}\right) \qquad \textbf{[37]}$$

in which Δ appears in only one argument of the unknown function Ψ.

62 We are now able to refer to experimental facts, and the first we shall use is that when everything else is kept unchanged, the rate of heat transmission in any given apparatus is approximately proportional to the temperature head, when that head is not very large. Treating this as if it were an exact relation, we conclude that in equation [37], Ψ must contain Δ as a factor and may therefore be written

$$\Psi\left(\frac{C\,\Delta}{S^2}, \frac{\lambda}{D\,S\,\rho\,C}\right) = \frac{C\,\Delta}{S^2}\,\Psi_1\left(\frac{\lambda}{D\,S\,\rho\,C}\right),$$

in which the new unknown function Ψ_1 has only a single argument. Equation [37] thus reduces to

$$H = D^2\,S\,\Delta\,\rho\,C\,\Psi_1\left(\frac{\lambda}{D\,S\,\rho\,C}\right) \qquad [38]$$

63 To proceed further by experiment we might change the fluid, thus altering ρ, C, and λ simultaneously; or we might change D by using an apparatus of the same shape but different size; but the most obvious thing to do is to try the effect on H of varying the speed of flow S, while keeping everything else constant. However, even without further experiment we may feel nearly certain that if the flow is very turbulent and the scouring action on the surfaces sufficiently violent, the effect of conductivity λ on H must be small compared with that of specific heat, and that where λ appears in the equation for H it can only be in terms with small exponents. As a rough approximation, we may disregard conductivity altogether and if we do so we have

$$\Psi_1\left(\frac{\lambda}{D\,S\,\rho\,C}\right) = A = \text{constant},$$

so that equation [38] reduces to

$$H = A\;D^2\,S\,\Delta\,\rho\,C, \qquad [39]$$

in which A is a characteristic constant shape factor for apparatus of the given design.

64 To show that the above approximation is not a wild assumption we may refer to the fact that Nusselt,[1] working with compressed gases in round pipes found H nearly proportional to $S^{0\cdot8}$, while Stanton,[2] using liquids, found that H was proportional to a power of S which was a little less than unity,—equation [39] indicating that the exponent should be exactly unity. Nusselt's result would require that [38] should have the form

$$H = A\;D^{1\cdot8}\,S^{0\cdot8}\,\Delta\,\rho^{0\cdot8}\,C^{0\cdot8}\,\lambda^{0\cdot2},$$

in which the effect of conductivity is still perceptible though small,

[1]Zeitschrift des Vereines deutscher Ingenieure *53*, pp. 1750 and 1808; 1909.
[2]Phil. Trans. Royal Soc. London; *A190*, p. 67; 1897.

while that of specific heat is slightly less than on our former simple assumption that the effect of was negligible. Stanton's result would cause still less change in equation [39].

65 In the present confused state of the subject of heat transmission we have been obliged, for the sake of completing our illustration, to make use of several admittedly rough approximations. But though equation [39], thus obtained, is certainly not exact, it is probably for most ordinary heat transmitting devices nearly enough correct to be worth a little further physical interpretation. If we notice that the whole mass of fluid M, which passes per second, is proportional to the product of cross section, speed, and density, i.e., to $D^2 S \rho$, we have by [39]

$$\frac{H}{\Delta} = B\,M\,C$$

in which B is a constant shape factor; or in words: the rate of heat transmission per degree temperature head, in an apparatus of given design and with a given fluid, is directly proportional to the mass flow M. Or if the fluid is not always the same, $\frac{H}{\Delta}$ is proportional to the total thermal capacity $M\,C$ of the fluid which passes through the apparatus in unit time.

66 Another way of putting it is, that since $\frac{H}{M\,C}$ is the amount by which temperature of the fluid changes while flowing through the apparatus, and since $\frac{H}{M\,C\Delta} = B =$ constant, the fluid will change its temperature by the same fraction of the temperature difference Δ, regardless of its speed, its nature, and the size of the apparatus, provided only that the shape factor B is constant. This rather startling statement is, at least, not altogether contradicted by experience. For it is known that by forced draught the steaming capacity of a boiler may be greatly increased without much increase in the percentage of heat lost in the flue gases, and this shows that with the increased speed through the flues, the gases have still fallen in temperature by about the same number of degrees, i.e., by about the same fraction of the difference Δ between their mean temperature and that of the tubes, which remains nearly constant.

67 Since all the above relations are only rough approximations, there is no occasion to go farther with them here. But it is hoped that enough has been said to show that the application of dimen-

sional reasoning to the planning of experiments and the interpretation of observed data may sometimes be well worth while in view of the fact that the only labor involved, is at most, the solution of a few simple linear algebraic equations.

CONCLUSION

68 It would be an easy matter to extend this paper indefinitely by treating other problems as, for instance, the static strength of built up structures, the mechanical efficiency of engines or transmission mechanisms, the flow of water in open channels, or the heating of electric generators. The intention has been to select for treatment illustrative examples which had some intrinsic interest and where some tangible result might be obtained. In many cases, the application of the principle of dimensional homogeneity leads to results which are worthless because they merely suggest that we ought to do something which we know is impracticable. But to offset this chance of failure there is the fact that whatever information or suggestions we do get are free, because the application of the theorem is so very simple.

69 The method is not a *theoretical* one in the ordinary sense,—there is nothing hypothetical about it. It is purely algebraic and tells us with certainty, that *if* certain quantities and no others are connected by a physical relation, the equation which describes the relation must be reducible to a certain form: the only chance of mistake is in overlooking some essential factor in the problem. Since the process of reasoning is purely mathematical, we cannot, of course, get out at the end any more than we put in at the beginning when we use our physical common-sense and experience to write down the original list of variables for the problem in hand. But we get out what we put in in a form which often makes it much more available than when it went in.

70 Finally, it may be stated again that the foregoing developments make no claim to essential newness, the purpose having been to call attention to and possibly arouse interest in a very useful kind of reasoning which is by no means so familiarly used as it deserves to be.

APPENDIX

THE DEDUCTION OF THE Π THEOREM

I. Let $Q_1, Q_2, \ldots Q_n$ be a number of physical quantities of different kinds, e.g., a length, a force, a density, etc., which are involved in some physical phenomenon such as the operation of a machine under its working conditions. If these n quantities suffice for describing the phenomenon completely, the value of any one of them is completely determined when the values of the others are given, and this mutual dependence may be stated symbolically by writing the equation

$$F(Q_1, Q_2, \ldots Q_n) = 0 \qquad [1]$$

An equation of this sort, containing symbols which stand for measured numerical values of physical quantities, is a physical equation.

The equations used in engineering are of two kinds. They may be *theoretical*, i.e., based on general principles, like the equation for the energy of a fly wheel or St. Venant's equation for the flow of gases; or they may be *empirical*, i.e., deduced directly from experiments on some particular machine or phenomenon without regard to anything else; formulas for the windage of fly wheels or the loss of head in water mains are empirical equations. But in either case, unless they are mere mathematical formulas such as those of trigonometry, the equations are physical equations and are amenable to the same rules as all other physical equations.

II. Any complete physical equation has the general form

$$\Sigma\left(M\, Q^a_1\, Q^b_2 \ldots Q^n_n\right) = 0 \qquad [2]$$

in which the Σ represents summation of a number of terms of the form indicated. The exponents $a, b, \ldots n$ are constants for each term, though they may differ from term to term. There may be any number of terms but the coefficient M of each term is a pure or *dimensionless* number—such as π or $\sqrt{2}$—the value of which does not depend on the sizes of the units used in measuring the Q's, so long as the interrelations of the units among themselves are unchanged.

No purely arithmetical operator such as *log* or *sin* can be applied to an operand which is not a pure number. When such expressions appear to occur in physical equations, as they often do, it is always found on closer examination that the things operated on are in fact only ratios or other dimensionless quantities. The results of the indicated operations are therefore themselves dimensionless numbers independent of the sizes of the units of the Q's, and they may be included in the dimensionless coefficients M.

III. Upon dividing equation [2] through by any one term, we have

$$\Sigma\left(N\, Q^{\alpha}_1\, Q^{\beta}_2 \ldots Q^{\nu}_n\right) + 1 = 0 \qquad [3]$$

Now all the terms of any physical equation must have the same dimensions, and the coefficients N have no dimensions because they are merely ratios of the dimensionless coefficients M. It follows that the exponents of each term of equation [3] must have some such set of values that a dimensional equation of the form

$$[Q^{\alpha}{}_1 Q^{\beta}{}_2 \ . \ . \ . \ Q^{\nu}{}_n] = [1] \tag{4}$$

is satisfied when the known dimensions of the Q's are inserted.

Let Π_1 represent any dimensionless product of powers of the Q's of the form indicated by (4); and let Π_2, Π_3, . . . Π_i be all the other such expressions which can be made up independently by using different sets of values for the exponents. Then the expression $(\Pi_1^{x_1}\Pi_2^{x_2} \ . \ . \ . \Pi_i^{x_i})$ is also dimensionless, no matter what the exponents x may be. Consequently equation [3] will satisfy the dimensional requirement of having all its terms of the same dimensions—of zero dimensions in this case—if it has the form

$$\Sigma N \Pi_1^{x_1}\Pi_2^{x_2} \ . \ . \ . \ \Pi_i^{x_i}+1=0 \tag{5}$$

Since the number of terms, the values of the coefficients N, and the values of the exponents x may be anything whatever without affecting the dimensions of any term, the first member of equation [5] is merely some entirely indeterminate function of the Π's. Hence the most general and unrestricted form which the physical equation [1] can have, subject only to the requirement of dimensional homogeneity, is

$$f(\Pi_1, \Pi_2, \ . \ . \ . \ \Pi_i)=0 \tag{6}$$

in which f represents some completely unknown function of which the form remains to be found, either empirically by direct experiment, or theoretically from such general physical principles as may be applicable. In more ordinary language, this means that the statement that a number of physical quantities Q are mutually related as symbolized by equation [1], is equivalent to the statement that all the independent dimensionless products Π which can be formed from the Q's, are also mutually related in some definite manner symbolized by equation [6].

To illustrate what is meant by *independent* dimensionless products we may consider the two expressions

$$\Pi_1 = \frac{D\,G}{\rho\,S^2} = D\,G\,\rho^{-1}S^{-2}$$

$$\Pi_2 = \frac{\mu}{D\,\rho\,S} = \mu\,D^{-1}\rho^{-1}S^{-1}$$

which occur in section 3, on the flow of liquids. In the first place, if we raise either of these to any power, we get a new dimensionless quantity; e.g., $\left(\frac{D\,S\,\rho}{\mu}\right)^2$ has no dimensions; but it is not *independent* of Π_2 because its numerical value is fixed when that of Π_2 is given. Furthermore, any such combination as

$$\Pi_1\Pi^{-2}{}_2 = \frac{D\,G}{\rho\,S^2}\times\left(\frac{D\,S\,\rho}{\mu}\right)^2 = \frac{D^3\,G\,\rho}{\mu^2}$$

is a new dimensionless product, but it is not independent of Π_1 and Π_2 because its value is fixed by theirs. On the other hand, Π_1 and Π_2 are themselves independent because neither can be obtained from the other.

IV. To measure n kinds of quantity we require n units, but these need not all be adopted arbitrarily for they can in general be derived from, i.e., described or defined in terms of, some smaller number of fundamental units. Let k be the number of fundamental units required for fixing the n kinds of unit needed in measuring the quantities Q_1, Q_2, . . . Q_n. In mechanics all the necessary units can be derived from only three, such as force, length, time, or work, speed, density. Even in the most general problems, dealing with thermal and electromagnetic as well as mechanical quantities k is never greater than five.

If k is the number of fundamental units needed, there is always, among all the n units, at least one set of k which are independent of one another and might themselves be used as fundamental units if we paid no attention to the question of representing and preserving the fundamental units by satisfactory primary standards. Let $[Q_1, Q_2, \ldots Q_k]$ represent such a set and let the remaining $n-k$ units be denoted by $[P_1, P_2, \ldots P_{n-k}.]$ Then each of the P's may be derived from the Q's in accordance with a dimensional equation which may be written

$$[Q^{\alpha}{}_1 Q^{\beta}{}_2 \ldots . Q^{\kappa}{}_k . P] = [1] \tag{7}$$

Since there are $n-k$ of the P's, there are $n-k$ separate independent equations of the form [7], and no more.

If in equation [7] we substitute for P and the Q's their dimensional equivalents in terms of any convenient fundamental units (see equations [9] to [11] of Section 3 in the body of the paper), the requirement that the total exponent of each fundamental unit shall vanish furnishes k independent linear equations which suffice for the determination of the exponents $\alpha, \beta, \ldots \kappa$ of equation [7]. If, after determining these exponents for any particular P, we set

$$\Pi = Q^{\alpha}{}_1 Q^{\beta}{}_2 \ldots . Q^{\kappa}{}_k P \tag{8}$$

the quantity Π satisfies the requirement of being a dimensionless product of the specified form. There are $n-k$ independent equations of the form [7] and therefore the same number of Π's, hence $i = n-k$, and the number of *independent* Π's is k less than the whole number of different kinds of quantity Q.

V. We have hitherto confined our attention to a relation among quantities that are all of different kinds. If several quantities of any one kind are involved in the relation to be described, they may all be specified by the value of any one and the ratios r', r'', etc., of the others to that one. Dimensional considerations cannot tell us anything about the manner in which these dimensionless ratios r appear in the equation which describes the relation, but their possible influence must be indicated, and this may be done in an entirely general way by introducing them as additional independent arguments of the unknown function f. The limitation imposed by the requirement of dimensional homogeneity upon the possible forms of physical equations may therefore be conveniently summarized in the following statement:

Any complete physical equation which describes a relation subsisting among quantities of n different kinds, of which k kinds are independent and not derivable from one another, is reducible to the form

$$f(\Pi_1, \Pi_2, \ldots \Pi_{n-k}, r', r'', \ldots) = 0 \tag{9}$$

in which the r's represent all the independent ratios of quantities of the same kind, and each Π is determinable from a dimensional equation of the form

$$[\Pi] = [Q^{\alpha}{}_1 Q^{\beta}{}_2 \ldots . Q^{\kappa}{}_k . P] = [1] \tag{10}$$

VI. If equation [9] is solved for any one of the Π's, for instance Π_1, it may be written

$$P_1 = Q^{a}{}_1 Q^{b}{}_2 \ldots . Q^{k}{}_k \varphi(\Pi_2, \Pi_3, \ldots . \Pi_{n-k}, r', r'', \ldots) \tag{11}$$

in which

$$a = -\alpha_1, \; b = -\beta_1, \text{ etc.}$$

If it is desired to obtain an equation of the form [11] with a particular quantity P_1 appearing separately and in the first member only, this quantity should, from the outset, be excluded from the list of quantities which are to be used as the Q's in handling the $n-k$ equations of the form [10]. This precaution is not always necessary, but it is always *sufficient* to ensure that the quantity in question shall appear in only a single one of the Π's and shall therefore be separable.

Equation [9] may also, of course, be put into the form

$$r' = \Psi\,(\Pi_1, \Pi_2, \;.\;.\;.\; \Pi_{n-k}, r'', r''', \;.\;.\;.\;)$$

which is sometimes useful.

VII. Since equation [11] contains an unknown function φ, the form of which cannot be found by dimensional reasoning, the equation does not give us any definite information in the general case when all the quantities which appear in the second member are allowed to vary independently. But if all the r's are held constant, and if the Q's and $P_2, P_3, \;.\;.\;.\; P_{n-k}$ are allowed to vary, not arbitrarily but only in such ways as will keep the values of $\Pi_2, \Pi_3, \;.\;.\;.\; \Pi_{n-k}$ constant, then we do have a definite statement of the dependence of P_1 on the Q's. For under these conditions, although its general form remains unknown the function φ degenerates into a dimensionless constant N, because all its arguments are constant. Hence under these conditions equation [11] assumes the definite form

$$P_1 = N\, Q^{a}{}_1\, Q^{b}{}_2 \;.\;.\;.\; Q^{k}{}_k \qquad [12]$$

A single experimental measurement of a set of simultaneous values of P_1, and the Q's suffices to determine the numerical value of N; and by equation [12] with this experimental value of N, the value of P_1 may be computed, without further experiment, for any other values of the Q's which satisfy the requirement, noted above, of keeping $\Pi_2, \Pi_3, \;.\;.\;.\; \Pi_{n-k}$ and the r's constant.

The chief utility of the principle of dimensional homogeneity is found in its application to problems in which it is practicable to arrange matters so that the r's and Π's of equation [11] shall remain constant and the definite equation [12] therefore be satisfied. This is what has to be done when we use model experiments for getting information about the behavior of the full sized originals, and the practicability or impracticability of satisfying the required conditions (which is evident upon inspection of the list of Π's and r's) is what determines whether we can or cannot obtain reliable information from models.

DISCUSSION

M. D. Hersey (written). The author has struck the keynote of a new development of technical physics, which will eventually play the same part in mechanical engineering that physical chemistry has begun to play in the chemical industries.

The importance of technical physics, as a branch of subject matter, is already so clearly recognized in Germany that laboratories are being established devoted exclusively to this field. But the development which I think we may now anticipate is something distinct from this, and a natural sequel to it: I refer to the development of technical physics, not as a branch of subject matter, but as a method of reasoning.

It is from such a standpoint that technical physics becomes analogous to physical chemistry which is the planning and interpretation of chemical experiments in the light of thermodynamics and the phase rule; and the Π-theorem is closely analogous to thermodynamics and the phase rule. Thermodynamics affords certain rigid connecting links between seemingly isolated experimental results, while the phase rule tells us the number of degrees of freedom of a chemical system. The Π-theorem likewise affords rigid connecting links which not only serve as a check on the consistency of our results, but may greatly cut down the labor of experimenting. Thus, on applying the Π-theorem to lubrication, we find, under certain conditions, that the coefficient of friction, f, must be some function or other of the *two* variables $\mu n/p$ and $D^3 n/Q$ alone; in which μ denotes viscosity, n revolutions per unit time, p bearing pressure, D journal diameter, and Q volume of oil pumped through bearing in unit time. Hence, the same change in f will be produced by a given change in the argument $\mu n/p$, whether this change is, in turn, caused by varying μ in one direction or by varying p in the other direction; and so on. These facts, all implicitly contained in the Π-theorem, can, for the sake of emphasizing our analogy, be expressed by the equations

$$\frac{df}{d\mu/\mu} = -\frac{df}{dp/p} \text{ whence } \frac{df}{dp} = -\frac{\mu}{p}\cdot\frac{df}{d\mu}$$

$$\frac{df}{dD^3/D^3} = -\frac{df}{dQ/Q} \text{ whence } \frac{df}{dD} = -3\frac{Q}{D}\frac{df}{dQ}; \text{ etc.}$$

And, just as the phase rule tells us that the number of degrees of freedom in a chemical system is $F = K - P + 2$, K being the number of components which coexist in P phases, so also the Π-theorem tells us that the number of degrees of freedom in a physical system is $f = p - k - 1$, k being the number of fundamental units needed to describe a relation subsisting among the p physical quantities. For it diminishes by k the number of factors which have to be varied experimentally.

The author has himself stated that the paper contains nothing essentially new. Any illusion to the contrary would be an impediment to the successful use of the methods presented. The kernel of the paper is a theorem which is merely a restatement of the requirements of dimensional homogeneity, announced by Fourier nearly a hundred years ago, and extensively used by Rayleigh and others. But the fact that the paper contains nothing essentially new does not diminish its value. Gibbs' phase rule, too, was new only in form, not in substance, yet it served as the crystallizing influence which caused an immense number of latent ideas to fall into line, and we may expect the Π-theorem to play a similar rôle.

This inevitable development of technical physics into a unified branch of science, which will acquire the same fundamental place in the engineering curriculum that physical chemistry now holds in the chemical curriculum, can be facilitated if writers on the problems of hydro- and aerodynamics, heat transmission and the like will be as introspective as possible, explicitly calling attention not only to their results, but to their methods of reasoning as well. For in every successful artifice of reasoning, there must be some element which is universal and capable of being generalized and made into a working tool.

Melach I. Nusim (written). The conditions for similarity, discussed in the paper, have been noted and applied with great advantage by designers of centrifugal compressors and pumps. Two centrifugal compressors, if they are geometrically similar, have the same efficiency provided the following relation is maintained between the rated flow, Q, the peripheral speed of the impeller, S, and the impeller diameter, D:

$$Q_1/S_1D_1^2 = Q_2/S_2D_2^2 = \text{constant}$$

The experimental data on one particular size of compressor can be utilized to predict with accuracy the performance of a number of sets, provided the compressors are made geometrically similar and rated according to the relation mentioned above. In terms of the mean effective pressure, P, the r.p.m., N, and the flow, Q (volume per unit of time), the relation is equivalent to

$$QN^2/P^{\frac{3}{2}} = \text{constant}$$

A. R. Dodge said he had investigated the drop in pressure of superheated steam under similar conditions to those of Stanton's and Pannel's experiments on air and water illustrated in Fig. 1, using a 1-in. smooth drawn brass pipe, steam jacketed. Over the range covered (between ordinates of 3.8 to 5.6) the results coincided exactly with those shown in this curve, showing that the curve applies for steam, in addition to air, water and oil. It ought, therefore, to apply for any fluid, and simplify existing formulae for pressure drop.

He had also made a number of steam tests with commercial iron pipe 2 to 8 in. in diameter, and found that the curves were parallel to and above that shown in Fig. 1, the distance above depending on the roughness of the pipe.

John R. Freeman said that he possessed data of his own experiments with ordinary rough pipes, which might be of value in solving the problems presented. He said he had conducted a very extended range of experiments, in 1893, on pipes of all degrees of smoothness, from seamless brass pipe to exceedingly rough pipe.

L. W. Wallace said that in connection with investigations to determine certain facts in reference to locomotive sparks, it became necessary to know to what height sparks are ejected from locomotives under various operating conditions, and he asked whether experiments with a specially designed model locomotive would give data that would be comparable with the actual height the sparks would be thrown, the size of the sparks, etc.

The Author. In reply to Mr. Wallace, it is conceivable that model experiments might be so devised as to furnish the desired information, but the difficulties appear, at first sight, rather formidable. It is impossible to say off-hand, before examining the problem carefully, whether an attempt to solve it in this way would have any prospect of success. It would seem much simpler to study the actual emission of sparks from a locomotive by making runs at night and taking photographs—possibly kinematograph records—from two points on the train, one close to the locomotive and one much farther back.

In reply to Mr. Freeman, while Stanton and Pannell's experiments on smooth brass pipes were possibly somewhat more accurate than Saph and Schoder's, the latter had also worked with galvanized pipes. In trying to get an equation which could be made to represent the resistance of both smooth and rough pipes by varying only a single quantity—representing the roughness—the author had therefore used Saph and Schoder's results exclusively, because in a preliminary study consistency was more important than accuracy. He had found, however, that the data were not sufficient for his purpose, and it would be a matter of great interest to him if he were priviledged to examine Mr. Freeman's experimental results.

In reply to Mr. Dodge, it must be a satisfaction to all concerned with the subject of pipe resistance that the results of his wide experience with steam agrees so well with those obtained by Saph and Schoder for water, and by Stanton and Pannell for both water and air. We may safely conclude that the basis of physical ideas from which the dimensional treatment starts is sensibly correct, and that no important element in the problem has been overlooked. The results obtained by dimensional reasoning are so instructive and the problem of pipe resistance so important, that the confirmation which Mr. Dodge has given is a valuable contribution to the subject.

The example, brought forward by Mr. Nusim, of the practical utilization of the notion of dynamical similarity is very interesting. The author's object in presenting his paper was to call attention to the method which, he is convinced, will in time come to be one of the engineer's handy tools, like the two laws of thermodynamics. But since he is aware that his opinion of the value of the method may be received somewhat sceptically by professional engineers, testimony in its favor from one engaged in practical designing work is doubly welcome.

In reply to Mr. Hersey, mechanical engineering is an art, not a science; and the ability and imagination of the individual engineer will always be its most important element. But common sense tells the engineer to get as much outside help as he can in solving his problems, and one source of such help is physics. As Mr. Hersey points out, the aid to be got from physics does not consist merely in new determinations of physical constants or in experimental investigations of physical problems which are of special interest to engineers. It consists also in the systematic use of the scientific method of physics in analyzing problems, planning experiments, and coördinating known facts so as to bring to bear on any new problem all the available knowledge, of whatever sort and wherever obtained, which may seem to be pertinent. This is the technical physics which is destined not only to work in its laboratories on problems presented by engineers, but to be recognized as an inseparable companion of sound and progressive mechanical engineering.

No. 1488

LAPS AND LAPPING

AN INVESTIGATION OF THE CUTTING PROPERTIES OF ABRASIVES WHEN USED WITH DIFFERENT LAPS AND DIFFERENT LUBRICANTS

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The use of a loose-grained abrasive in connection with a lubricant, for working down surfaces, dates back to very ancient times. The method was first used in the grinding and polishing of precious stones. The workmen were called lapidaries and their tools, charged with the abrasive, called laps.

2 Later the process was applied to the working of hardened steel and, from this, gradually extended to cover a wide variety of operations common to machine shop practice.

3 Several years ago, having occasion to seek definite information on the subject, it became evident that there were few, if any, exact data available. This led to the construction of a machine with which quantitative results could be obtained with various combinations of abrasive lubricant, and lap material. Tests on the machine were confined to surface lapping.

4 There are two methods of using a surface lap which, for want of better definitions, will be termed the "wet" and the "dry" methods. In the wet method there is a surplus of oil and abrasive on the surface of the lap. As the specimen being lapped is moved over it there is more or less movement or shifting of the abrasive particles. The action may be conceived to be somewhat as follows:

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Presented at the Spring Meeting, Buffalo, N. Y., June 1915, of The American Society of Mechanical Engineers.

A particle becomes embedded in the softer surface of the lap, it remains stationary for awhile, but is finally dislodged by friction or change in direction of the rubbing surface. It then rolls around for awhile, only to lodge again and have the process repeated. Meanwhile the grains of abrasive are being broken up into smaller sizes, their sharp edges and corners worn away, the cutting effect grows less, until it becomes necessary to recharge the lap with fresh material.

5 With the dry method, the lap is first charged by rubbing or rolling the abrasive into its surface. All surplus oil and abrasive is then washed off, leaving a clean surface, but one that has embedded uniformly over it, small particles of the abrasive. It is then like the surface of a very fine file or oil stone and will cut away hardened steel that is rubbed over it.

6 While this has been termed the dry method, in practice the lap surface is kept moistened with kerosene or gasoline. This is for the purpose of preventing small spots of steel, called "birds eyes," building up on the lap surface. But there is not enough lubricant applied to "float" the piece being lapped and, hence, this has been called the dry method.

7 Among mechanics there is some difference of opinion as to whether the cutting is more rapid with the wet or the dry method. By advocates of the dry method it is pointed out that in this the abrasive is permanently embedded in the lap surface, there is actual contact between the abrasive and the steel specimen, there is no floating on an oil film, no rolling on loose particles, that the cutting action is more nearly true, and hence more rapid than when these conditions fail to obtain.

8 Materials used for laps may be cast iron, copper, steel, tin, lead, or any combination of the softer metals to give varying degrees of hardness. The choice depends on first cost, permanence of form, rapidity of cutting, and ease of redressing.

9 The lubricants most commonly used are lard and machine oils, kerosene, and gasoline. Alcohol and turpentine have been recommended. It is well known that turpentine can be used to advantage when drilling hard steel. It causes the drill to "take hold" of the steel better. Whether it has a similar effect when hardened steel is cut by an abrasive is a question.

10 Abrasive materials are usually emery, alundum, corundum, carborundum, or others of a similar kind, but sold under

various trade names, as Crystolon, Axolite, Carbondite, etc. Diamond dust is also used and ground glass, oil stone powder, and ground pumice stone find limited application for certain kinds of work.

OBJECT OF TESTS

11 In the selection of the abrasive, both as to kind and size of grain, the metal of which the lap is made, the lubricant and pressure to be used, there is wide range of possible combinations for this work. The object of the experiments was to secure, if possible, reliable data on the following points:

a The relative efficiencies of the different abrasives.

b The relative efficiencies of different lubricants.

c The rate of cutting with laps made of cast iron, soft steel, and copper.

d The wear of the laps, compared one with the other and with the amount of steel ground off with each.

e The effect of pressure on the rate of cutting.

f The rate of cutting by the wet and the dry methods.

12 The problem was more difficult than appeared at first, and it was only after three months of experimental work, after having completed the machine, that the results obtained were considered of sufficient accuracy to warrant going ahead with the work.

13 The usual method was followed of keeping all variables constant except one, and having determined the effect of that one, to proceed to the next. It developed, however, that some of the variables affecting the results were not entirely within control. Thus, for instance, the size of the grains of abrasives is one of the factors affecting the rate of cutting. By the nature of the process the size of the grains is continually changing, being ground down to smaller sizes. This change in size depends on the pressure and to some extent on the lubricant and the material of the lap surface. Hence, the rate of change is of itself a variable.

14 Again, when using volatile liquids, like gasoline, turpentine, and alcohol, fresh additions had to be made to the plate to make up the loss from evaporation. At first the liquid was supplied to the plate through the medium of sight feed oil cups. This was impracticable, because at different stages of a test run, which required about an hour and fifteen minutes to complete, the liquid was required at a different rate. Also a different rate was required for

different pressures and it was impossible so to regulate the feed as to keep a constant degree of saturation on the lapping plate. It was found more satisfactory to supply the lubricant by hand from an ordinary oil can, watching closely and adding the lubricant as needed. This, of course, introduced a personal factor.

15 Another disturbing element was the pressure of the atmosphere on the test specimen. It is not believed that this was in any way serious, but there were occasions when the rate of cutting appeared erratic and variations in pressure seemed to offer the only reasonable explanation. Any one who has rubbed two smooth surfaces together, with a liquid between them, will have experienced the effect of atmospheric pressure. At times the surfaces will seem forced together more strongly than at others. This occurs when the air is wholly or partially excluded from between them. Further, this action is irregular, which means that it is impossible to maintain absolutely uniform pressure between the surfaces. The effect is more marked with large surfaces. During the experimental work, test specimens of 1-in. diameter were tried, but the results with these were not so uniform and consistent as with smaller sizes. The size finally adopted was $\frac{5}{8}$-in. diameter, giving an area of 0.306 sq. in.

16 The condition of the lap surface seemed to have a considerable influence on the rate of cutting. After a run with a pressure of 5 lb. per sq. in. on the specimen, irrespective of which lubricant was used, the plate would show a natural color and possess a uniform velvety surface. With the higher pressures, however, the plate often took on a more or less glazed appearance. In fact, it seemed as if the abrasive worked into the surface during a run with low pressure, would be smoothed down and form a kind of scale or false surface when the higher pressures were used. This was more in evidence with some lubricants than with others. To illustrate more clearly, the copper plate after a run with lard oil would be of a bright copper color, showing no evidence of discoloration. With kerosene, when the higher pressures were used, the plate became darker in color, showing that some of the abrasive had been worked into its surface. This, of course, changed somewhat the nature of the surface, but was unavoidable.

17 These things are mentioned to show more clearly the nature of the problem and to account for small variations in the results. Thorough precaution was taken to reduce all uncertain factors to

the lowest possible limit and confidence is felt that the results are well within the limits of practical accuracy.

DESCRIPTION OF TESTING MACHINE

18 The machine which was designed and used for these tests is shown in Figs. 1 and 2. The horizontal shaft *B*, driven by motor

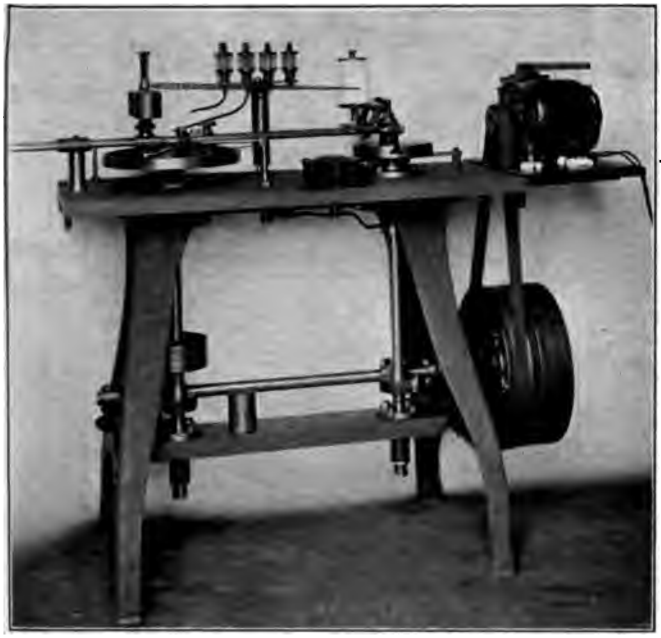


FIG. 1 MACHINE DESIGNED FOR TESTS

A, transmits motion through spiral gears to two vertical shafts *C* and *D*. Shaft *C* carries the lapping plate at its upper end and shaft *D* the slotted crank disk *E*, by means of which, and the connecting rod *F*, the specimen holder *H* is made to reciprocate between the center and outer edge of the lapping plate. The motions of the lapping plate and specimen holder cause the latter and its specimen to follow a path relative to the plate like that shown in Fig. 3, which is from a figure actually drawn on the machine. Originally the plate made about 3 revolutions to 1 of the crank disk, but this was changed to about 1.8 to 1, which gave better results.

19 As at first constructed the machine was autographic, but

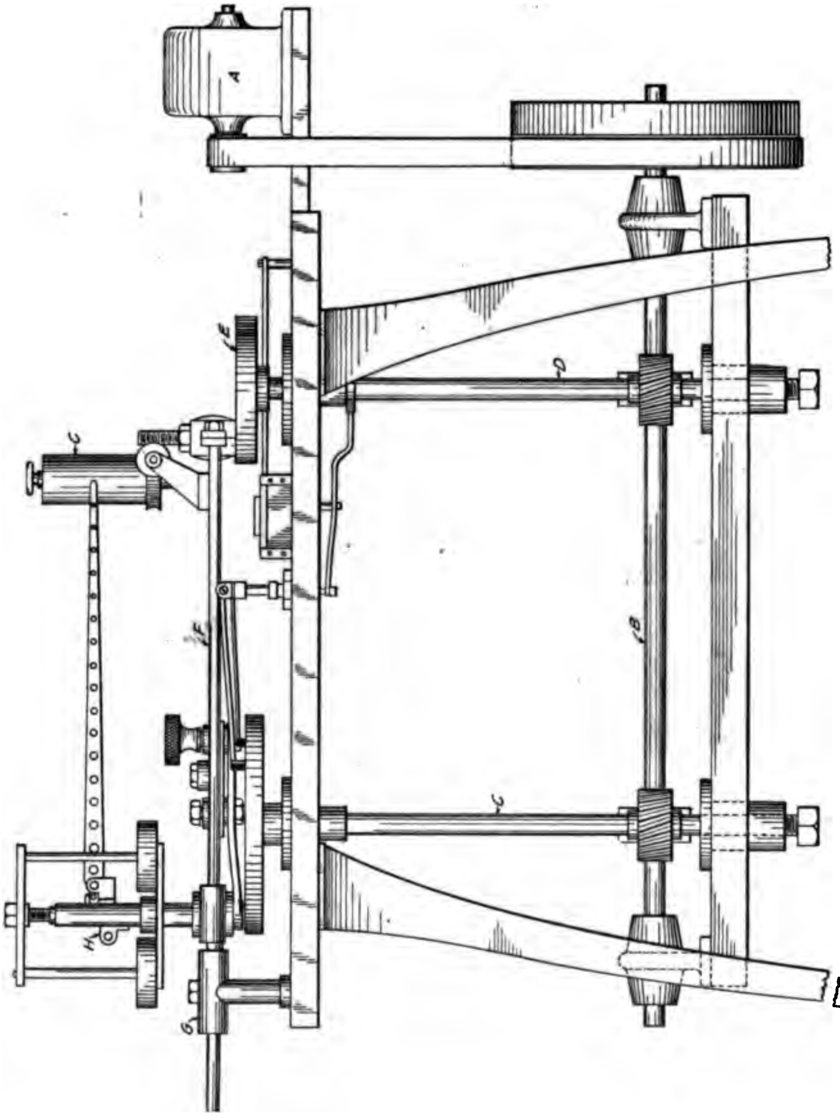


FIG. 2 SIDE ELEVATION OF MACHINE

too many uncertainties were introduced by the friction of the pencil and other conditions and this feature was finally abandoned.

20 The specimen holder is shown in section in Fig. 4. The specimen is held in the hardened steel bushing *O* in end of lever *L*, which receives its motion through connecting rod *F* to which it is attached by studs *M* and *N*. The specimen is held against the lapping plate by the pressure of the weights shown in the figure, which is transmitted to the specimen through the yoke *E*, screw *S* and pin *J*. These parts are supported by a tube *A* held by the split block *B* which in turn is clamped to the connecting rod *F*.

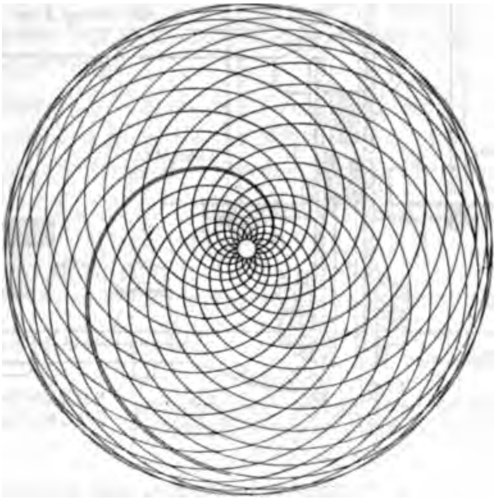


FIG. 3 OUTLINE TRACED BY MACHINE

21 The nut *D*, through which passes the screw *S*, is of tool steel, hardened, ground and lapped to a smooth working fit in the tube. At *K* is a hardened steel plug, which is also a smooth working fit in the tube. This plug has a conical seat at each end in which bear the tapered ends of screw *S* and pin *J*. The object of this construction is to allow a slight lateral movement of the specimen without disturbing the alignment or in any way increasing the friction of the moving parts in the vertical tube. A perfectly free vertical motion is thus obtained whereby the weights can follow up the wear of the specimen with practically no friction.

22 The bushing *O* which carries the specimen is counterbored in order that any side thrust may be brought as near the lower

surface of the specimen as possible. In order to move the specimen from under the tube for examination or removal, the knurled nut *N* is loosened, thus permitting the arm *L* to be turned about the stud *M*.

23 It was the original intention to have the specimen carried directly in the vertical tube, but after some experience with this and several other forms they were discarded. The present holder

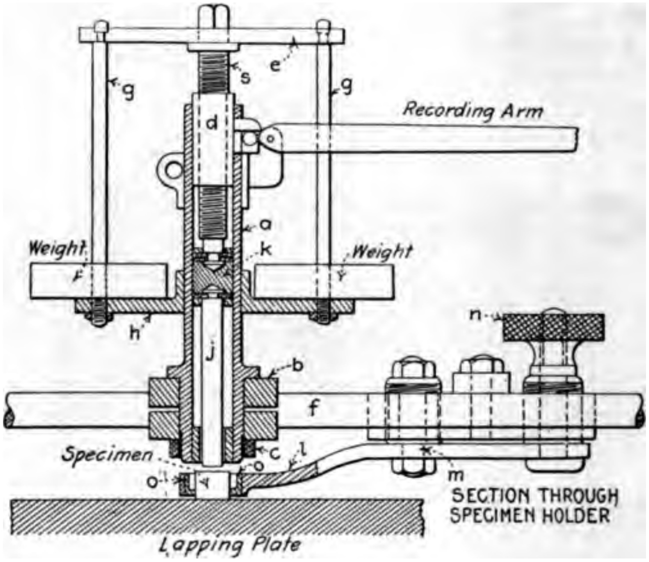


Fig. 4 Section through Specimen Holder

was developed as a result of experience gained during the preliminary stages of the work.

THE LAPPING PLATE

24 The lapping plate is shown in section in Fig. 5. The lap surface is at *A*. A gray iron plate, cast originally ½ in. thick and finished down to 3/8 in. thick, is held to the lower plate *B* by means of screws. The outer rim *C* is also fastened by screws and is removable.

25 At first it was attempted to use a plate cast in one piece, but it was found that it was more porous near the center than at the edge and further that it was impracticable to face the plate in a lathe without leaving pit marks in the surface. Obviously these

conditions would produce unsatisfactory results and it was therefore decided to use a removable upper plate of uniform thickness which would be practically uniform in texture and could be readily finished to a true surface, clear to the edge, by grinding.

26 Five plates were cast from the same heat, so if one or more developed defects or was worn out, another of the same metal could be substituted. The plates were of a good grade of foundry iron, but no attempt was made to get a special mixture.

27 The copper and steel laps were built up the same way. Each consisted of a cast iron backing plate faced with its proper material.

28 The copper plate was made of plate copper ¼ in. thick. Since this was too thin to be tapped into, it was secured to the cast plate by means of solder.

29 The steel plate was made of fire box steel ¼ in. thick and secured in the same manner as the copper plate. Like the cast iron plate, the surfaces of these were finished by grinding.

DISTRIBUTOR

30 The distributor, or wiper, is an important feature of the machine. Referring again to Fig. 2, this will be seen at *T*, and is shown more in detail in Fig. 6.

31 It was essential that there be a uniform distribution of the charge of abrasive over the entire surface of the plate. Centrifugal force was depended upon to work the charge from the center of the plate outward. A wiper, consisting of a strip of wood resting on the plate and inclined at an angle of 15 deg. to a radial line from the center, was counted on to shear the charge back toward the center. At times this worked finely, leaving enough of the charge pass under the wiper to give an even and uniform distribution. At other times the wiper would apparently adhere tightly to the plate, wiping it nearly clean and bunching the charge in the center. Notches were cut in the underside of the strip of wood, through which a portion of the charge could always flow. This helped very materially, but the distribution was further improved by giving motion to the wiper. This was effected through the medium of an eccentric *R* placed on the shaft *D*. To prevent an accumulation of part of the charge around the inner surface of the outer ring, a small auxiliary wiper was placed as shown at *Q*. Details of the notched piece of wood are shown in Fig. 7.

TEST SPECIMENS

32 The test specimens were of hardened tool steel. It was first attempted to use a hollow specimen of the form shown in Fig. 8, in order to secure uniformity in hardening and to reduce the tendency to rock while being moved over the lap surface. After many trials this form was given up. It seemed absolutely impossible to duplicate runs with any degree of certainty. The cause of this

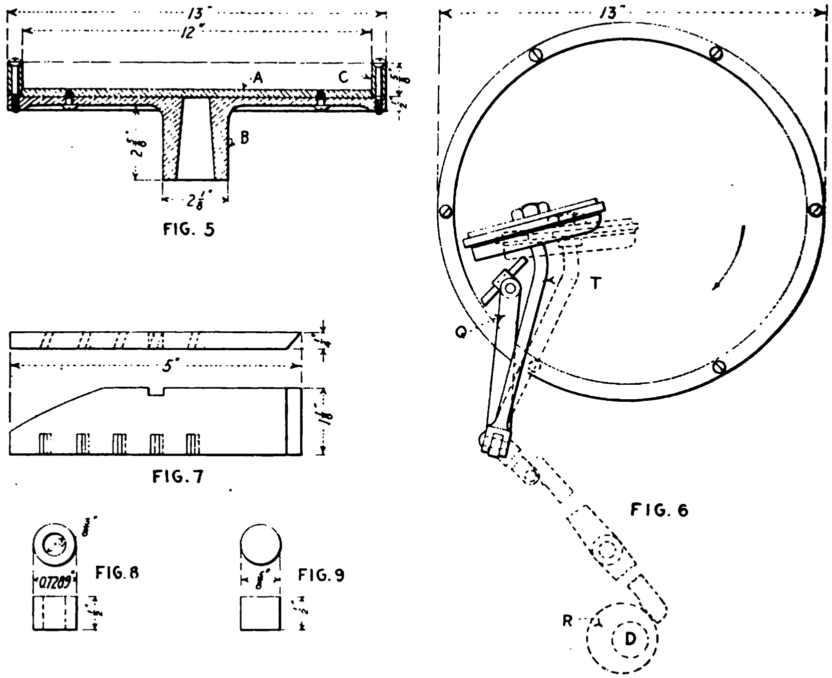


Fig. 5 Section through Lapping Plate
Fig. 6 Details of Distributor
Fig. 7 Distributor Shoe
Fig. 8 Test Specimen, Form Rejected
Fig. 9 Test Specimen, Form Used

is not exactly clear, but it may be due to the fact that the surface in contact with the plate being perfectly flat and the edges perfectly square and sharp would allow the specimen to shear the abrasive ahead of it instead of riding over it.

33 The specimens finally used, as shown in Fig. 9, were made of 5/8-in. drill rod. Fifty-five pieces 1/2 in. long were cut from the

same bar and numbered consecutively. They were hardened by heating in the muffle of a gas furnace, the temperature of which was regulated by a pyrometer, and quenching in clear water to which salt had been added to the amount of 3 lb. to the gallon.

SCOPE OF TESTS

34 It was not the intention to try out all the different abrasives on the market. Three were selected as being representative, namely, Naxos Emery, obtained from the Safety Emery Wheel Co., Springfield, Ohio; Carborundum, direct from the Carborundum Company, Niagara Falls, and Alundum, from the Norton Company, Worcester, Mass.

35 All the tests were carried out with abrasive No. 150. Each of the abrasives was tried with seven different lubricants, five different pressures, and three different laps. The lubricants were lard oil, machine oil, kerosene, gasoline, turpentine, alcohol, and soda water.

36 Starting a series of tests, say with emery as the abrasive and with cast iron lap, the first test was made with lard oil and with a pressure of 5 lb. per sq. in. on the specimen. The pressure was then increased to 10 lb. per sq. in., other conditions remaining the same, and another run made. The pressures were then increased to 15, 20, and 25 lb., giving a group of five runs with lard oil.

37 Machine oil was then substituted for lard oil and a like group of tests made, after which tests were made with the other lubricants in the same way. This gave for cast iron-emery a series of 35 tests.

38 Carborundum was next substituted for emery and the same number of tests repeated. This was followed by a like series with alundum, making the total number of tests with cast iron lap 105.

39 The cast iron lap was then replaced by one of steel and a second series of 105 tests run with the steel lap. This was followed by a like series with the copper lap. Therefore, for making comparisons of the action of the three abrasives and also for the three lap materials, cast iron, copper, and steel, we have 105 tests of each.

DETAILS OF TESTS

40 The quantity of abrasive used for each test was as follows:

3 grams of emery
2.46 grams of alundum
2.40 grams of carborundum.

These weights gave equal volumes, and this was considered to give a fairer test than if equal weights had been taken, because it gave equal density or the same number of grains of the abrasive per square inch of plate surface. The charge was weighed for each test.

41 When making tests with lard and machine oils, 7 cc. were used. This was applied at the beginning of the test and no further

Test No. 92 date 3-12-13 Observer - KNIGHT

ABRASIVE - Emery - Grade 150 - Lubricant M. Oil Pressure 15 lbs.

Reading of Counter	Reading of Counter	Revolu-tions	Weight beginning of run	Weight end of run	Weight ground off
21000	21500	500	19330	19240	90
	22000	500		19186	54
	23000	1000		19139	47
	24000	1000		19110	29
	25000	1000		19090	20
			19330 19090		90 144
			240		191 220
					240

FIG. 10 BLANK USED IN RECORDING TESTS

additions made. With the light and volatile liquids 7 cc. was too much to start with, as the specimen would get down to the lap surface and shear the abrasive ahead of it. With these liquids, the test was started with 3 cc. and then additions made from time to time to make up the loss from evaporation. Before starting a run, the charge of abrasive and lubricant was distributed with the fingers as uniformly as possible over the surface of the lap.

42 A test was continued until the crank which moved the specimen over the plate had made 4000 rev. At the end of 500 rev., the specimen was removed, cleaned with gasoline, and weighed. This was repeated at the end of the next 500, and then for each 1000 until the 4000 rev. were completed.

43 The log of a single test is shown in Fig. 10, while Figs. 11 to 17, inclusive, show the plotted results of the individual tests of the cast iron-emery series. As plotted, the ordinates represent the amount in milligrams ground from the specimen, abscissa the number of revolutions of the crank which swept the specimen over the plate.

44 The individual tests of this series are shown plotted not because they are better or worse than the others, but to show at a glance the general form of individual curves, how the shape varied with the pressure, and the degree of consistency in the various tests.

45 A summary of all the results is given in Table 1.

TABLE 1 SUM OF AMOUNTS GROUND FROM SPECIMENS WITH PRESSURES OF 5, 10, 15, 20, AND 25 LB. PER SQ. IN.

		Machine Oil	Lard Oil	Kerosene	Gasoline	Turpentine	Alcohol	Soda Water	Total
Cast Lap	Emery	1320	1673	3324	3955	2336	2392	3105	18105
	Alundum	1849	2313	3687	4291	3230	3251	3112	21733
	Carborundum	2825	3340	4230	4520	4458	4427	3873	27673
	Total	5994	7326	11241	12766	10024	10070	10090	67511
Steel Lap	Emery	1744	2460	2720	3034	2560	2608	2839	17965
	Alundum	1800	2537	3622	3627	3400	3388	3338	21712
	Carborundum	4199	4649	3805	3980	3983	3527	4045	28188
	Total	7743	9646	10147	10641	9943	9523	10222	67865
Copper Lap	Emery	3250	3454	3756	3813	3598	3961	3780	25612
	Alundum	3971	4065	3763	3960	4171	4097	4472	28499
	Carborundum	4148	4540	3692	3724	4251	4081	4210	28646
	Total	11369	12059	11211	11497	12020	12139	12462	82757
	Grand total for each lubricant	25106	29031	32599	34904	31987	31732	32774	

COMPARISON OF RESULTS

46 In considering the general results, the various factors are so inter-related that a discussion of one naturally becomes involved with all the others, hence more or less repetition is unavoidable.

47 Obviously, it is impracticable to compare each individual result with every other result. The plan has therefore been adopted of combining the five curves of the different pressures into one curve. This is called the master curve for that particular combination, and is effected by simply summing up the values of the ordinates for different points on the curves and using these values

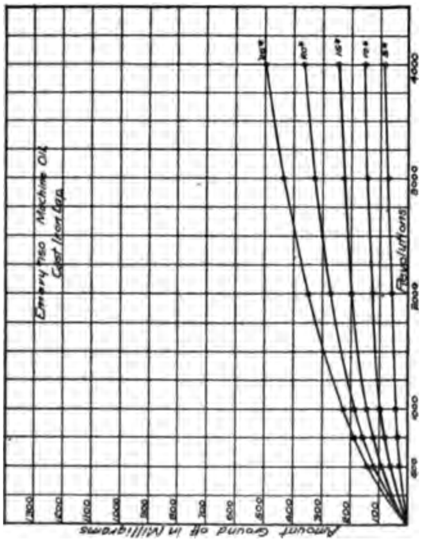


FIG. 11

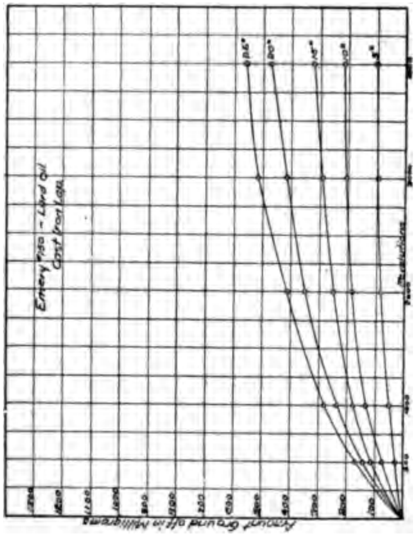


FIG. 12

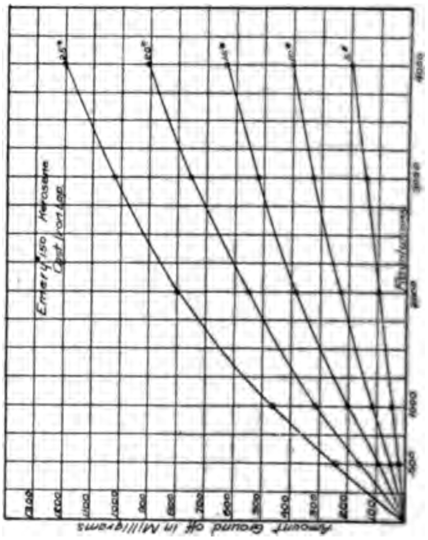


FIG. 13

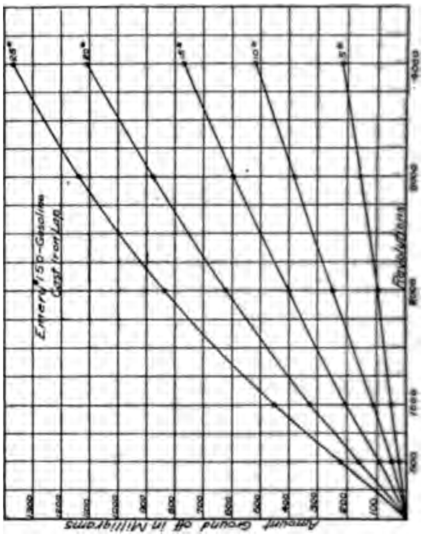


FIG. 14

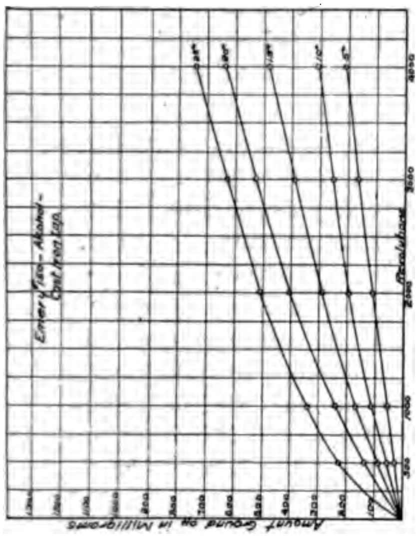


FIG. 16

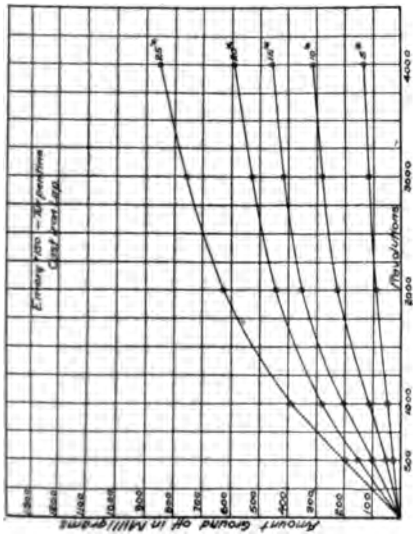


FIG. 15

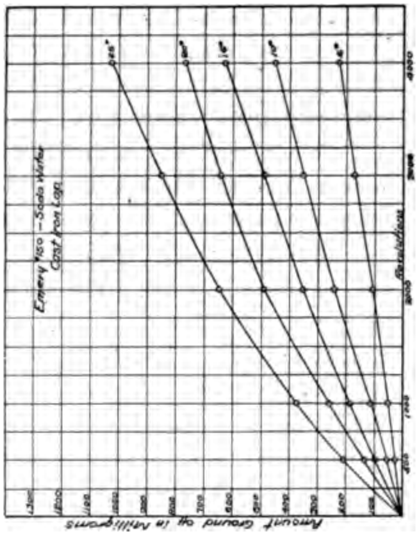


FIG. 17

in plotting the master curve. If these values were divided by five, the result would be an average curve of the five pressures. The curves as they stand, without such division, are practically the same thing.

48 This method is not strictly correct, because it assumes the curves to be of the same form and that the rate of cutting is proportional to the pressure. This latter is very nearly true but the shape of the curves changes somewhat with the pressure. In general, the curve starts off concave upward and at some point in the run changes to convex. This change takes place at different distances from the origin. It is further out for the light pressures and approaches the origin as the pressure is increased. This condition is clearly indicated in the set of curves for gasoline, Fig. 14. Since, however, all results are combined in the same way, it is thought that this method forms a fair basis of comparison. The X marks on the curves indicate about where the maximum rate of cutting is reached.

49 It will be understood then that each of the comparative curves for the different lubricants, abrasives, and lap materials is the combined curve of five separate runs.

50 *Lubricants.* The action of the different lubricants presents an interesting study. The same lubricant acts differently with the different abrasives and again differently with different laps. It might be surmised that the main difference would be due to changes in viscosity or to the lubricating properties of the various lubricants. There is no doubt that these are important factors, but it is equally certain that they are not the only ones.

51 Taking the results of the emery-cast iron series of tests alone, as shown plotted in Fig. 18, it would seem that the change in the viscosity and lubricating properties would offer a reasonable and fairly complete explanation of the difference in their behavior.

52 Gasoline, which is lowest in these respects, has the highest rate of cutting. It will be noted, too, that its rate of cutting is well maintained, being nearly as high at the end of 4000 rev. as at the beginning. Next is kerosene, the rate of cutting of which is high, but which shows slightly more of a decreasing rate as the end of the run is approached. Soda water is below that of kerosene, but maintains its rate of cutting more nearly like gasoline. Alcohol and turpentine have about the same rate of cutting at the start as soda water, but decrease more rapidly as the run proceeds. Both these liquids have

peculiarities which will be considered further while discussing individual lubricants.

53 The rate of cutting for both lard and machine oil drops off very rapidly. At the end of a run, that for machine oil is about 166 per 1000 rev., while that for gasoline is about 850, these two standing at the extremes of the range.

54 What appears to happen is this: As the grinding proceeds the emery is constantly reduced to a finer powder and the charge is being contaminated with material ground from the specimen and the lap surface. With a lubricant like gasoline there is no muck formed, the charge remains comparatively clean and the thickness of the film of gasoline is so little that the specimen may follow up the reduction in size of grain of abrasive and, hence, a nearly uniform rate of cutting is maintained.

TABLE 2 CAST IRON LAP

	Emery	Alundum	Carborundum
Highest	Gasoline.......3955	Gasoline.......4291	Gasoline.......4520
Lowest	Machine oil.....1320	Machine oil.....1849	Machine oil.....2825
Difference	2635	2442	1695

55 On the other hand, with the heavy oils, as the grinding proceeds the charge becomes muck-like, the lubricant is thickened and the specimen rides on the thickened oil film. When the grains have been reduced in size to a little less than the thickness of the oil film, the cutting action almost ceases.

56 Passing, however, to the alundum-cast iron series, a comparison of the lubricants of which is shown in Fig. 19, there is not so much difference in their rate of cutting as with emery. Lard and machine oil are in the same relative position to each other and to the other lubricants, but they have approached more nearly to the others in rate of cutting. Reference to Fig. 20 will show that the difference further disappears when carborundum is used as the abrasive material. Table 2 is presented to further emphasize this. Of the seven lubricants used, this table shows the ones which gave the highest and the lowest values when used with the cast iron lap and different abrasives.

57 The curves for the steel lap and the copper lap are shown in Figs. 21 to 26. An examination of these will reveal a further

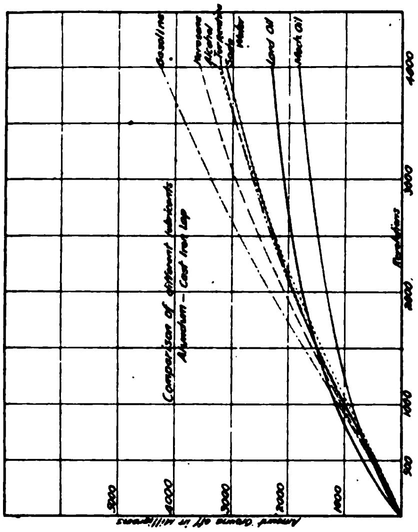


FIG. 19

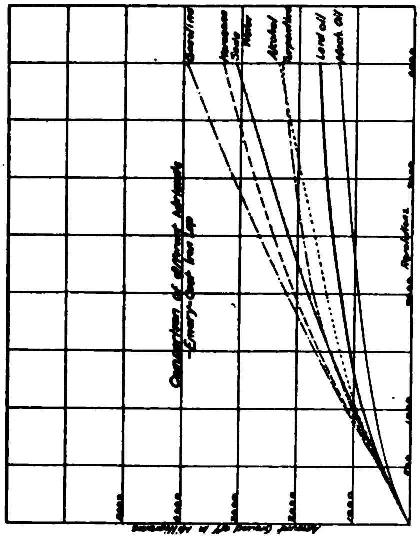


FIG. 18

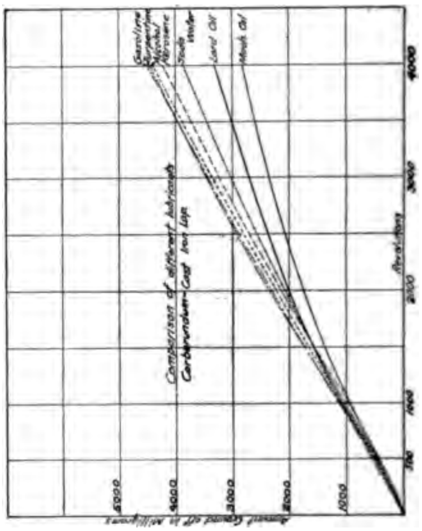


FIG. 20

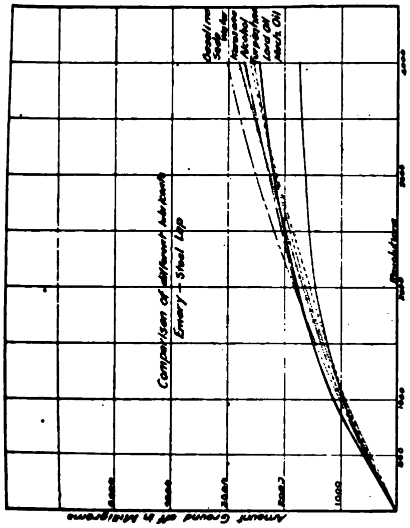


Fig. 21

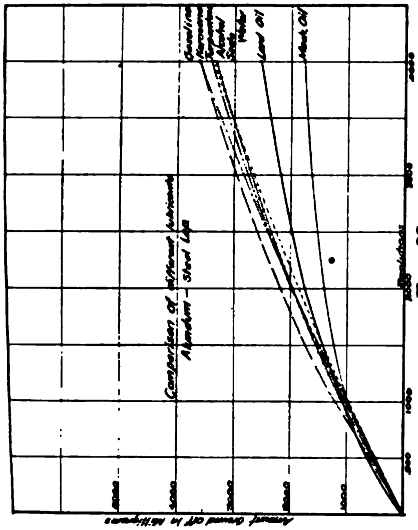


Fig. 22

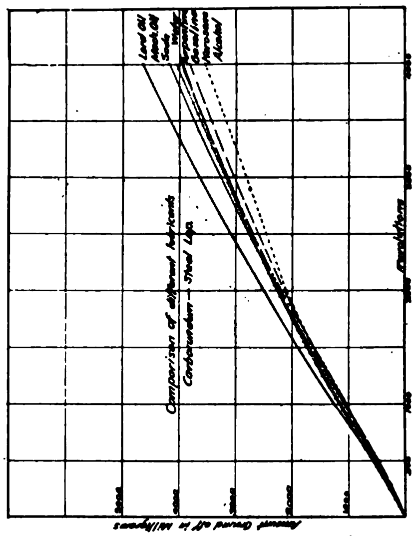


Fig. 23

Comparison of different Lubricants on Steel Lap

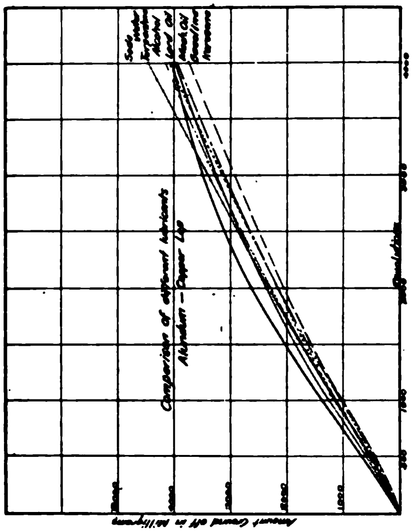


FIG. 25

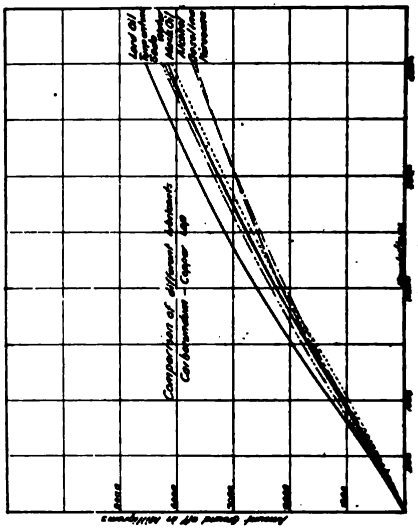


FIG. 26

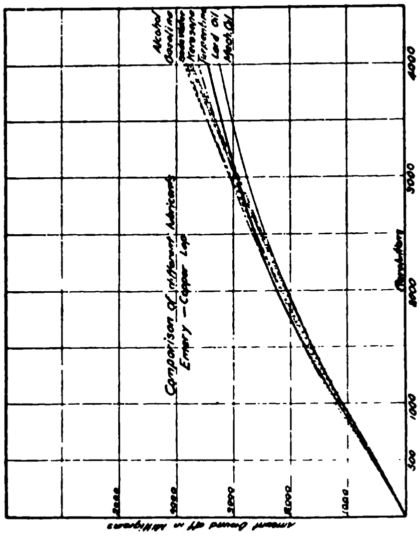


FIG. 24

closing-in of the curves, or, in other words, a nearer approach to each other in rate of cutting. Lard and machine oil take on higher values. These two oils, with emery on the cast iron lap, give lower values than any of the other lubricants, while the same two, with carborundum on the steel lap, give higher values than any other of the lubricants. It seems evident, therefore, that factors other than viscosity and lubricating properties play an important part in the rate of cutting.

58 As a possible explanation, it is suggested that the lubricant, together with the material ground from the specimen and the lap surface, forms a kind of a matrix or bonding material for the abrasive, and that the rate of cutting is largely dependent on the nature of this matrix. Evidently there will be a different matrix formed with each different combination of lap, abrasive, and lubricant.

59 It is probable, too, that there is some property of the lubricants which enables some to cause a sharper or cleaner cutting action than others. We know that for thread cutting and many other machine tool operations, a cutting tool used with lard oil takes a cleaner cut and leaves a smoother surface than when used with machine oil. There is evidence that some such property is present and exerts an influence on the results obtained with the different lubricants.

INDIVIDUAL LUBRICANTS

60 *Lard and Machine Oil.* On account of their differences and yet of their similarity of action, lard and machine oil will be considered together. It is to be observed

a That in tests under all conditions, their curves are of the same form and follow each other closely.

b That lard oil without exception gives the higher rate of cutting.

c That in general the initial rate of cutting is higher than with the lighter lubricants, but falls off more rapidly as the run proceeds.

d That both the highest and lowest results of the whole number of tests were obtained with these two lubricants. The lowest with machine oil, emery-cast iron lap, with lard oil a little above it; the highest with lard oil, carborundum-steel lap, with machine oil a little below it. That is, while they have maintained the same relative position to each other, both have advanced until, with the steel lap and carborundum, they lead all other lubricants in rate of cutting. Compare the curves, Fig. 18, with those of Fig. 23.

61 Table 3 shows this progressive increase in the values obtained with the different combinations:

TABLE 3 AMOUNT GROUND FROM SPECIMEN WITH MACHINE AND LARD OIL FOR THE DIFFERENT COMBINATIONS OF LAP AND ABRASIVE

	Machine Oil			Lard Oil	
Cast Lap	Emery	1320	Cast Lap	Emery	1673
	Alundum	1849		Alundum	2313
	Carborundum	2825		Carborundum	3340
Steel Lap	Emery	1744	Steel Lap	Emery	2460
	Alundum	1800		Alundum	2557
	Carborundum	4199		Carborundum	4649
Copper Lap	Emery	3250	Copper Lap	Emery	3459
	Alundum	3997		Alundum	4065
	Carborundum	4148		Carborundum	4540

62 *Gasoline and Kerosene.* On the cast iron lap gasoline shows the highest results of any of the lubricants tested. It is not so good on copper and still less so on steel. Taking into account all three abrasives, its relative value on the different laps is as follows:

Cast iron 127 Copper 115 Steel 106

63 Contrasting these results with those obtained with lard and machine oil, we find that with them there is an increase in the rate of cutting with the steel and copper lap, whereas with gasoline there is a decrease.

64 Kerosene shows more nearly the characteristics of gasoline than of the heavier oils. Like gasoline, it gives the best results on cast iron and the poorest on steel. It does not work so well with carborundum on the copper lap. While the result with this combination is not so low as some others, it is low for carborundum, because, in general, this gave higher results than the other abrasives. In fact, for the entire series of 315 tests, there were but two instances in which the values obtained for carborundum were not higher than those with emery, and these two were with gasoline and kerosene on the copper lap. Mention has already been made of the discoloration of

the copper plate when these liquids were used. It seems as if some of the abrasive was worked into the surface and then smoothed down, giving, instead of a clean copper surface, one more or less hard and glazed. This was more in evidence with the higher pressures and no doubt caused a falling off in the rate of cutting. However, just why the effect should be more pronounced with carborundum than with the other abrasives is not clear.

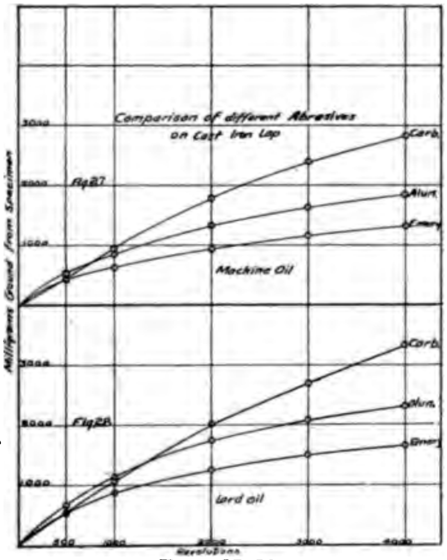
65 *Turpentine and Alcohol.* There is no evidence to show that turpentine possesses any superior advantage over the other lubricants. On any lap it does good work with carborundum. With emery it does fair work on the copper lap, but with emery on the cast iron and steel lap it is distinctly inferior.

66 When used on cast iron or steel with emery, the charge soon becomes pasty or muck-like. The resultant residue is very dark, almost black, and at the end of a run seems fine and smooth with very little of its gritty nature left. As a considerable portion of turpentine evaporated during the course of a run, and the loss made up by fresh additions, it was at first thought that the thickening of the charge was due to a gummy residue being left by such evaporation. However, with carborundum the charge showed no such thickening, the residue being more like that formed with machine oil. This accounts for the better showing turpentine makes when used with carborundum.

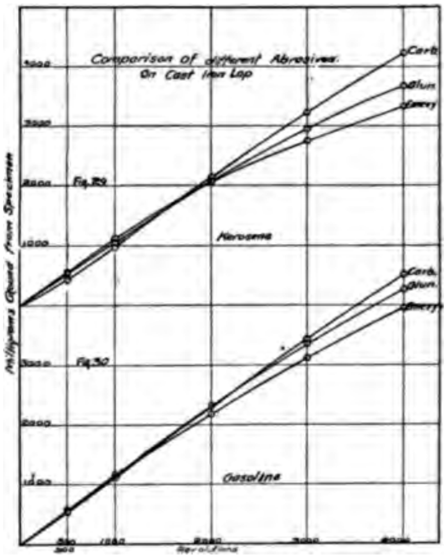
67 Alcohol in some ways acts very much like turpentine. It also gives the lowest results with emery on the cast iron and steel laps. Like turpentine, the residue at the end of a run is fine and smooth. But the residue instead of being black and pasty is of a deep reddish brown color and inclined to be foamy or spongy.

68 Alcohol and turpentine were the only two lubricants that gave any evidence of having a solvent or chemical action on the abrasive, and these with emery only. Since, however, neither showed any particular merit for this work over other lubricants, it was thought useless to pursue that phase of the subject any further.

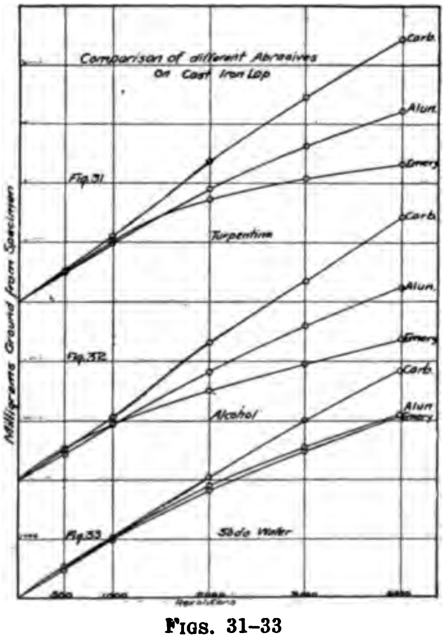
69 *Soda Water.* Soda water gives reasonably good results with almost any combination of lap and abrasive. Referring again to the comparison curves for the lubricants, Figs. 18 to 26, it will be seen that it maintains a fairly constant position intermediate between the other lubricants. While it is seldom the best, it is never the worst. It does its best work on the copper lap and poorest on steel, although there is not much difference between its work on the cast iron and



Figs. 27–28

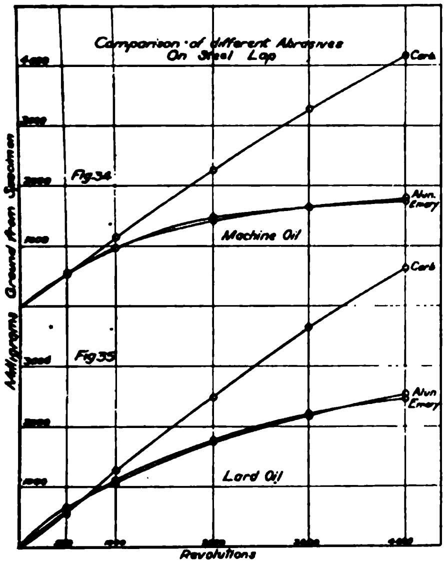


Figs. 29–30

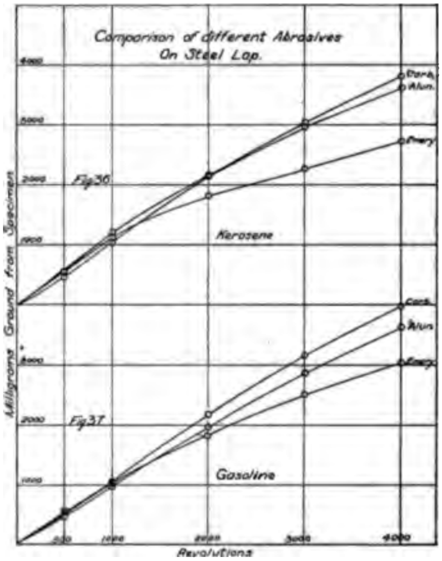


Figs. 31–33

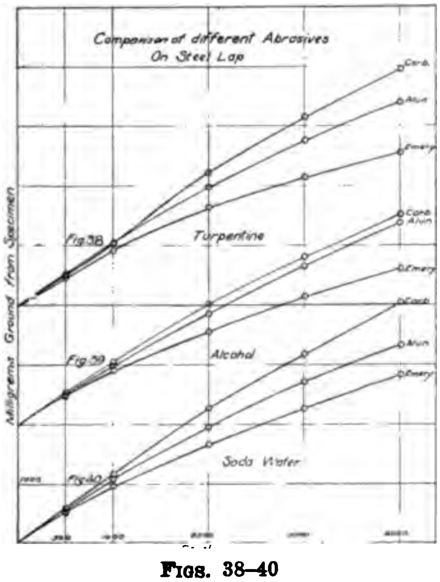
Comparison of different Abrasives on Cast Iron Lap



FIGS. 34–35

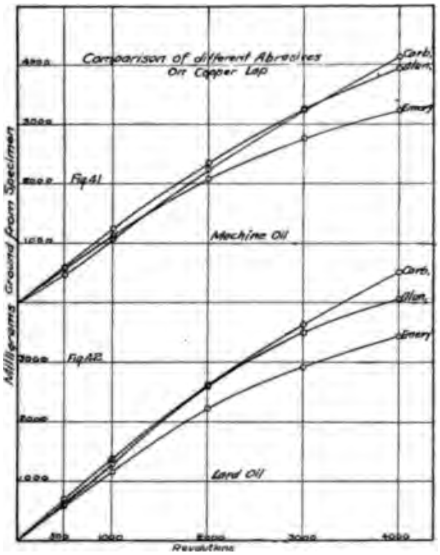


FIGS. 36–37

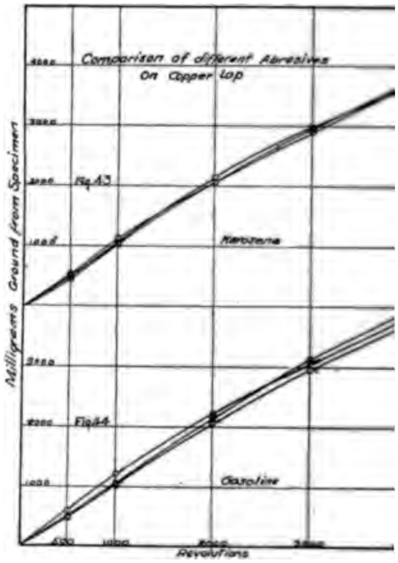


FIGS. 38–40

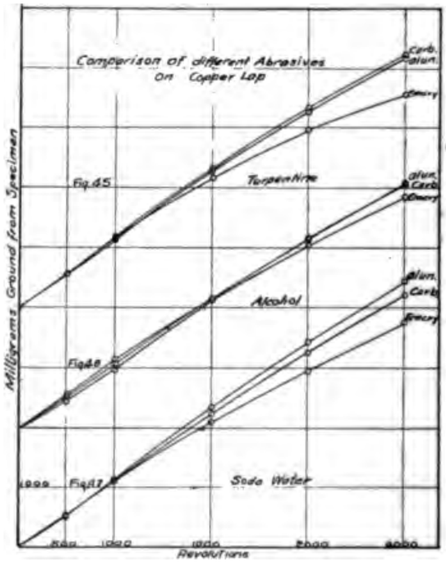
COMPARISON OF DIFFERENT ABRASIVES ON STEEL LAP



FIGS. 41–42



FIGS. 43–44



FIGS. 45–47

COMPARISON OF DIFFERENT ABRASIVES ON COPPER LAP steel. On the cast iron lap it does better work than machine or lard oil, but not so good as gasoline or kerosene. There was one instance, that with alundum on the copper lap, in which soda water gave the highest results of any of the lubricants used with that particular combination. It is not a nice liquid to work with. The residue is muck-like and sticky. When dry it forms a hard, crusty substance. While undesirable from this standpoint, as far as results go and as to first cost, it should rank ahead of turpentine and alcohol.

THE ABRASIVES

70 It may be well to call attention to the fact that emery and alundum are similar abrasives, both being aluminum oxides of the form Al_2O_3. Emery is a natural product, more or less contaminated with iron or other impurities. Alundum is an artificial product and in general, of greater purity than the natural product. Carborundum, on the other hand, is an entirely different material, being a carbide of silicon, SiC. Naturally, then, emery and alundum might be expected to show more nearly the same characteristics, while carborundum would deviate more or less for them.

71 That this is true, an examination of the curves, Figs. 27 to 47, will show. These curves show graphically the rates of cutting for the three abrasives for all the different combinations of lap and lubricant.

72 It will be noticed that carborundum usually starts off at a lower rate than the other abrasives, but once started, its rate is maintained better. Its curve, in general, is more nearly a straight line. The charge or residue as the grinding proceeds remains cleaner and sharper and is not inclined to become pasty or muck-like, as is so frequently the case with emery.

73 Alundum, both as to its rate of cutting and cleanliness of residue, is, in general, intermediate between carborundum and emery.

74 Taking the total amount of steel ground from the specimens with each of the abrasives as a basis of comparison, they stand as follows:

Carborundum 84,507 Alundum 71,944 Emery 61,682

75 These figures are for all three laps. Divided up according to the amounts ground off with each individual lap, the results are as follows:

TABLE 4 AMOUNTS IN MILLIGRAMS GROUND FROM SPECIMENS WITH DIFFERENT COMBINATION OF LAP AND ABRASIVE

Cast Lap	Emery.......18105	Steel Lap	Emery....17965	Copper Lap	Emery...25612
	Alun........21733		Alun......21712		Alun....28499
	Carb........27673		Carb......28188		Carb.. .28646

76 It is to be observed that there is a greater difference in the action of the abrasive with the cast iron and steel lap than with the copper. With the copper lap, carborundum shows but little gain over the cast iron and steel, while with emery and alundum, the gain is considerable.

77 It may be pointed out, again, that the evidence all the way through tends to the conclusion that there is for each different combination of lap and lubricant a definite size grain of abrasive that will give maximum rate of cutting. With all, except the two heavy lubricants, some reduction in size of grain below that used in the tests (No. 150) seemed necessary before the maximum rate of cutting was reached. But this reduction in size goes on continuously and soon passes below that which gives maximum cutting. It is at the point of passing this definite size that the curve changes from concave to convex, or, in other words, the cutting changes from an increasing to a decreasing rate. As before explained, the change depends on all four factors, namely, pressure, abrasive, lubricant, and lap material.

78 Emery appears to be more brittle and passes through the change quicker than the others, with alundum next and carborundum the least susceptible to such a change. This accounts for the initial rate of cutting with carborundum being lower than the other abrasives.

COMPARISON OF THE LAPS

79 The form and material of laps have already been given. Their hardness, as determined by the research department of the Westinghouse Electric and Manufacturing Co., was as follows:

By the Brinell Method

Cast iron 109	Steel 87	Copper 43.6

By the Sclerescope

Cast iron 28	Steel 18	Copper 5

80 A comparison of the three laps with all combinations of abrasive and lubricant is given in the set of curves, Figs. 48 to 68.

81 The total amount ground from the specimen with each of the three laps was:

With cast iron	67511
With steel	67865
With copper	82757

This shows that taking the whole number of tests as a criterion, there is scarcely any difference between the steel and cast iron, but that copper does somewhat the best work. Since, however, there is a great difference between the highest and lowest values obtained with each individual lap, it would seem more logical in comparing their relative merits to compare the highest values obtained with each. On this basis they stand in this order:

Steel with carborundum,—lard oil	4649
Copper with carborundum,—lard oil	4540
Cast iron with carborundum,—gasoline	4520

Here, again, there is not so much difference, but it shows that with the proper abrasive and lubricant, steel and cast iron are equally as good (for all practical purposes) as copper.

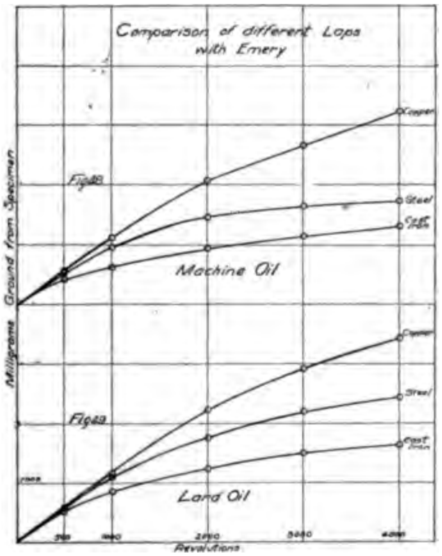
WEAR OF LAPS

82 One of the remarkable facts brought out was the great difference in wear of the laps both as regards material of which they were made and by the different abrasives. The wear on all laps was about twice as fast with carborundum as with emery, while with alundum the wear was about one and one-fourth times that with emery. On an average the wear of the copper lap was about three times that of the cast lap. This is not absolute wear, but wear in proportion to the amount ground from the specimen. Table 5 shows this very clearly.

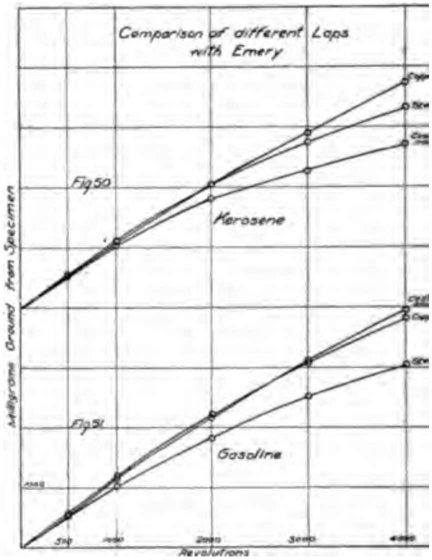
TABLE 5 AMOUNTS GROUND FROM THE LAP SURFACE FOR EACH 100 MILLIGRAMS GROUND FROM THE SPECIMEN

Emery	Alundum	Carborundum	Total
Cast iron....... 81.2	118	158	357.2
Steel...........114	149	190	453
Copper.........233	295	410	938

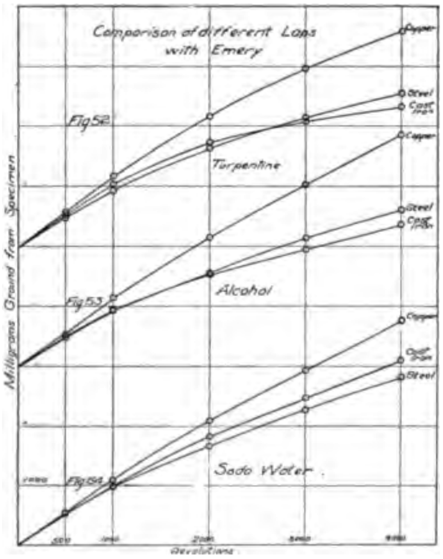
83 Comparing the total wear of the laps with their hardness by the Brinell test, it is found that the wear is very nearly in inverse ratio to the hardness. Thus



Figs. 48–49

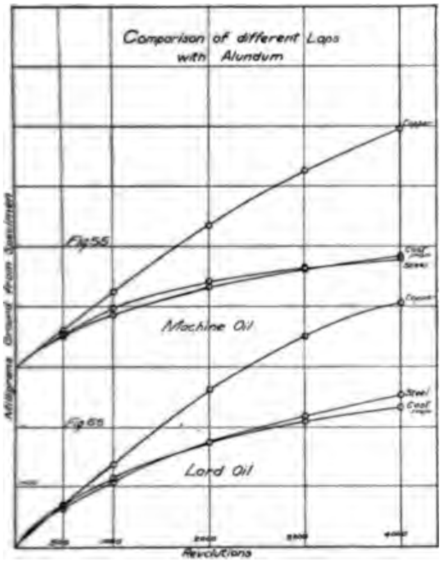


Figs. 50–51

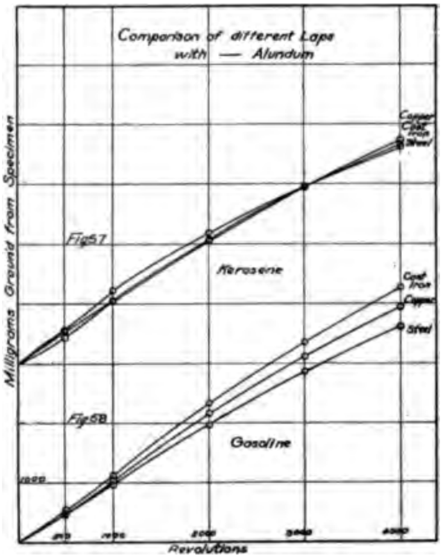


Figs. 52–54

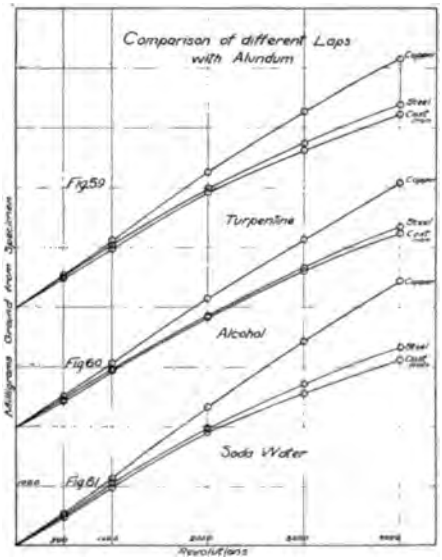
Comparison of different Laps with Emery



FIGS. 55–56

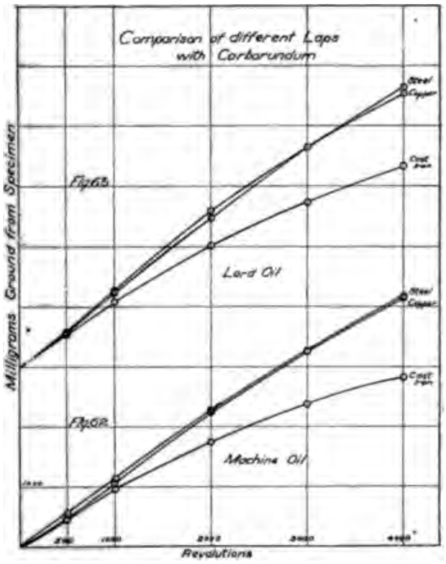


FIGS. 57–58

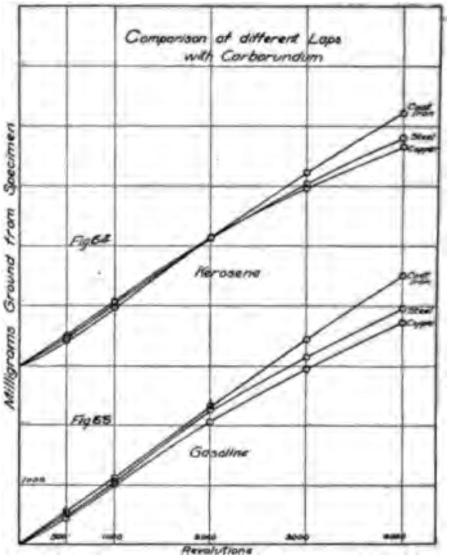


FIGS. 59-61

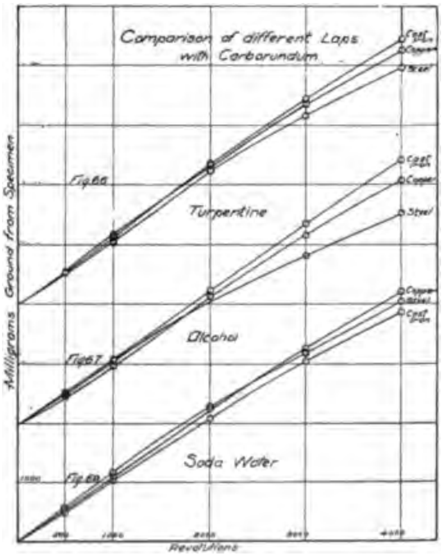
COMPARISON OF DIFFERENT LAPS WITH ALUNDUM



FIGS. 62–63



FIGS. 64–65



FIGS. 66–68

COMPARISON OF DIFFERENT LAPS WITH CARBORUNDUM

	Wear of Lap	×	Hardness	= a Constant
Cast iron.......	357.2	×	109	= 38,934
Steel	453	×	87	= 39,411
Copper	938	×	43.6	= 40,896

No such relation holds between the scleroscopic hardness and the wear.

84 As regards permanence of form, then, cast iron is altogether better than either steel or copper, and taking into account its first cost and that with proper abrasive and lubricant its rate of cutting is practically as good as copper or steel, it is far and away the best lap material.

85 From results obtained on the wear of the laps, it is evident that the theory of the lodgment of the abrasive in the softer lap surface of the lap is not well founded. The action appears to be more mutual between the surfaces. That is, while a grain is making a cut or scratch across the steel specimen, it is at the same time cutting one in the lap surface, and, encountering less resistance, cuts a longer and deeper scratch in the lap surface than in the hardened steel.

86 It is probable that carborundum, which is a hard, sharp abrasive, gets more of a "foot hold" in the hardened steel than the other abrasives and thus does more damage to the lap surface.

PRESSURE

87 Within the limits of the pressures used; that is, up to 25 lb. per sq. in., the rate of cutting is practically proportional to the pressure. That this is not strictly true is because of the change in size of grain as the grinding proceeds. There is an increasing rate for a time and than a decreasing one. The greater the pressure the quicker this change. During the early part of a run the rate of cutting is not only increased by additional pressure but takes on another increment, due to the change in size of grain. After the maximum rate has been reached, this increment is negative and hence the rate of cutting is not quite proportional to pressure.

88 The higher pressures, 20 and 25 lb. per sq. in., did not do so well on the copper lap as on the others. There was some evidence tending to show that for this lap the practical limits of pressure had been reached.

89 Of the 63 combinations tried out, the 15 giving best results have been selected and are presented in Table 6:

TABLE 6 COMPARATIVE VALUES OF THE BEST COMBINATIONS, TAKING EMERY-CAST IRON LAP AND MACHINE OIL AS UNITY

Carborundum	—Steel lap	—Lard oil	3.52
Carborundum	—Copper lap	—Lard oil	3.44
Carborundum	—Cast lap	—Gasoline	3.42
Alundum	—Copper lap	—Soda water	3.39
Carborundum	—Cast lap	—Turpentine	3.37
Carborundum	—Cast lap	—Alcohol	3.35
Alundum	—Cast lap	—Gasoline	3.25
Carborundum	—Copper lap	—Turpentine	3.22
Carborundum	—Cast lap	—Kerosene	3.20
Carborundum	—Copper lap	—Soda water	3.19
Carborundum	—Steel lap	—Machine oil	3.17
Alundum	—Copper lap	—Turpentine	3.15
Carborundum	—Copper lap	—Machine oil	3.14
Alundum	—Copper lap	—Alcohol	3.10
Carborundum	—Copper lap	—Alcohol	3.09

DRY LAPPING

90 Experiments on dry lapping were carried out on the cast iron, steel and copper laps used in the previous tests and also on one of tin made expressly for the purpose. Like the others, the tin lap was made up of a cast iron backing plate provided with a facing of block tin about ¼ in. in thickness.

91 Carborundum alone was used as the abrasive and a uniform pressure of 15 lb. per sq. in. was used on the specimen throughout the tests. In dry lapping much depends on the manner of charging the lap.

92 The first tests were made on the steel lap, charged by rubbing the abrasive into the surface with a cast block 3 in. by 2 in. With a small quantity of carborundum, *F*, and lard oil the surface was worked down until of a uniform slaty color and free from deep scratches or marks left by the grinder. It was then washed clean with gasoline and *used perfectly dry.*

93 Next the plate was charged in the same way but instead of being washed with gasoline, the surface was simply *wiped clean with waste.* This left the plate practically dry but with a light film of oil adhering to its surface.

94 The results of these tests are shown plotted in Fig. 69. It is to be noted that with the plate perfectly clean and dry the cutting drops off quite rapidly after the first 100 rev. At the end of 500 rev.

the total amount ground from the specimen was but 10.6 milligrams. The upper curve, Fig. 69, shows the results when the plate was wiped clean, but not washed with gasoline. In this case the amount ground off at the end of 500 rev. was 63 milligrams or 6 times as much as with the plate perfectly dry.

95 Experiments demonstrated that between these two extremes, *results of any magnitude could be obtained, depending on how thoroughly the free abrasive was removed from the lap surface.* When the lap is simply wiped clean there still remains a film of oil over its surface and this carries a certain amount of free abrasive. This can be readily demonstrated by allowing a few drops of gasoline to strike the plate. As the gasoline spreads, as it does rapidly, a dark fringe appears around its outer edge. This dark fringe is free abrasive. When the lap surface is washed thoroughly with gasoline all free abrasive is removed, leaving only that which is embedded in the surface as the active material.

96 For all subsequent tests the plates were washed thoroughly clean, in order that there should be no variation in results due to a varying amount of free abrasive on the lap surface.

97 It makes a difference, too, whether the abrasive is rubbed or rolled into the surface. For the purpose of comparison, each lap was charged in 4 different ways:

a By rubbing carborundum "F" into the surface with cast block.

b By rubbing carborundum No. 150 into the surface with cast block.

c By rolling carborundum "F" into the surface with steel roller.

d By rolling carborundum No. 150 into the surface with steel roller.

98 For each test three duplicate runs were made and the results averaged. These values are tabulated in Table 7 and are shown plotted in Figs. 70 to 73.

99 The greatest difference due to different charging is shown by the tin lap. When abrasive No. 150 is rolled into its surface the cutting is about two and one-half times as fast as when the same abrasive is rubbed in. Also it is about three times as fast as when abrasive "F" is rolled, and six times as fast as when "F" is rubbed in. The same condition holds true for the copper lap, but not to so great an extent. With this lap the difference between the highest and

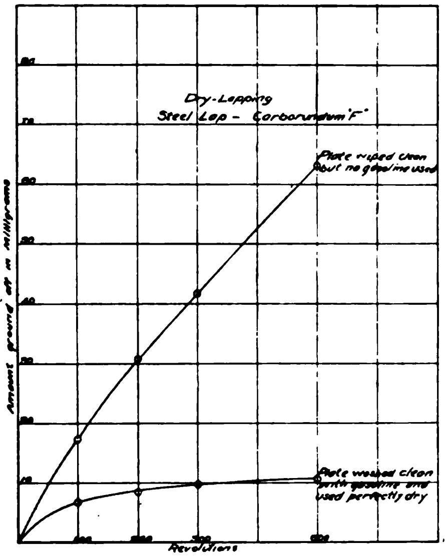
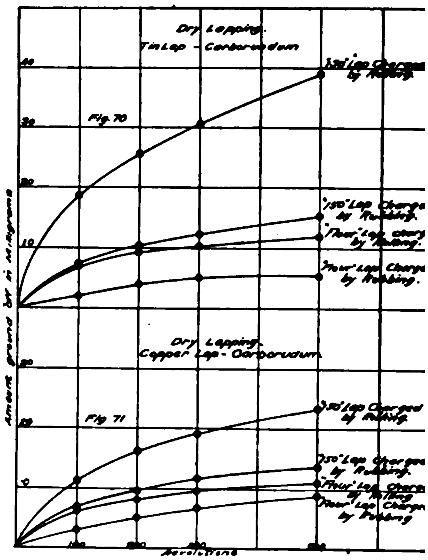
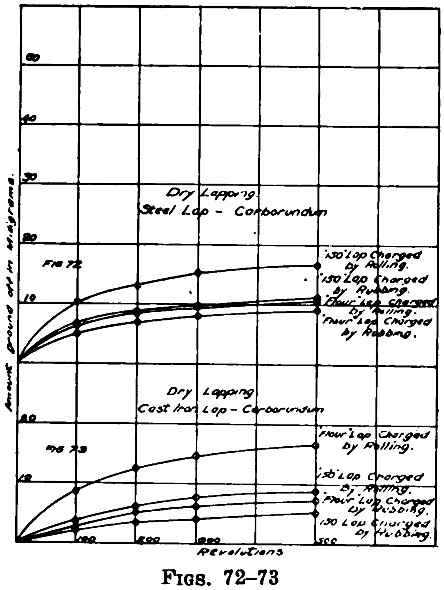


Fig. 69



Figs. 70–71



Figs. 72–73

Dry Lapping, Carborundum, with Steel, Tin, Copper and Cast Iron Laps

TABLE 7 RESULTS OF TESTS ON DRY LAPPING

	Lap	Revolutions 100	200	300	500
		Milligrams ground from specimen			
Carborundum No. 150 lap charged by rolling	Cast	3.6	6	7.6	8.6
	Steel	10.3	13	15.3	16.6
	Copper	11.3	16.3	19	23.3
	Tin	18.6	25.6	30.6	39
Carborundum No. 150 lap charged by rubbing	Cast	2	3.3	4	5
	Steel	6.6	8.6	9.6	11
	Copper	6.6	9.6	11.6	13.6
	Tin	7.3	10.3	12.3	15.3
Carborundum "F" lap charged by rolling	Cast	8.6	12.6	14.6	16.6
	Steel	6.3	8.3	9.3	10.6
	Copper	6	8	9.6	11
	Tin	7	9.3	10.3	12
Carborundum "F" lap charged by rubbing	Cast	2.6	5	6	7
	Steel	5	7	8	9
	Copper	3	5	6.6	8.6
	Tin	2	4	5	5.3

lowest results is about 2.2 to 1, the highest being for No. 150 rolled in, the lowest "F" rubbed in. Rolling a charge of No. 150 abrasive into copper, then, is less effective than rolling into tin. On the steel lap it is still less effective, while on the cast lap this method of charging is least efficient of all.

100 This condition is clearly indicated by the curves in Fig. 74. For these all laps were charged by rolling No. 150 carborundum into the surfaces. With the cast lap, however, the best results were obtained when charged by rolling with grade "F."

101 It thus appears that with soft and ductile material like copper and tin the best results are to be obtained by rolling a comparatively coarse abrasive into the surface, but that with a harder and more brittle material like cast iron a finer grade should be used.

COMPARISON OF THE WET AND DRY METHOD

102 It must be evident that with so many different results, a comparison between the wet and dry methods is more or less unsatisfactory. In dry lapping the rate of cutting decreased rapidly after the first 100 rev. of the machine—much more rapidly than with the wet

method. It seems no more than fair, then, in making comparisons to consider the amounts ground off during the first 100 rev. Further, the highest result obtained with each lap is taken as the basis of comparison. With these data, it is found that with the tin lap, charged by rolling carborundum No. 150 into the surface, the rate of cutting, dry, approaches that of the wet. With the other laps, the rate for dry is about ½ that of the wet. Table 8 exhibits this:

TABLE 8 COMPARISON OF WET AND DRY LAPPING: PRESSURE, 15 LB.; ABRASIVE, CARBORUNDUM; 100 REV. OF MACHINE

	Best results with			
	Cast lap	Steel lap	Copper lap	Tin lap
Wet................	20	24	22	....
Dry................	8.6	10.3	11.3	18.6

103 It may be of interest to know the rate of cutting in linear measure. With the size of specimen used, the removal of 39 milligrams represented a length of 0.001 in. With a pressure of 15 lb. per sq. in., the average of the best results was just about 22 mg. for 100 rev. of the machine. The length of path traversed by the specimen was 36 in., or 3 feet., per rev. Hence, the specimen moved over the lap a distance of 300 ft. to have ground from its surface 0.00056 in., or 0.00019 in. for 100 ft. of travel over the lap surface.

104 With dry lapping on the tin lap, the best result was 18.6 mg. for 100 rev., which gives 0.00016 in. per 100 ft. of travel. This is with a pressure of 15 lb. per sq. in. on the specimen, and, of course, with a higher pressure a greater amount would be ground off.

CONCLUSIONS

105 In order to present the matter in a more usable form, the main facts, as developed by the investigation and deductions therefrom, are here again set forth:

a The initial rate of cutting is not greatly different for the different abrasives.

b Carborundum maintains its rate better than either of the others, alundum next, and emery the least.

c Carborundum wears the lap about twice as fast, and alundum 1¼ times as fast as emery.

d There is no advantage in using an abrasive coarser than No. 150.

e The rate of cutting is practically proportional to the pressure.

f The wear of the laps is in the following proportions:

Cast iron 1.00 Steel 1.27 Copper 2.62

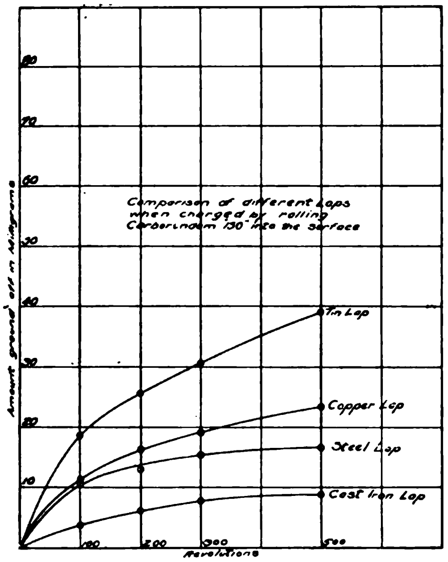


FIG. 74 COMPARISON OF DIFFERENT LAPS CHARGED WITH CARBORUNDUM, No. 150

g This wear is inversely proportional to the hardness by the Brinell test.

h In general, copper and steel cut faster than cast iron, but where permanence of form is a consideration, cast iron is the superior metal.

i Gasoline and kerosene are the best lubricants to use with cast iron lap; kerosene, on account of its non-evaporative qualities, being first choice.

j Machine and lard oil are the best lubricants to use with copper or steel lap. They are least effective on the cast lap.

k For all laps and all abrasives (of those tested), the cutting is faster with lard oil than with machine oil.

l Alcohol shows no particular merit for the work.

m Turpentine does fairly good work with carborundum, but in general is not as good as kerosene or gasoline.

n Soda water compares favorably with other lubricants. Taken as a whole, it is slightly better than alcohol and turpentine.

o Wet lapping is from 1.2 to 6 times as fast as dry lapping, depending on material of the lap and manner of charging.

Acknowledgments

The authors are under obligations for assistance in various ways, and their thanks are hereby tendered to the following: Safety Emery Wheel Co., Springfield, Ohio; Columbus Die, Tool & Machine Co., Columbus, Ohio; The Seagrave Co., Columbus, Ohio; Research Dept., Westinghouse Electric & Mfg. Co., Pittsburgh, Pa.

Also to the following members of the staff of Ohio State University: Professor H. C. Lord, Dept. of Astronomy; Professor J. E. Boyd, Dept. of Mechanics; Professor T. E. French, Dept. of Engineering Drawing; Professor F. E. Sanborn, Dept. of Industrial Arts; Professor F. H. Haskett, Dept. of Architecture; Mr. J. A. Foust, Instructor in Forging.

Appended to the paper was a comprehensive bibliography of the subject of Laps and Lapping. This has been placed on file in the Library of the Society.

DISCUSSION

Charles E. Gillett, in a written discussion, considered that the point which is most forcibly brought out in the paper is that, in lapping, carbide of silicon acts more efficiently on hardened steel than do the aluminous abrasives. In grinding, one of the absolute laws is that aluminous abrasives act more efficiently on materials of high tensile strength, like steel and its alloys, while carbide of silicon is most efficient on materials of a low tensile strength, such as cast iron, brass, bronze, etc. However, lapping may bring into action different properties of the abrasives than is obtained in the solid abrasive wheel.

Considering that the action of an oilstone is like that of a lap which has been charged by either having the abrasive rubbed or rolled into it, the results of a series of tests to compare the cutting qualities of oilstones made of aluminous and carbide of silicon abrasives may be of interest. The only variable in the tests was the composition of the oilstones—one aluminous and three carbide of silicon abrasives being used. The lubricant was a light grade of machine oil.

A summary of the results is: Milligrams removed from chisel per 1000 ft. travel across the oilstones.

Alundum	145
Carbide of Silicon No. 1	33
Carbide of Silicon No. 2	42
Carbide of Silicon No. 3	90

These figures indicate that, in the form of an oilstone, the aluminous abrasive cuts the hardened steel tool much faster than carbide of silicon. The sizes of abrasives used in the oilstones corresponds very closely to the sizes used in the tests in the paper.

The authors state "emery appears to be more brittle and passes through the change quicker than the others, with alundum next and carborundum the least susceptible to such a change." Thus it would seem that in blocking down, where the abrasive grains are compressed between the lap and work, ability to resist crushing is a more important factor than actual hardness or toughness. Emery, being relatively weaker in compression, breaks down into the impalpable sizes faster than the other abrasives, hence does less effective work during the period of test. This presupposes that the coarser sizes of abrasives cut the faster.

With the abrasive held in a ceramic body, such as an oilstone or grinding wheel, the tougher aluminous grains stand up under the shearing action of the test piece longer than the brittle carbide of silicon abrasives. The tougher abrasive grains maintain their cutting edges and remain sharp longer than the brittle abrasives, which soon break down, dull over, become glazed and stop cutting.

An examination of the curves in the paper shows that there is no great variation in the initial rate of cutting of any of the abrasives when tried under similar conditions. Any such slight variation might well be caused by a microscopic change in test conditions. The wide variation in results obtained in wet and dry lapping or from charging the laps by rubbing or rolling shows the wide limit of the possible results that might be obtained.

One of the largest small tool manufacturers in this country experimented with all abrasives available for lapping and found 65F alundum to be the fastest cutting material for the purpose. The carbide of silicon flours were found to cut very fast, but it was impossible to eliminate scratches so deep that they could not be removed under commercial conditions. At the same time the amount of wear on the lap increased noticeably by the use of carbide of silicon.

It would seem that, if carbide of silicon grain is more efficient than the aluminous abrasives in lapping hardened steel, it would be more universally employed in place of the emery and alundum flours which seem to hold the field at present. It may be that the action of loose abrasive grain is materially different than when held in a body; however the cases are few, in actual practice, where carbide of silicon is more efficient than alundum when working on materials of a high tensile strength. With this exception, the authors have shown the action of loose grain abrasives in the fine numbers to be similar to the coarser sizes when used in solid wheel form.

W. A. Knight. It is hardly to be expected that the action of abrasives will be the same when held by a bonding material, as in a wheel or oil stone, and when used in the free state, as on a lap. In the former, three things may happen to an individual grain: (1) It may, in the case of a very weak bond, be torn out bodily from its seat, on the slightest contact with the work. (2) When the bond is stronger it may resist displacement for a time, until its sharp edges or corners are dulled to such an extent that increased friction finally displaces it. (3) With a very hard bond it may resist displacement entirely and the action between the grain and the work become one of wear rather than cut.

In lapping there is but one path through which the grain may pass; it is being continually broken down into smaller and smaller sizes. During this passage from large to smaller sizes, it passes through a size that will give a maximum rate of cutting. This size appears to bear a definite relation to the thickness of oil film between the lap and specimen. As soon as the size of grain passes below this thickness, the cutting grows very much less. It would seem desirable in lapping that this definite size be maintained as long as possible, or that the rate of change of size of grain be low, and carborundum fulfills this condition better than either of the other abrasives.

Experiments were made on quartz sand which show that the initial rate of cutting with this material is not far below that of emery, but that it lasts but a comparatively few revolutions of the lapping plate.

In regard to scratching, it appears natural that a hard abrasive would be more liable to scratch than a softer one. I have been informed that in lens grinding emery or alundum is preferred to carborundum on that account. Still, for lapping hardened steel diamond dust is used for the finest work, and this is conceded to be the hardest of all abrasives.

No. 1489

MEETINGS SEPTEMBER—DECEMBER

THE SEPTEMBER MEETING

A general meeting was held in San Francisco, September 16 and 17, just prior to the International Engineering Congress, 1915, which convened on September 20. The headquarters was the Clift Hotel, and sessions were held in the Hall of the Native Sons of the Golden West.

Members journeying to the meeting from New York traveled on an "Engineers' Special" train arranged for the national engineering societies visiting the Congress and the Panama-Pacific International Exposition. The local committee met this train and escorted the visitors to the headquarters for registration.

The first session was opened on Thursday morning, September 16, by Charles C. Moore, president of the Exposition, extending a welcome on behalf of San Francisco and the State of California, and President Brashear and Vice-President Dickie responded. Two papers on the Exposition were presented.

Following this session, luncheon was tendered the visiting members and guests at the Old Faithful Inn, and a tour of the Exposition was made, terminating in the Court of Abundance, where the Exposition management, through Mr. Moore, presented a commemorative medal to the Society, and Dr. Brashear responded.

On Thursday evening there was an informal dinner of the American Society of Civil Engineers at the Old Faithful Inn, and a dinner dance under the auspices of the American Institute of Electrical Engineers at the Hotel St. Francis, at both of which functions members and guests of our Society were welcomed.

The second professional session was held on Friday morning, when three papers were presented and discussed. All the papers are listed in the program below.

On Friday afternoon and during the remaining days before the Congress, visits and excursions were made to points of engineering interest.

During the meeting all arrangements were in charge of the San Francisco local committee, consisting of F. W. Gay, *Chairman*, F. H. Varney, *Vice-Chairman*, C. F. Braun, *Secretary*, H. L. Terwilliger and J. T. Whittlesey, whose efforts in contribution to the success of the event were highly appreciated.

After the meeting many members and guests stayed in San Francisco for the Congress and for the presentation by the Exposition of a commemorative medal to Dr. Brashear, as Pennsylvania' most notable citizen, on September 22, and for Engineers' Day at the Exposition, on September 24, when the engineers responsibl for the construction of the Exposition were specially recognized.

PROGRAM

Thursday Morning, September 16

Address of welcome at the Native Sons Hall by Charles C. Moore, president of the Panama-Pacific International Exposition, and response by George W. Dickie, Vice-President of the Society.

PROFESSIONAL SESSION

ENGINEERING FEATURES OF THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION, Guy L. Bayley.

MECHANICAL ENGINEERING AT THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION, George W. Dickie.

Thursday Afternoon

Luncheon at Old Faithful Inn.

Trip around the Exposition, terminating at the Court of Abundance with special ceremonies on the part of the Exposition management recognizing the presence of the Society, and with the presentation of a medal to the Society by the president of the Exposition. Response by Dr. Brashear.

Friday Morning, September 17

PROFESSIONAL SESSION

THE HEAVY OIL ENGINE, ITS PRESENT STATUS AND FUTURE DEVELOPMENT, A. H. Goldingham.

Discussed by H. R. Setz, The Author.

THE DIESEL ENGINE AND ITS APPLICATIONS IN SOUTHERN CALIFORNIA, W. H. Adams.

Discussed by H. R. Setz, Roger D. DeWolf, R. W. Crowley, G. W. Dickie, The Author.

THE STRENGTH OF GEAR TEETH, G. H. Marx and L. E. Cutter.

Discussed by L. D. Burlingame, Kate Gleason, The Author.

Friday Afternoon

Technical excursions.

MEETINGS OF SECTIONS

PROVIDENCE, SEPTEMBER 22

Paper: Manufacture of Leather Belting, F. H. Small, chemist, The Graton and Knight Manufacturing Company, Worcester, Mass. Published in THE JOURNAL, December, 1915.

CINCINNATI, SEPTEMBER 30

Joint meeting with Engineers' Club of Cincinnati. Paper: Coal Tar Products and Road Building, John S. Crandall, formerly of the Highway Engineering Department of the Pennsylvania State College.

MILWAUKEE, SEPTEMBER

Illustrated talk on Municipal Waste, Henry A. Allen, Mem. Am. Soc. M. E.

ST. LOUIS, OCTOBER 6

Subject: The Little River Drainage District, Wm. A. O'Brien.

NEW YORK, OCTOBER 12

Paper: Motion Study for Crippled Soldiers by Frank B. Gilbreth, Mem. Am. Soc. M. E. Published in THE JOURNAL, December, 1915.

BOSTON, OCTOBER 13

Joint meeting of Civil, Electrical and Mechanical Engineers, under the auspices of the American Institute of Electrical Engineers. Subject: Load Dispatching as Handled by Large Electrical Power Distributors. Papers by P. Kent of the Boston Edison Company and Mr. Masters of the Boston Elevated Company.

BUFFALO, OCTOBER 20

Address: The Training of Mechanics by W. B. Hunter, director of the Fitchburg (Mass.) High School.

CINCINNATI, OCTOBER 21

Joint meeting with Engineers' Club of Cincinnati. O. Monnett, of the American Radiator Company, described his experiences in connection with the elimination of smoke in Chicago.

MINNESOTA, OCTOBER 21

Paper: History of the Panama Canal, by Quincy A. Hall, Mem. Am. Soc. M. E.

PHILADELPHIA, OCTOBER 26

J. A. Steinmetz, Mem. Am. Soc. M. E., presented an illustrated paper reviewing the development of the aeroplane in France and Germany and describing methods of aeronautical warfare.

ST. LOUIS, OCTOBER 26 AND 27

Under the auspices of the St. Louis Section of the Am. Soc. M. E., the Engineers' Club, together with the associated local societies, held a meeting in honor of Dr. John A. Brashear, President, Am. Soc. M. E. After luncheon at the City Club, Dr. Brashear gave a short address entitled Reminiscences. In the evening Dr. Brashear gave an illustrated talk on the great telescopes of the world and the discoveries made with their aid.

The following day Dr. Brashear addressed the students at Washington University.

PROVIDENCE, OCTOBER 27

Illustrated Address: Modern Highway Pavements, Arthur W. Dow, mechanical engineer of the Dow and Smith Company, New York.

BUFFALO, NOVEMBER 3

Paper: Multiplicity of Cylinders, Mr. Hunt of the Packard Motor Car Company.

NEW YORK, NOVEMBER 9

Paper: An Investigation of Gas Producer Power Plants in New York City and Vicinity, Charles M. Ripley, consulting engineer of New York. Paper and discussion published in THE JOURNAL, December, 1915.

MILWAUKEE, NOVEMBER 10

Paper: High Pressure Boiler Design, Robert Cramer, Mem. Am. Soc. M. E.

LOS ANGELES, NOVEMBER 13

Visit to the laboratory of the Mt. Wilson Solar Observatory. Dr. Hale spoke on the purpose and aims of the Observatory, illustrating his talk with slides. F. G. Pease, Mem. Am. Soc. M. E., member of the Observatory staff, gave an illustrated address on the mechanical equipment and the work of the laboratory.

BUFFALO, NOVEMBER 17

Address: Schoop Process of Spraying Metals, John Calder, Mem. Am. Soc. M. E.

NEW HAVEN, NOVEMBER 17

Fall meeting, with afternoon and evening sessions. Charles L. Warner, Mem. Am. Soc. M. E., presented a paper on the ingenious wire-forming machines made by the Baird Machine Company; Albert A. Dowd, consulting engineer of New York, gave valuable information and shop hints on work-holding devices for lathes, boring mills and planers. At the evening session Ralph E. Flanders, Mem. Am. Soc. M. E., presented a paper on gear cutting machinery and Geo. O. Gridley, Mem. Am. Soc. M. E., closed the session.

CINCINNATI, NOVEMBER 18

Joint meeting with Engineers' Club of Cincinnati. Subject: Electromagnets by C. R. Underhill, chief electrical engineer of the Acme Wire Company, New Haven, Conn.

MINNESOTA, NOVEMBER 18

Afternoon and evening sessions. Papers: Stationary Gas Engines, Ray Mayhew, Mem. Am. Soc. M. E.; Gas Engine Ignition, S. C. Shipley, Mem. Am. Soc. M. E.; Carburetion in Gas Tractor Work, W. G. Clark, engineer for the Wilcox-Bennett Carburetor Company of Minneapolis; Gas Tractor Engines, E. Russell Greer, mechanical engineer, Lion Tractor Company; Use of Special Steel in Gas Tractors and Automobile Construction, S. L. Hoyt, assistant professor of metallography, University of Minnesota.

PROVIDENCE-BOSTON, NOVEMBER 18

Joint meeting of the Providence Association of Mechanical Engineers and the Boston Section of the Am. Soc. M. E., held in Providence. Addresses: Explosives and the Engineer, Prof. Charles

E. Munroe, Dean of George Washington University; Experiences of an Engineer in Public Office, Morris L. Cooke, Mem. Am. Soc. M. E.; The Development of a National Telephone System, M. C. Rorty, engineer of the American Telephone and Telegraph Company. Account published in The Journal, December, 1915.

CHICAGO, NOVEMBER 19

Lecture: Helping Crippled Soldiers, by Frank B. Gilbreth, Mem. Am. Soc. M. E. Hon. Jacob M. Dickinson spoke of the urgent need of work of helping those disabled in the European War.

ATLANTA, NOVEMBER 22

Meeting of Affiliated Technical Societies of Atlanta. Illustrated lecture: Steel Making from the Ore to the Finished Product by Mr. Speller of the National Tube Works, McKeesport, Pa. J. E. Latta, of the Underwriter's Laboratory of Chicago, Ill., showed moving pictures of the testing work of the laboratory.

PHILADELPHIA, NOVEMBER 23

Paper: Industrial Safety and Principles of Management, by W. P. Barba, Mem. Am. Soc. M. E. Published in this volume.

WORCESTER, NOVEMBER 23

Paper: Motion Study for Crippled Soldiers, Frank B. Gilbreth, Mem. Am. Soc. M. E.

ST. LOUIS, NOVEMBER 24

Joint meeting of the Associated Societies of St. Louis, under the auspices of the American Society of Civil Engineers. Illustrated lecture: Pearl Harbor Dry Dock, H. R. Stanford, U. S. N., chief of the Bureau of Yards and Docks.

BOSTON, NOVEMBER 30

Paper: Motion Study for Crippled Soldiers, Frank B. Gilbreth, Mem. Am. Soc. M. E.

BUFFALO, DECEMBER 1

Lecture: Internal Conveyors, Fay B. Williams, Mem. Am. Soc. M. E.

ST. LOUIS, DECEMBER 1

Joint meeting under the auspices of the American Society of Engineering Contractors. Paper: Equitable Specifications and Contracts, Hillis F. Hackedorn, president of the Hackedorn Contracting Company of Indianapolis. Published in THE JOURNAL, May, 1916.

BUFFALO, DECEMBER 15

Address: Engineers in Politics, C. E. Drayer, Secretary of the Cleveland Engineering Society. Published in THE JOURNAL, April, 1916.

MILWAUKEE, DECEMBER 15

Address: The Relationship That Should Exist between the Engineers' Society and the Administration of Municipal Affairs, W. B. Hanlon, president of the Cleveland Engineering Society.

PROVIDENCE, DECEMBER 15

Illustrated address: Geology and Engineering, Prof. Charles W. Brown, head of the department of geology, Brown University.

CINCINNATI, DECEMBER 16

Joint meeting with Engineers' Club of Cincinnati. F. L. Raschig discussed the paper on Engineering Features of the Panama Pacific International Exposition by Guy L. Bayley, Mem. Am. Soc. M. E., published in this volume.

MINNESOTA, DECEMBER 16

Meeting held in St. Paul. Papers: Engineering Education in the British Isles, Prof. J. J. Flather, Mem. Am. Soc. M. E.; Government Specifications, Harry L. Brink, Mem. Am. Soc. M. E.

ST. LOUIS, DECEMBER 18

Dinner Meeting. Paul Brown of the *St. Louis Republic* spoke on The Economic Development of the American Railway.

THE ANNUAL MEETING

At the thirty-sixth Annual Meeting of the Society held in the Engineering Societies Building, New York, December 7 to 10, 1915, there was a record attendance of 1437, of which 819 were members. The visiting members came from every section of the country and included delegates from fourteen local sections, two of them being from California. Twenty-nine papers, seven of which were contributed by local sections and thirteen, including reports, by professional committees, were presented and discussed.

An exceptionally large audience attended the opening session on Tuesday evening, December 7, when Dr. Brashear delivered his presidential address on Science in its Relation to Engineering. The address was followed by a presidential reception by the Society.

On Wednesday morning a business session was held, followed by a professional session. On Wednesday afternoon three simultaneous professional sessions were held. Two simultaneous professional sessions were held on Thursday morning and one on Friday morning. A program of the sessions giving the titles of papers is appended.

As a mark of respect to the memory of the late Dr. Frederick W. Taylor, Past-President of the Society, who died on March 21, 1915, the regular business of the Wednesday morning session was suspended at 11 a.m. for a period of an hour, during which a memorial meeting to Dr. Taylor was held. Brief addresses were given by Henry R. Towne, Past-President Am. Soc. M. E., H. L. Gantt and Rear-Admiral Casper F. Goodrich, all intimate friends and associates of Dr. Taylor.

In the matter of entertainment, an innovation was a smoker held in the rooms of the Society on Wednesday evening; this was a members' get-together meeting in charge of the New York Local Committee and proved to be a great success. A reception, under the auspices of the Ladies' Committee, was held on Wednesday afternoon, and on Thursday Mrs. Harrington Emerson entertained the ladies at luncheon at the Hotel Astor.

The annual reunion of members and guests took the form of a dinner and dance at the Hotel Astor on Thursday evening.

On Wednesday and Thursday afternoons excursions were conducted by members of the Excursion Committee to selected points of engineering interest in New York and vicinity.

Ten college reunions were held this year on Friday evening, which has now come to be regarded as "College reunion night."

Due to the efforts of the various committees in charge, the annual meeting was a very successful one. The chairmen of the committees were: Committee on Meetings, John H. Barr; New York Section Committee, Edward Van Winkle; President's Reception, S. D. Collett: Acquaintanceship, John P. Neff; Dinner, J. J. Swan; Excursion, E. J. Prindle; Ladies' Reception, Mrs. Harrington Emerson.

PROGRAM

Tuesday Evening, December 7

Opening session. President's address: Science in its Relation to Engineering, Dr. John A. Brashear. Report of tellers of election of officers and introduction of the President-elect.

Reception by the Society to the President, President-elect, ladies, members and guests.

Wednesday Morning, December 8

Business Meeting

Reports of the Council, Standing Committees and Special Committees, constitutional amendments and new business.

Frederick W. Taylor Memorial meeting.

PROFESSIONAL SESSION

Design of Fire Tube Boilers and Steam Drums, F. W. Dean.

Discussed by W. F. Kiesel, Jr., Thos. E. Durban, The Author.

A Novel Method of Handling Boilers to Prevent Corrosion and Scale, Allen H. Babcock.

Discussed by L. M. Booth, F. F. Vater, Geo. H. Gibson, M. F. Newman, Walter M. McFarland, Howard Stillman, Wm. Kent, E. N. Trump, Frederick E. Geibel, J. F. Walsh, The Author.

Gas Producers with By-Product Recovery, Arthur H. Lymn.

Modern Electric Elevator and Elevator Problems, David Lindquist.

The Application of Engineering Methods to the Problems of the Executive, Director and Trustee, Hollis Godfrey.

Turbines vs. Engines in Units of Small Capacities, J. S. Barstow.

The Connors Creek Plant of the Detroit Edison Company, C. F. Hirshfeld.

Proportioning Chimneys on a Gas Basis, A. L. Menzin.

Discussed by A. G. Christie, The Author.

Wednesday Afternoon

SIMULTANEOUS PROFESSIONAL SESSIONS

TEXTILE SESSION

Higher Steam Pressures, Robert Cramer.

Discussed by Wm. Kent, R. J. S. Pigott, G. I. Rockwood, R. H. Rice, W. N. Polakov, Carl C. Thomas, D. S. Jacobus, The Author.

Heating by Forced Circulation of Hot Water in Textile Mills, Albert G. Duncan.

Discussed by Chas. H. Bigelow, F. W. Parks, A. F. Ernst, The Author.

Relative Value of Private and Purchased Electric Power for Textile Mills, Frank W. Reynolds and Dan Adams.

Discussed by Fred N. Bushnell, John A. Stevens, R. J. S. Pigott, F. J. Bryant, A. L. Williston, Chas. H. Bigelow, W. N. Polakov, Dan Adams.

The Engineer and the Business of Fire Insurance, Jos. P. Gray.

MACHINE SHOP SESSION

Automatic Mechanical Control of Lathes and Screw Machines, L. D. Burlingame.

Discussed by C. M. Conradson, Ralph E. Flanders, Elmer H. Neff, Norman Marshall, H. K. Hathaway, The Author.

Electric Operation and Automatic Electric Control for Machine Tools, L. C. Brooks.

Discussed by H. D. James, H. F. Stratton, C. D. Knight, H. K. Hathaway, H. J. Eberhardt, Ralph E. Flanders, Elmer H. Neff, The Author.

Safety Code for the Use and Care of Abrasive Wheels.

RAILROAD SESSION

Operation of Parallel and Radial Axles of a Locomotive by a Single Set of Cylinders, Anatole Mallet.

Discussed by E. A. Averill, Carl J. Mellin, W. F. Kiesel, Jr., G. R. Henderson, W. E. Woodard, Geo. L. Fowler, E. B. Katte, Roy V. Wright, C. D. Young, S. M. Vauclain, F. J. Cole, W. F. M. Goss, The Author.

Four-Wheel Trucks for Passenger Cars, Roy V. Wright.

Discussed by G. R. Henderson, S. G. Thomson, L. R. Pomeroy, Geo. W. Rink, Alphonse A. Adler, E. B. Katte, C. D. Young, The Author.

Wednesday Afternoon

Reception and Tea given by the Ladies' Committee in the rooms of the Society.

Technical Excursions.

Wednesday Evening

Smoker held in the rooms of the Society.

Thursday Morning, December 9

SIMULTANEOUS SESSIONS

POWER PLANT SESSION

The Heat Insulating Properties of Commercial Steam Pipe Coverings, L. B. McMillan.

Discussed by Leonard Waldo, F. M. Farmer, Herbert N. Dawes, A. M. Greene, Jr., L. R. Ingersoll, C. M. Sames, The Author.

Performance and Design of High Vacuum Surface Condensers, Geo. H. Gibson and Paul A. Bancel.

Discussed by Leo Loeb, R. N. Ehrhart, The Authors.

CIRCULATION IN HORIZONTAL WATER TUBE BOILERS, Paul A. Bancel.

Discussed by Geo. L. Fowler, Wm. Kent, Hosea Webster, John C. Parker, A. M. Greene, Jr., A. A. Cary, The Author.

UNIQUE HYDRAULIC POWER PLANT AT THE HENRY FORD FARMS, Mark A. Replogle.

Discussed by Clemens Herschel, R. L. Daugherty, The Author.

MISCELLANEOUS SESSION

THE FLOW OF AIR THROUGH THIN-PLATE ORIFICES, Ernest O. Hickstein.

Discussed by P. F. Walker, E. D. Leland, H. B. Bernard, G. T. Voorhees, J. T. Wilkin, C. C. Thomas, S. A. Reeve, A. M. Greene, Jr., The Author.

ELASTICITY AND STRENGTH OF STONEWARE AND PORCELAIN, James E. Boyd.

Discussed by Ralph D. Mershon, R. C. Carpenter, L. E. Barringer, F. M. Farmer, T. D. Lynch, John F. Ancona, Elliott H. Whitlock, Percy H. Thomas, John A. Brashear, The Author.

FOUNDATIONS, Charles T. Main.

Discussed by M. M. Upson, Chas. H. Bigelow, A. G. Monks, Sanford E. Thompson, F. A. Waldron.

OIL ENGINE VAPORIZER PROPORTIONS, Louis Illmer.

Discussed by Wm. T. Price, The Author.

Thursday Afternoon

Luncheon of the delegates to Conference of Local Sections and Chairmen of Sub-Committees on Increase of Membership in club room of Craftsman Building.

Luncheon at the Hotel Astor given for the ladies of the Society by Mrs. Harrington Emerson.

Technical Excursions.

Thursday Evening

Dinner and Dance at Hotel Astor.

Friday Morning, December 10

PROFESSIONAL SESSION

INDUSTRIAL SAFETY SESSION

STANDARDIZATION OF SAFETY PRINCIPLES, Carl M. Hansen.

Discussed by Frederick R. Hutton, Frank E. Law, John H. Barr, John W. Irwin, W. M. Kidder, David S. Beyer, Jas. O. Gibbons, Geo. M. Price, The Author.

MODERN MOVEMENT FOR SAFETY FROM STANDPOINT OF MANUFACTURER, Melville W. Mix.

Discussed by L. D. Burlingame, Frederick R. Hutton, Frank E. Law, John H. Barr, Leonard Waldo, F. D. Patterson.

THE ATTITUDE OF THE EMPLOYER TOWARDS ACCIDENT PREVENTION AND WORKMEN'S COMPENSATION, W. H. Cameron.

Discussed by John P. Jackson, Chas. W. Baker, Lew R. Palmer.

INDUSTRIAL SAFETY AND PRINCIPLES OF MANAGEMENT, W. P. Barba.

No. 1490

ENGINEERING FEATURES OF THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION

By Guy L. Bayley, San Francisco, Cal.

Member of the Society

An exposition is a modern city built to order and as such must be provided with every utility and convenience with which the public is familiar. Even though the existence of the exposition be short, none of these may be omitted or restricted. In the planning and construction of such a project all branches of engineering play a part and, while there are many details which require special treatment, the real engineering problem lies in the modifying of standard practice to secure the temporary service desired at the lowest possible cost, with the use of such material as will facilitate the final dismantling of the exposition and the realization of a high salvage value.

2 It is not within the scope of this paper to describe the organization and methods used to construct, on time and within the engineers' estimates, a project costing close to $15,000,000, but rather to describe briefly the more important engineering features of it. Detailed information is given in but few instances as most of the features have been covered by articles which have appeared in the technical press, and other material is in the course of preparation, so that a fairly complete record will be left of the engineering of this Exposition.

3 For the benefit of future exposition builders it is planned to file with the Library of the Society copies of all Exposition reports together with data as to estimates, construction methods, costs and load curves covering the use of various utilities—guiding information which would have been invaluable in the present instance if it had been available.

Presented at the Panama-Pacific International Exposition Meeting, San Francisco, September, 1915, of The American Society of Mechanical Engineers.

HYDRAULIC FILLS

4 At the time the Exposition took possession of its site, that portion of the grounds to be occupied by the main buildings (about 70 acres) was under 12 ft. of water at mean high tide, and it was necessary to fill this section and, in addition, the low lands within the Presidio (a U. S. Military reservation). The filling was done by suction dredges which operated off-shore a distance of about 300 ft. in depths of from 30 to 50 ft. The discharge carried 8 per cent to 10 per cent of solid matter which ran 60 per cent to 70 per cent sand, the balance being sea mud and silt. The dredged material was handled so that it displaced about 5 ft. of soft ooze which existed in the bottom of the basin, the ooze being dissolved and carried out through the waste gates. In order to expedite the work of removing the soft portions of the original bottom, water instead of a mixture was pumped at intervals. To reclaim this portion of the site required the pumping of 1,300,000 cu. yd. of material at a cost of $218,000.

5 Shortly after the fill had been completed, settlement stakes were driven and readings taken to establish the rate of settlement, which was rapid at the start and decreased shortly to a slow but uniform rate. After the initial settlement had taken place borings were made at the sites of all buildings of the main group to determine the character of the underlying material and depth of hardpan. Test piles were driven to ascertain the supporting value of the ground at various depths. These tests showed that the dredger fill not only had no supporting value, but actually produced a load on the piles driven through it, tending to drag the piles down at the speed of its own settlement. As the result of a large number of tests, it was decided not to depend upon the thin strata of blue mud and clay underlying the dredger fill but to drive the piles to a penetration of 1 in. to the blow of a No. 1 Vulcan steam hammer in the hard layer of green sand and clay overlying the hardpan.

6 In the areas not affected by the dredger fill, there was a layer of overlying soft clay, and tests showed that short piles terminating in the sand offered practically as much resistance as those driven through the sand into the clay. The sand had a supporting value of 3000 lb. per sq. ft. and spread footings were considered but it was found that short piles averaging 14 ft. provided a cheaper and simpler construction. It was also thought that the adoption of pile footings would offer greater resistance in case of earthquake, previous ex-

perience having indicated that structures were most affected when resting on filled ground by means of spread footings.

7 The decision not to drive into the substratum of soft clay resulted in a great saving. The results of these tests were of the greatest value as they provided definite data for the design of foundations. The information gained was furnished to bidders and naturally resulted in lower prices, as contractors were able to know the lengths of piling required and thus avoid unnecessary waste. The total number of piles driven was 15,654 and the linear feet 645,692. The average cost was 24½ cents per lin. ft. below cut-off, and the average length of pile 41.2 ft.

8 The dredger fill in the Presidio was shallow, not exceeding 6 ft. at any point. The total amount pumped was 400,000 cu. yd. which reclaimed 114 acres of land at a cost of $84,000. The buildings in this area are light frame structures and as a rule rest on spread footings.

STRUCTURAL DESIGN

9 The structural details of most of the buildings have been fully covered in the technical press and only the salient points in their design will be mentioned here.

10 With the exception of the frames of the Tower of Jewels and the Palace of Fine Arts and the dome of the Palace of Horticulture, the buildings are timber structures. All the buildings, except Festival Hall, have pile foundations for the framework. The floor sub-structures of those buildings located on the dredger fill have pile foundations and the others spread footings or mud sills, depending upon the nature of the ground. The pile foundations for the Tower of Jewels are capped with reinforced concrete. Prior to the decision on the type of foundations to be used in each instance, tests were made of the ground on which the buildings were to be erected. These tests gave accurate data as to the length of piles required and the allowable loading. The piles varied from 13 to 75 ft. in length, although a few 120 ft. long were required under the Transportation building. The safe carrying capacity of one pile was found to be 20 tons.

11 From an engineering viewpoint, the Tower of Jewels is probably the most interesting structure on the grounds. It is 435 ft. high and 120 ft. square in plan from the ground level to elevation 152 ft., from which level to elevation 335 ft. it takes the form of a truncated pyramid. From elevation 335 ft. to 364 ft., the frame is made up of four vertical columns 20 ft. apart and braced with rods. Above this

elevation the frame forms a tower 8 ft. by 8 ft. in plan to elevation 397 ft., from which point runs a central post which supports a ball 17 ft. in diameter. The framing of the tower required 1403 tons of structural steel.

12 The glass dome of the Palace of Horticulture has an extreme height of 185 ft. and is 152 ft. in diameter. The upper portion has the form of a half sphere while the lower portion is cylindrical. The supporting frame of the dome was figured as a true dome and consists of 24 steel ribs tied together by 11 horizontal rings. The dome and cylinder are carried on plate girders and trusses which in turn are supported by 8 structural piers 65 ft. in height.

13 The framing of the dome of Festival Hall presented many difficulties owing to the size of the dome and the large arched openings on two sides. The curved dome was built on a supporting pyramidal dome 140 ft. in diameter with 16 trussed ribs. This dome was carried by four main piers and 8 columns. The ribs occurring over the arched openings were carried by two trusses of 77-ft. span. The pyramidal dome was designed as a true dome, the ring tension at the supports being cared for by two $3\frac{1}{8}$-in. rods. The dome roof with a diameter of 172 ft. was built of sheathing supported by 2-in. by 6-in. rafters resting on studded walls which were carried by beams spanning the main ribs. The exterior ceiling was suspended from the dome ribs.

14 For a frame structure the Palace of Machinery claims attention on account of its size—968 ft. by 368 ft. with a height of 136 ft. in the transverse bays and 120 ft. in the longitudinal bays. The spans of the longitudinal and transverse bays are 75 ft. The arched trusses give the building an attractive appearance and the framing is of interest, particularly at the intersections of the longitudinal and transverse bays. The columns of the longitudinal bays were designed to support traveling cranes carrying loads up to 30 tons.

15 Owing to fire protection requirements, the frame of the Palace of Fine Arts was built of steel. The building is curved in plan, 950 ft. long and 135 ft. wide. Three hinged arches were used for this building, the span being 131 ft. and the height from the floor to the center hinge 48 ft. Steel channels placed beneath the floor served as ties for the lower hinges. Light bracing trusses were used to connect the arches and form supports for the purlins. The roof and walls were constructed with cement-plaster on metal fabric, the roof with a thickness of 3 in. and the walls $2\frac{1}{2}$ in.

16 The Auditorium, located in the Civic Center, was built at an expense of over $1,000,000 and is a permanent fireproof structure of

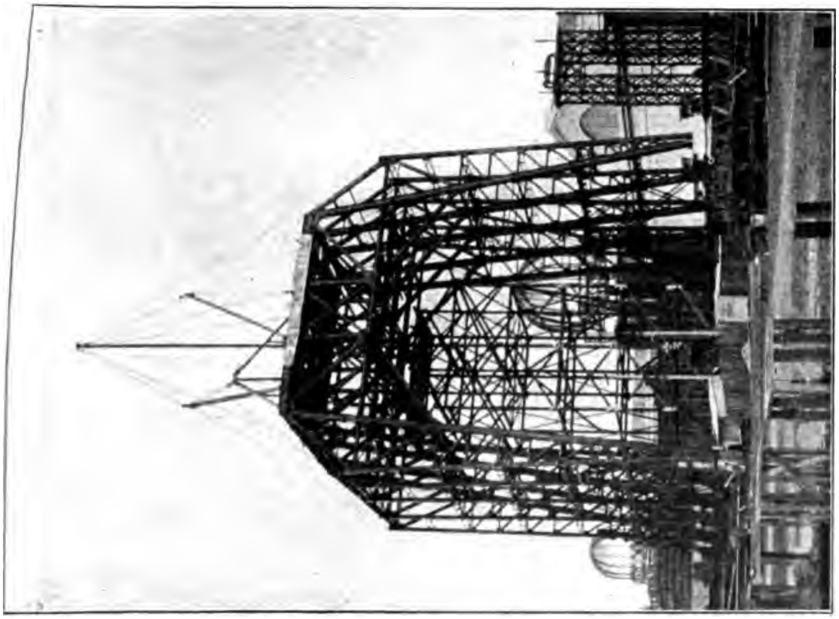


FIG. 1 STEEL FRAME OF TOWER OF JEWELS TO ELEVATION 152 FT.

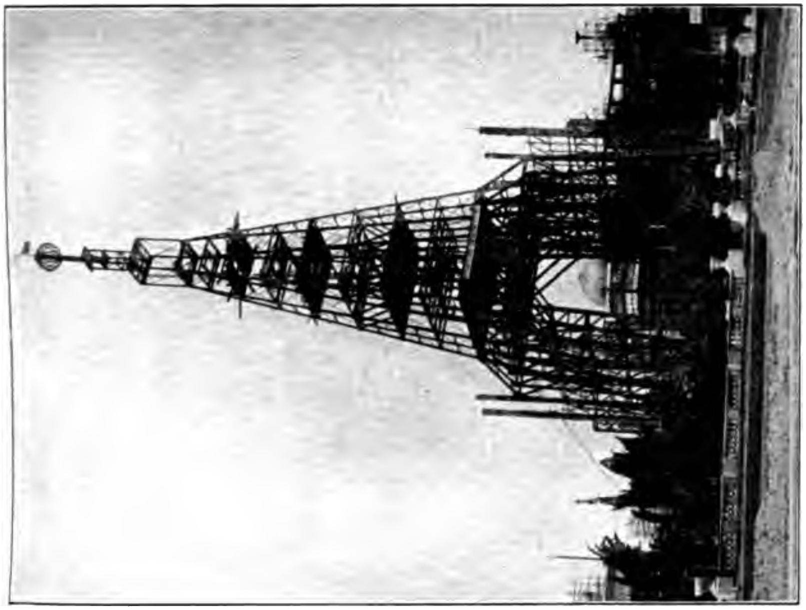


FIG. 2 COMPLETED STEEL FRAME OF TOWER OF JEWELS

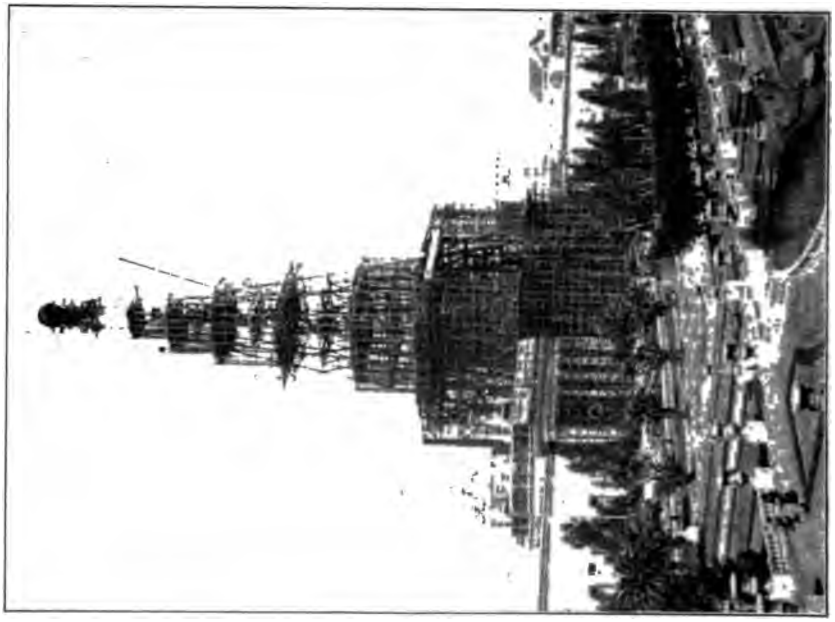


Fig. 3 Timber Frame over Steel Frame Tower

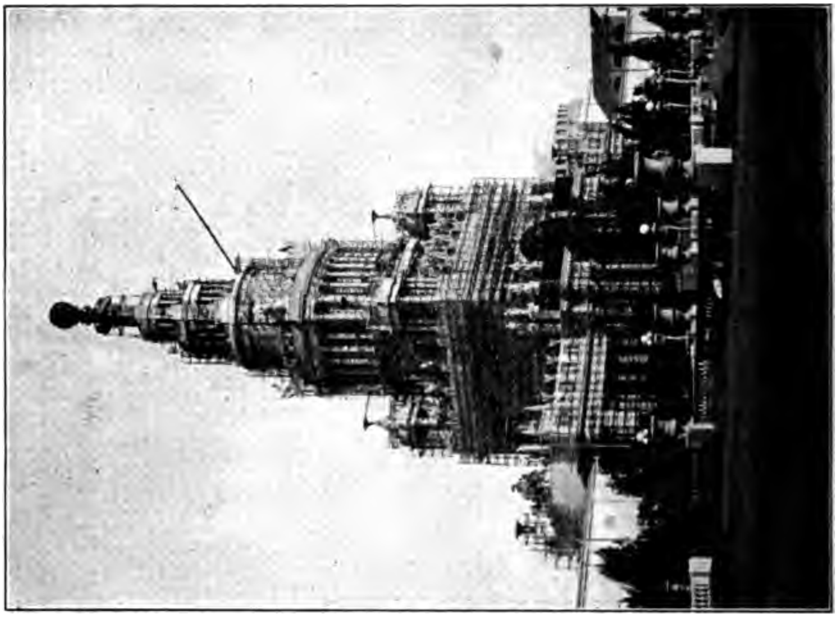


Fig. 4 Tower of Jewels Plaster Work being Applied

FIG. 5 HOISTING ERECTION METHOD, PALACE OF HORTICULTURE DOME

FIG. 6 CONNECTION OF RIBS TO GIRDERS, PALACE OF HORTICULTURE DOME

FIG. 7 STEEL DOME OF PALACE OF HORTICULTURE

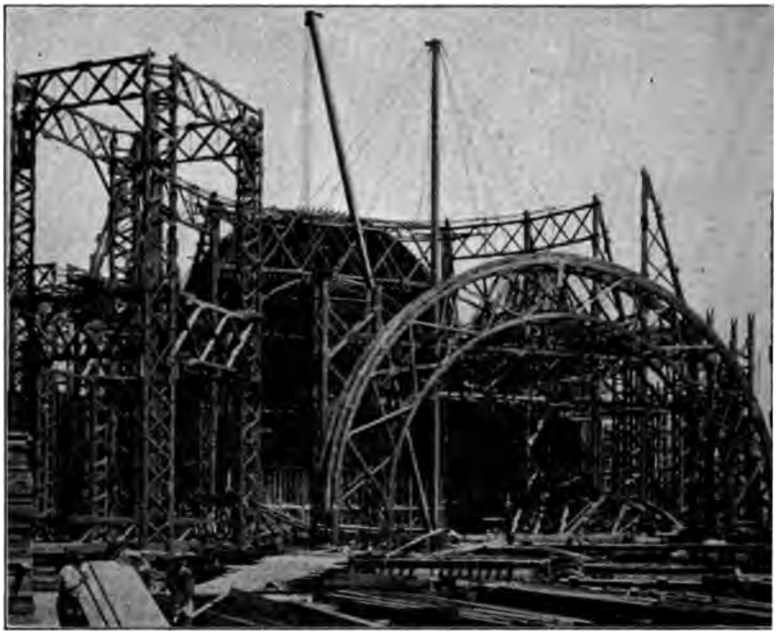


FIG. 8 FRAMEWORK OF FESTIVAL HALL

FIG. 9 DOME AND ARCHED OPENING, FESTIVAL HALL

FIG. 10 FESTIVAL HALL PARTIALLY COMPLETED

FIG. 11 COMPLETED DOME OF PALACE OF HORTICULTURE

FIG. 12 TYPICAL TIMBER DOME CONSTRUCTION

FIG. 13 FRAMING OF HALF DOME, FOOD PRODUCTS BUILDING

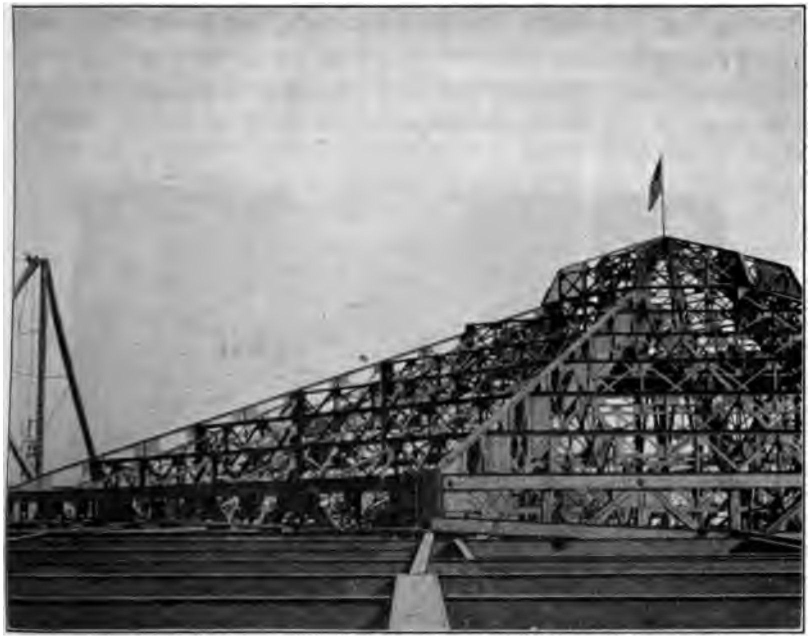


FIG. 14 DOME OF AUDITORIUM

FIG. 15 FRAMING OF HALF DOME, PALACE OF EDUCATION

four stories, occupying a city block 200 ft. by 187 ft. The main hall has a seating capacity of 10,650, and there are 10 halls or large rooms for conventions. Flanking the main hall on each side and extending the height of two stories are banquet rooms 56 ft. by 136 ft. The only

FIG. 16 INTERIOR OF DOME, MINES BUILDING

feature of the building that is unusual is the dome, which may be described as a truncated octagonal pyramid with a maximum diameter of 205 ft. 6 in. and a height of 40 ft. The height of the spring line of the dome from the floor is 79 ft. The dome framework is of interest, not only on account of its great diameter, but also from the fact that it was designed as a true dome. The 8 main trussed ribs are 6 ft. deep at the upper end and 10 ft. at the lower, and are carried by 8 columns. The horizontal thrust at the bottom is taken up by four $1\frac{7}{8}$-in. by 6-in. eyebars. The supporting columns are connected by trusses having a depth of 14 ft.

17 There are no unusual features in the structural design of the eight buildings which form the main group, although the framing of the domes and half domes were interesting problems on account of the use of timber where steel would have been used in ordinary

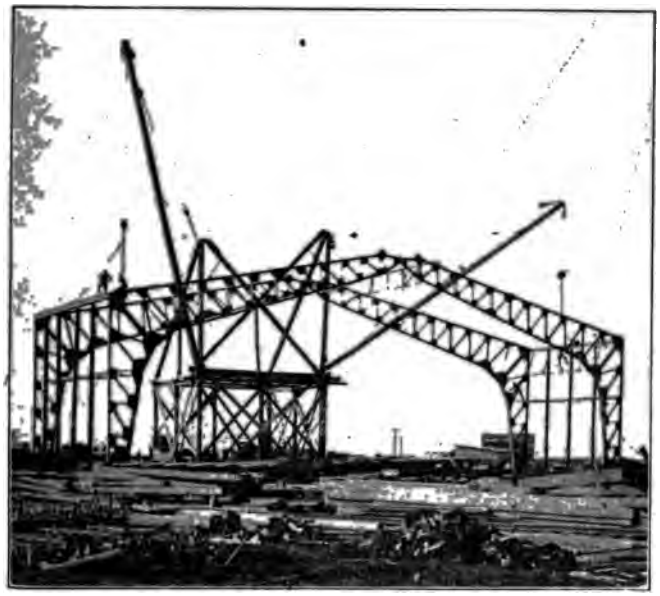


FIG. 17 THREE-HINGED ARCH CONSTRUCTION, PALACE OF FINE ARTS

practice. The domes are 100 ft. in diameter and 162 ft. high, each with 32 ribs resting on a circular girder which in turn is supported by four girders with connecting diagonals at the corners, thus providing eight points of support for the ring. Half domes 112 ft. high and 72 ft. wide form the entrances to the Palaces of Food Products and Education and are constructed of double, 3-hinged timber arches for carrying the walls and arched roof, and a steel 3-hinged arch for carrying the trussed ribs. The construction of the walls is quite different from that employed at previous expositions in that a single line of 2-in. by 6-in. studs, 18-in. centers, are used with horizontal girders every 13 ft., instead of the usual double line of studs laced

together. The floors in these buildings, as elsewhere, are constructed of 2-in. shiplap. Where spread footings were used, the floor was laid on 2-in. by 8-in. joists, 2-ft. centers with 9-ft. spans, supported by

FIG. 19 FRAMING OF PALACE OF MACHINERY

6-in. by 8-in. girders with 9-ft. spans, while with pile footings 2-in. by 12-in. joists are used 2-ft. centers, 14-ft. spans, with 10-in. by 16-in. girders, 20-ft. spans.

18 The brevity with which this phase of the work has been treated

is no index to its importance, as the making of the structural designs formed a large part of the engineering work. Necessarily, once the elements of a design had been worked out, there was much duplication, but a great deal of skill and originality were shown in applying to timber framing much of the knowledge which has been acquired in recent years relative to steel frame structures.

19 At the start rules were made governing all structural work in which the allowable loads, stresses and other designing data were established, with the result that the designs were consistent throughout. Some idea of the volume of work handled by the structural bureau may be gained from the number of drawings made, there being 781 sheets covering an area of 13,277 sq. ft., from which 27,355 prints were made. When the work was at its height 70 structural draftsmen were employed by the structural bureau.

SEWER SYSTEM

20 Use was made of a number of city sewers which crossed the site, one of which was large enough to be used as the main outfall for all sanitary sewage. A separate system was installed for storm water, with frequent connections to the bay. The plan of using separate sewers was adopted as it was permissible to place the storm sewer near the surface and use a cheap form of construction. The amount of storm water to be cared for was figured from the San Francisco rainfall rate curve, which gives a rate of 2.16 in. per hour for a 5-min. interval and a rate of 0.598 in. for a 60-min. interval. The handling of the sanitary sewage in the low area within the Presidio necessitated a pumping plant, which was located near the center of the district and discharged into the main outfall through 1170 ft. of 20-in. wood stave pipe.

21 The sewer system was simple and direct and was based on the use of second quality vitrified pipe in sizes from 8-in. to 15-in. and banded wood stave pipe for sizes up to 30-in., except that in the States and Foreign sites, where watertightness was a requirement, wood stave pipe as small as 10-in. was used. The sewer system, including catch basins, cost $142,000 and comprised some 28 miles of pipe.

TRANSPORTATION

22 No engineering features of importance were associated with the transportation problem, but in view of the fact that the handling of building material and exhibits proved to be one of the largest tasks

in the construction of the Exposition, the methods used will be described.

23 Owing to the location of the Exposition being along the shore of the Bay of San Francisco, it was possible to unload lumber direct from vessels to the Exposition's wharves, and as lumber formed the bulk of the material to be handled the problem was much simplified. The lumber came from Oregon and Washington ports in steam

FIG. 20 UNLOADING LUMBER AT EXPOSITION WHARF

schooners, and wharf facilities were provided to accommodate five vessels unloading at one time. The principal problem was to get the lumber away from the wharves which provided little or no storage, and the plan adopted was to use 2-wheeled lumber trucks upon which the lumber was landed by ship's tackle. Single horses were used to drag the trucks, the driver guiding the truck by hand.

24 The unloading rate per vessel reached 30,000 ft. per hour at times, and to care for this quantity of lumber required some 200 trucks, 68 horses and 140 men. One of the wharves was provided with track facilities for unloading direct on to cars in case it should be found impossible to handle the lumber fast enough with horses and trucks, but this occasion never arose.

25 Plank roads were laid around the four sides of each building site and over these the lumber was hauled to the adjacent storage space. The lumber was so ordered that an entire cargo would be for one designated building and there was, therefore, no necessity for sorting on the wharf. The work of receiving, hauling and rough-piling the lumber was let to contract at prices varying from 35 cents per 1000 ft. for short hauls to $1.10 per 1000 ft. for long hauls. The total amount of lumber handled under contract was 68,131,000 ft. and the amount paid to the contractor for this work was $68,000. The plank roads referred to consisted of 3-in. planks 16 ft. long laid on 3-in. sills, and were laid by the exposition's forces, approximately 1,950,000 ft. of lumber being used for this purpose alone. The plank roads were an excellent solution of the difficulty incident to extensive building operations over an area which contained no roadways, and while their cost was considerable, in view of their effectiveness and later use for protection of permanent roadways, the investment was a wise one.

26 In addition to the amount of lumber unloaded by contract there was handled over the wharves, in the manner described, about 9,500,000 ft. of lumber. The balance of the lumber used by the Exposition and its participants was purchased locally and delivered by teams and wagons, and amounted to about 32,000,000 ft.

27 Building materials other than lumber have been estimated as being about 800,000 tons. To handle these enormous quantities of materials expeditiously and without confusion required careful planning of temporary roadways and the systematic routing of traffic.

28 The Exposition Terminal Railway was built primarily for handling exhibits, but was used to advantage during the construction period for distributing 36,000 tons of construction material and 5,000 tons of general freight received on cars. This road was built by the exposition's forces and included 11½ miles of standard track, 3½ miles of which was laid within the exhibit buildings for unloading direct from the cars to the exhibitors' spaces. A spur track was run to the U. S. Transport Dock so that freight from ocean-going vessels could be loaded on the exposition's cars. Cars from the various transcontinental lines were brought to the grounds on barges, for the accommodation of which the Exposition built a freight slip.

29 The design of this slip follows the standard practice of the San Francisco Bay district in that the bridge forms a span between the boat and the shore. The bridge is hinged on the shore end, while the outboard end is suspended by cables which pass over sheaves to suitable counterweights. The counterweights are sufficient to over-

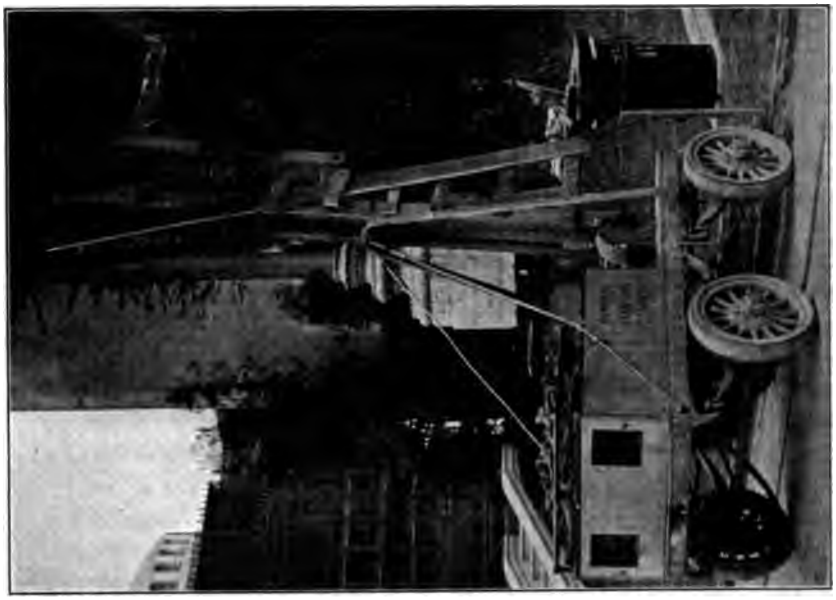
balance the weight of the bridge and eliminate danger of dropping the bridge into the bay. To lower the bridge that portion of the counterweights required for overbalancing is raised by hydraulic means and the bridge is allowed to land on the shelf or recess provided in the bow of the barge. In operation, the live load on the bridge is divided between the boat and the shore supports and the gallows frame carries only the weight of the bridge and counterweights—about 400,000 lb.

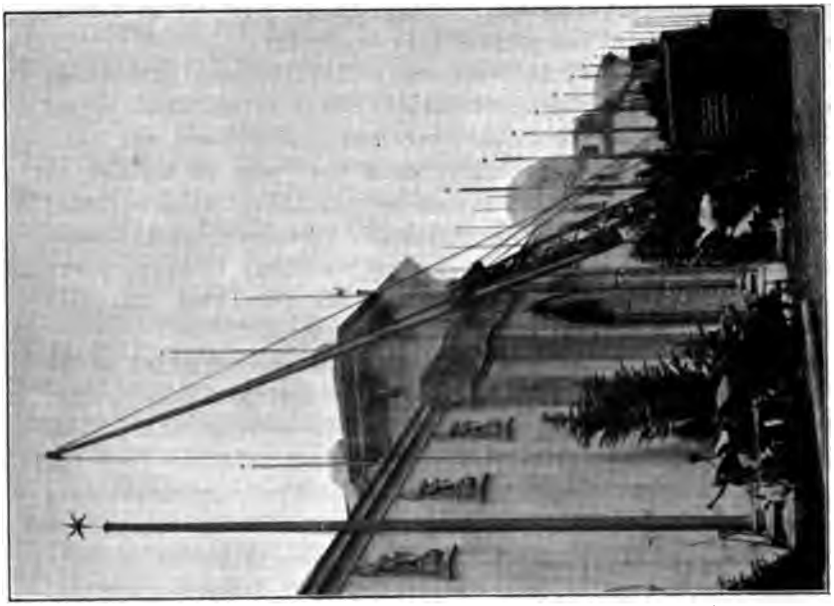
30 The hydraulic equipment consists of a vertical cylinder 28 in. in diameter and 12 ft. long, a motor-driven triplex pump and an accumulator. Cables run from the crosshead on the piston rod to the two auxiliary counterweights which rest on the main counterweights when the bridge is in the raised position. The water supply to the cylinder is controlled by a piston type valve fitted with cup leathers. This valve is operated by hand, but in case of overtravel of the bridge is automatically closed. The accumulator controls the motor operating the pump so as to maintain a constant pressure of 90 lb. per sq. in.

31 The bridge is equipped with three tracks and designed for Cooper's E-50 loading. It is 24 ft. in width and 100 ft. in length. Some 1800 cars were handled by the freight slip. This freight slip was later converted into a passenger slip for the accommodation of transbay ferry boats, and freight cars were brought to the exposition grounds by the Belt Line Railroad through the Fort Mason Tunnel which was completed just prior to the opening day.

32 A 20-ton locomotive crane with a 35-ft. boom and grab bucket was secured at an early date, and, in addition to its use for unloading road material from cars, spotting cars and similar work, was found most useful for unloading heavy boxed trees and the erection of standards and statuary, an extension boom being used in many instances. This crane was also extensively used for unloading heavy exhibits in all buildings, except in the Machinery building, where traveling cranes were available. A similar crane was used exclusively for unloading road rock from barges to a bunker.

33 The work of the electrical department was expedited by the use of a 1-ton storage battery crane truck having a 2-ton trailer. While adapted for general delivery work, the greatest value of this was realized in transporting transformers and oil drums from the warehouse to the numerous building vaults. The transformers were loaded on to the trailer and hauled to a point opposite the vault where they were lifted off the trailer one at a time and landed on dolly trucks. Planks were laid from the curb up the steep terraces and into the vaults, over which the dolly trucks were hauled by a rope running





around a block inside the vault and fastened to the back of the truck. Transformers up to 50 kw. in size were handled in this fashion with surprising ease and speed. After building an extension to the boom, the crane was used for setting gas and incandescent standards.

34 The crane was a great labor saver. It would run into the staff shop where the standards were made, pick up a standard, transport it to the desired location and set it in place on the foundation. The standards were kept in a vertical position during the entire operation and damage to the staff thus avoided. No other type of equipment proved as useful as the battery truck crane for handling loads where single pieces did not exceed one ton in weight, and more trucks of this type could have been used to advantage.

35 As the Exposition had undertaken to deliver all exhibits from the cars to the designated space and as it was known from past experience that most of the exhibits would arrive in the 60-day period prior to the opening, this problem called for careful consideration. In view of the fact that the cars would be run alongside unloading platforms in each building, and that most of the exhibits would arrive in packing cases, transveyors consisting of 4-wheeled hand trucks for use in connection with special detached platforms, were adopted for this service. The freight was loaded on the platforms which were later picked up by the transveyors and transported to the exhibit space.

36 Storage battery industrial trucks were also used for handling exhibits, a type being adopted which had an extension platform suitable for handling bulky goods. These trucks had a capacity of 2 tons each and frequently towed several transveyors in addition to carrying a full load. Since the opening of the Exposition they have been used for distributing supplies throughout the grounds, and for this work are superior to any other form of auto truck, as they are able to get around and through crowds quickly and without annoyance to visitors. Motor cars and trucks have played no small part in the construction of the Exposition, in that they have enabled men and materials to be moved rapidly when and where required. With work scattered over a territory a square mile in area and over two miles in length, many cases arose where slower methods of transportation would have proven costly and delayed the completion of the work on time.

37 The Exposition required for the use of its construction departments 20 Ford automobiles and 14 trucks, of which two were 5-ton, three 2-ton, two 1½-ton, four 1500-lb. and three 800-lb. Just prior to and some time following the opening day this equipment was operated 24 hr. per day, and additional equipment was rented to meet

the unusual situation at that time. While the construction program had been practically completed prior to opening day, there was an enormous amount of work incident to getting exhibits installed and cleaning up the grounds. It is estimated that within the last 26 hr. something like 10,000 tons of debris of various kinds were hauled to the dump and burned.

38 The transportation of visitors within the grounds was solved to the apparent satisfaction of the public by the use of the auto train consisting of a small tractor and several trailers. At previous expositions a considerable amount of walking was associated with all forms of transportation except wheel chairs. At St. Louis auto buses were used but they were not easy of access. The cars of the auto train are low enough to permit passengers to step directly on to the platform and the routes are such that the trains pass within 100 ft. of all buildings; in fact it is quite possible to see the whole Exposition, except the courts, from these auto trains. The seats run the whole length of the cars, which have an aisle in the center for the conductor. Each car accommodates 20 passengers, and the gasolene tractor, which is equipped with a Ford engine, is capable of hauling three cars. The running gear of the cars is so designed that they follow closely the path of the tractor. The braking is accomplished by a shoe to which the drawbars of connecting cars are connected. This shoe comes in contact with the pavement when slack is thrown into the train by the driver applying the brakes on the tractor.

39 A miniature steam railway runs from a point near the southwest corner of the Machinery building, along the water front to the Live Stock section, but owing to its distance from most of the buildings it is not a factor in handling the traffic. Storage battery chairs in sizes to accommodate two and three people are extensively used, but the old wheel chair with an attendant retains its popularity.

FIRE PROTECTION

40 Certainly no exposition had a fire protection system comparable with the one installed at this one and it may safely be said that few communities enjoy an equal security against fire. Located as it is within the city proper and close to the residential district, the Exposition represents a tremendous fire hazard to the city, and the problem of fire protection was carefully studied by the engineers of the Exposition in consultation with the Board of Fire Underwriters. In compliance with the general plan adopted, a high pressure water

system was installed throughout the main portion of the grounds and as far west as the Live Stock section.

41 The system was designed to operate as a part of the city's auxiliary water supply system and to be capable of delivering 15,000 gal. per min. at any hydrant, at a pressure of 200 lb. per sq. in. The distribution system includes 52,000 lin. ft. of pipe in sizes from 6-in. to 16-in. lap welded pipe and Dresser all-steel couplings are used throughout, with extra heavy flanged cast-iron fittings and valves. All hydrants are of the flush type, set in circular concrete manholes with wooden covers. The hydrants have two 3-in. outlets and are spaced approximately 300 ft. apart. With the exception of the Fine Arts and Horticulture buildings, the high pressure system is brought into all exhibit buildings. Four 8-in. pipes are run into each building, one from each side, and serve the inside and roof hydrants, the automatic sprinkler system, the cornice sprinklers and the roof monitors.

42 The 8-in. valves controlling the supply to the buildings are kept closed but the 4-in. by-pass valves are left open. This practice was adopted to limit the damage in case of a break inside the building. These 8-in. control valves are located outside the buildings, and in the same manholes are installed 6-in. cross connections to the low pressure supply, with check and gate valves, so that in the event of a failure in the high pressure supply the sprinklers will be fed from the low pressure system. The roof and inside hydrants have 3-in. outlets and are so located that any portion of the roof or floor can be reached with 150 ft. of hose. The monitors have a capacity of 1500 gal. per min. and are set so as to play on the buildings opposite them as well as on the roofs on which they are located.

43 The cornice sprinklers are arranged to produce a water curtain on the sides of the buildings which face one another and are less than 150 ft. apart. The standard spacing of cornice sprinklers is 8 ft., but they are placed closer at points where offsets or recesses occur in the building walls. Two designs of heads are used, one of which has a deflector to spread the water in the form of a 120-deg. fan, while the other has a circular opening which distributes the water in a plane at right angles to the axis of the head.

44 Automatic sprinklers are installed in all exhibit buildings with the exception of the Palaces of Machinery, Horticulture, Fine Arts, the Festival Hall and the administration portion of the California building. The domes in the eight buildings of the main group and the ceiling of the Palace of Machinery are not equipped with sprinklers, as the height is so great that the effectiveness of sprinklers is

questionable. The sprinkler system was installed in accordance with the rules of the National Board of Fire Underwriters at a cost of approximately $137,000.

45 In the design and construction of the buildings fire protection was borne in mind. Concealed spaces where a fire might flourish unobserved were carefully avoided, and heat curtains were used extensively to prevent circulation of air and insure the confinement of heat and operation of the automatic sprinkler and fire alarm systems. Reinforced concrete firewalls were built where the architectural requirements resulted in buildings being connected by colonnades or other decorative features. The exterior building walls were carried down to the ground surface and sheathed on the inside to a height of 12 ft. above the floor. The roofing used had a top sheet of either crushed brick or asbestos, and wire glass was used in all skylights.

46 Additional protection was afforded by hydrants connected to the domestic water supply system, which system was designed to deliver, at 80-lb. pressure, 3000 gal. per. min. without affecting the normal requirements of domestic service. Single 3-in. hydrants of the flush type were installed in the roadways about 400 ft. apart. Standpipes with 75 ft. of 1½-in. hose on racks were installed along the aisles and sidewalls of exhibit buildings and all participants not located in exposition palaces were required to equip their buildings with standpipes and hose reels with 50 ft. of 1½-in. hose.

47 The Exposition installed one 3-gal. chemical fire extinguisher for each 3500 sq. ft. of floor area in the exhibit palaces, and in addition required exhibitors and other participants to install a similar extinguisher for each 2500 sq. ft. of space occupied by them. At least four 33-gal., 2-wheel chemical extinguishers were installed in each of the exhibit palaces. Fire retardant paints were used, and all fabrics and inflammable material such as bunting, flags and scenic effects were treated with a fire-retardant compound.

48 To provide against a possible break in the high-pressure system or a shortage of water from any cause, suction pipes were installed in the various pools to be used in connection with fire engines. These pools have a storage of over 5,000,000 gal., and in an emergency it would also be possible to station fire engines on the various wharves and pump salt water.

49 The fire department is operated as a part of the city's department, although all the apparatus used was secured by the Exposition on a rental basis. Three fire houses were built on the grounds, one at the center of the zone, one near the southeast corner of the

main group and the other near the northwest corner of the main group. The fire-fighting equipment includes the following pieces, all of which are motor-driven: 3 fire engines with motor-driven pumps, 3 hose wagons, 2 ladder trucks, 1 squad wagon, 1 chemical engine and 1 chief's car. All fire alarm calls are answered by the city's fire department with equipment from nearby stations, but in case of a serious fire two fireboats and a large number of fire engines and auxiliary apparatus could be quickly brought into service.

50 A complete fire alarm system has been installed with boxes not over 400 ft. apart in all buildings and along roadways. These boxes are of the non-interfering type and are installed in combination with the police boxes which latter may be used by firemen if desired, as the central board is in the same office as the fire alarm switchboard and there are telephones in each of the fire houses. Each of the three stations is equipped with the usual gong, punch-register, tapper bell, take-up reel and automatic switch for turning on the building lights when the alarm comes in. A complete central office is located in the Liberal Arts building and is operated as an exhibit. The equipment includes a 12-circuit switchboard, an 8-circuit automatic repeater with 4 engine house circuits, signal wheel transmitter for telephone and automatic alarms; also a punching record, a take-up reel, an automatic time stamp for recording the exact time an alarm is received, and all necessary storage batteries and testing apparatus.

51 All exhibit buildings and many of the detached buildings, especially those having moving picture apparatus, are equipped with an automatic fire alarm system of the pneumatic tube type. Copper tubing not exceeding 1000 ft. in length is run around the ceilings or parts of buildings where heat from a fire would quickly affect the tubing. The tubing is 0.09-in. outside diameter with a hole approximately 0.045 in. in diameter. One end of each tube is connected to a detector consisting of a sensitive metallic bellows such as is used in aneroids. In the event of a fire the pressure of the air in the tubing is increased due to the expansion of the contained air and the bellows expands and closes an electric circuit, which in turn sends a signal to the central fire alarm station and to the annunciators placed at the building entrances to inform the fire department of the location of the fire. The bellows mechanism or detector is located in a steel cabinet which contains 5 detectors, each connected to about 1000 ft. of tubing. The buildings of the main group have at least 5 of these cabinets in each building and some as many as 8, while the buildings of minor importance have a lesser number.

52 The terminals of all tubing are brought to the cabinet to permit the testing of the tubing and detectors, which is done weekly. A leakage valve attached to each detector prevents a rise in pressure due to ordinary changes in temperature. The tubing is run in all ceilings and domes and under floors wherever open spaces exist. Between 95 and 100 miles of tubing and 72 detector cabinets have been installed in the exposition buildings at a cost of $30,600. This charge was based on unit prices of 60 cents per 100 sq. ft. of floor space protected where the tubing was located above the ground floor, and 40 cents per 100 sq. ft. where the tubing was run beneath the floor. The contract price also includes the operation and maintenance of the system during its period of use, the apparatus and tubing to revert to the contractor.

53 The fire protection measures adopted by the Exposition cost in the neighborhood of $900,000, but the fire losses up to August 1st have been insignificant and the security obtained has warranted the Exposition carrying only a nominal insurance on completed buildings.

MECHANICAL FEATURES

54 *Power Plant.* With three large electric companies in the field, each capable of supplying needs of the Exposition, the necessity for building a power plant was not considered. The Pacific Gas and Electric Company, which furnishes electric energy to the Exposition, has a substation on the grounds, and the interesting feature of a power plant in operation is supplied by the Sierra and San Francisco Power Company, whose 18,000-kw. plant is located within the grounds, about 500 ft. east of the Palace of Machinery. This plant is operated in conjunction with a hydroelectric system for supplying energy to the United Railroads and is an excellent example of a standby station with steam turbines and oil-fired boilers. Under an arrangement with the Pacific Gas and Electric Company, this steam plant is kept in operation continuously, so that in the event of an interruption in the regular source of supply it will carry the load of the Exposition. A boiler plant was planned for furnishing steam to exhibitors in the Palace of Machinery, but owing to the lack of demand for steam the installation was never completed.

55 *Heating and Ventilating Equipment.* With the exception of the spaces occupied as offices, no heating was provided for the main exhibit buildings other than the Service building, Administration building, Press building, Festival Hall, Palace of Horticulture and the Auditorium at the Civic Center.

56 The heating of Festival Hall is a departure from standard practice in that a system was installed using gas-fired hot-air heaters with forced circulation. While gas at 75 cents per 1000 cu. ft. is an expensive fuel, its use for the short period of the Exposition was justified by the saving in the initial cost of the plant as compared with a steam plant. The heaters and supply fans are located in two rooms, one on each side of the main auditorium. Each fan room contains two steel plate fans, each having a capacity of 1500 cu. ft. per min., and four hot-air furnaces each capable of burning 440 cu. ft. of gas per hour. The arrangement is such that one fan serves two heaters and delivers air through registers located in the columns around the main entrance, while the other fan discharges into a plenum chamber beneath the raised side seats, openings being provided in the risers for the discharge of air into the auditorium. Air is removed from the auditorium by two multi-blade exhaust fans installed beneath and on each side of the stage, and having a combined capacity of 5200 cu. ft. per min. All fans are belt-driven by direct current motors equipped with armature control.

57 In the California building, the administration quarters are heated by 8700 sq. ft. of direct radiation with vacuum returns. Steam for the heating system and for the hot-water supply and kitchen equipment is supplied by two oil-fired, cast-iron boilers operated at a pressure of 5 lb. Ventilation for the ball room is secured by two 60-in. disc fans located in the attic space.

58 To provide the necessary heat in the dome portion of the Palace of Horticulture, which is essentially a large conservatory, a hot-water system with forced circulation was adopted. The hot water is supplied from a model boiler plant located about 90 ft. south of the main building, in which are installed two oil-fired cast-iron boilers equipped with vertical, rotary burners. Owing to the architectural requirements of the neighborhood, it was undesirable to use a tall stack, and smokeless combustion has been secured with a stack which is only 30 ft. high from the burners and terminates in the staff basket. Forced hot-water circulation is produced by two 4-centrifugal pumps designed to operate against a head of 40 ft., each direct-connected to a 7½-h.p. induction motor. Recording thermometers and a flow meter enable accurate records to be kept of the amount of heat delivered by the plant.

59 The heating required for the section beneath the dome was calculated on the basis of 3,725,000 cu. ft. of space, 54,650 sq. ft. of glass surface, and 15,470 sq. ft. of wall surface. It was assumed that

there would be one complete change of air every three hours and a loss of 6 B.t.u. per hr. per sq. ft. of wall surface and 17 B.t.u. per hr. per sq. ft. of glass surface, making a total of 1,376,800 B.t.u. required per hour to maintain a temperature of 50 deg., with an outside temperature of 35 deg. The four rooms adjoining the dome section were figured for temperatures from 60 to 80 deg. fahr., which brought the total requirements of the building to 3,582,000 B.t.u. per hr., requiring the circulation of 190,000 lb. of water per hour, with an initial temperature of 250 deg. and a loss of 20 deg. Radiators of the cast-iron type were used instead of the customary pipe coils, with satisfactory results. Thermostatic control was installed in several of the rooms for regulating the temperature of ponds. The plan of using forced circulation and high temperatures made possible a material reduction in the size of the mains and the quantity of radiation required.

60 Minor heating installations were provided in various buildings, such as the Service building, which is heated by direct steam radiation of the single-pipe system, while the Press building is heated by the Rector system, which burns gas in connection with cast-iron radiator elements, the products of combustion being removed by means of a fan located in the basement. The offices in the exposition palaces and most of the buildings erected by participants are heated by gas radiators or gas stoves of the radiant type, flues being provided to carry away the products of combustion.

61 Owing to the fact that the Auditorium in the Civic Center is a permanent, four-story, fireproof building, a modern heating and ventilating system was installed. The main hall in this building is octagonal in shape, 187 ft. by 200 ft. in plan, with an area of 34,772 sq. ft., and occupies the center of the building to its full height. This space is flanked by two banquet rooms 56 ft. by 136 ft. on the ground floor, and there are numerous convention halls and offices on other floors, as elsewhere noted.

62 In order that the main hall might be used for dances and other functions, a system of heating and ventilating was adopted which left the floor free of obstructions. Fresh air is forced through the openings in the balcony risers and grilles along the face of the balcony. The grilles are supplied from separate ducts, whereas the openings in the risers receive their supply from a plenum space beneath the balcony seats. Exhaust fans are provided to remove the air through large grille plates beneath the balcony, along the side

walls and through smaller grille plates above the balcony, and just below the base of the auditorium dome.

63 For supplying fresh air, two fans are installed in the basement, each having a capacity of 145,000 cu. ft. per min. Two main exhaust fans, each capable of handling 70,000 cu. ft. per min., are installed in the attic, and two exhaust fans of 75,000 cu. ft. per min. capacity in the basement. Additional supply and exhaust fans are provided for the banquet rooms, toilets and kitchens, these fans having a total rating of 39,000 cu. ft. per min. The fans, of which there are 18, and the vacuum pumps, are belt-driven by motors aggregating some 250 h.p. Heating coils are used in connection with the various fresh air supply fans, and provision has been left for air washers should they ever be found necessary. Direct radiation is used for the upper corridors, convention halls and offices. It was the intention to obtain steam for this building from a Civic Center plant to be built by the city, but as this plant was not completed at the time the Exposition desired to occupy the Auditorium, a temporary steam plant was erected nearby.

64 *Pumping Plants.* In addition to the pumping plant described under the heading of Water Supply, numerous pumping equipments were installed throughout the grounds in connection with the fountains, the pools, the drainage beneath the buildings, the handling of sewage and the supply of salt water. Belt-driven centrifugal pumps were adopted as a standard, as they permitted the use of rented motors, were lower in first cost, and carry a higher salvage than direct-connected pumps. Another reason for the adoption of the belt drive was the ease with which the pump speeds could be changed to meet conditions of operation different from those planned. It was found desirable, in a number of instances, to make such changes, particularly in connection with the fountains and cascades; for while the quantity of water and the pressure required may be calculated in advance to a nicety, the artistic features produced by the playing water may prove a disappointment.

65 The pumping equipment of a fountain is a simple problem compared with that of laying out the distribution piping and deciding on the form, size and adjustment of the nozzles. It was found that the use of lead pipe for connection to nozzles expedited the work of adjustment and in many cases the nozzles themselves were formed from lead pipe. Experience indicates that valves or cocks should not be placed close to nozzles as they disturb the flow of water and prevent the realization of a smooth and solid stream. Where groups or decora-

tions are liable to be constantly wet, they should be made of cement, as staff will fail, even though the water may not play directly on it. If canvas is used for lining basins or pools it should be painted on both sides and thoroughly protected against damage from workmen's shoes. Sixteen motors, aggregating 755 h.p., are used for driving pumps to produce water effects.

66 *Transfer Table.* The arrangement of the Palace of Transportation necessitated bringing all rolling stock into the building on one track and distributing this equipment to 14 exhibit tracks by means of a transfer table. This transfer table had to be designed to carry the largest locomotives and cars of which, up to that time, there was any record. A standard steel transfer table would have cost from fifteen to twenty thousand dollars and its salvage value would have been problematical. The transfer table as built cost $7000, exclusive of foundation.

67 A plan was adopted whereby the tracks for carrying the transfer table were placed 5 ft. below the building floor level, but it was not necessary to provide a pit as the floor was about 10 ft. above the ground. Placing the tracks at this level was a further advantage as it lessened the amount of foundation piling required and simplified the construction generally.

68 The transfer table deck is 76 ft. long and it is made up of timber stringers on which are laid the ties and track. This deck is supported on 14 trucks, two trucks to a track. Each pair of trucks is tied together with stringers which carry the deck and transmit the load to the trucks through special steel bolsters. No springs are used and, barring the flexibility due to the timber work, the table is a rigid structure.

69 The transfer table has a 300 ft. length of run and is moved back and forth by two hauling lines and two tail lines. One hauling line and one tail line are attached to the deck near each end, and this arrangement keeps the table in alignment at all times. Each hauling line, with its companion tail line, is rove on one drum in such a manner that as one line runs off the drum the other fills the space vacated. This method successfully prevents the cables from overriding and admits the use of a smooth face drum. The two drums are geared to a countershaft which is belt-driven by a 35-h.p. variable-speed induction motor. A post type brake on the countershaft, although seldom used in operation, admits of checking the load. The cable-operating equipment was designed to exert a pull on the transfer table of 40 lb. per ton of maximum load, which has proven amply

Fig. 23 Transfer Table, Palace of Transportation

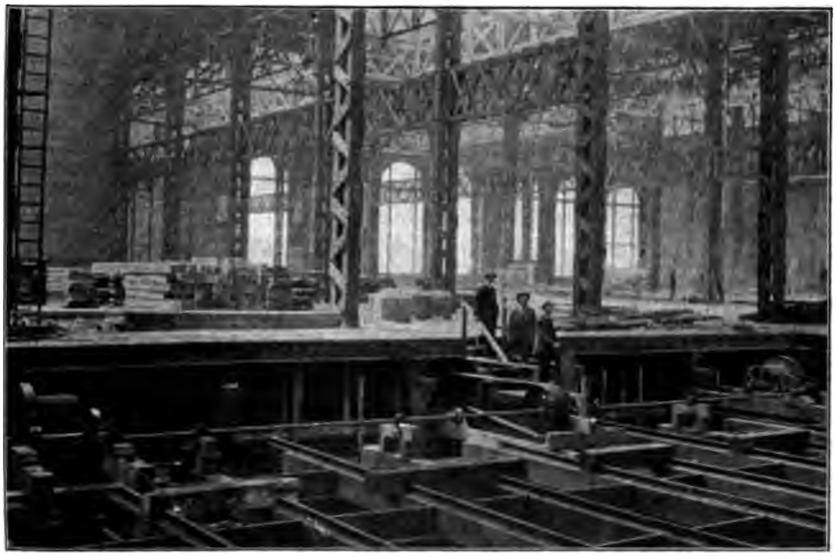


Fig. 24 Hauling Gear for Transfer Table

safe as the table has handled without effort one of the largest locomotives of the Mallet type. A motor-driven car puller was mounted on the table for hauling cars on and off, but it handled only the lighter rolling stock, the heavier work being done by a switching locomotive or a locomotive crane.

70 *Traveling Cranes.* To facilitate the installation of heavy exhibits three bays in the Palace of Machinery were equipped with traveling cranes. The center bay has two 30-ton cranes with 5-ton auxiliary hoists, and each adjacent bay has a crane of 20 tons capacity. These cranes have a span of 67 ft. 8 in. and are of standard steel construction, with 3-phase, variable-speed motors. Some trouble was experienced, owing to variation in the width of the span due to the shrinking and warping of the timber structure of the building. This difficulty was corrected partly by adjusting the rails, but largely by the use of special wheels with extra wide treads. The cranes handled some 4500 tons of machinery from the cars to the exhibit spaces, and in addition were used by exhibitors in the erection of their machinery. These cranes were all obtained on a rental basis, the sum of $13,600 being paid for their use, which amount also includes the erection and dismantling of the cranes.

71 *Refrigeration Plants.* Refrigeration service was considered desirable in the Palaces of Horticulture and Food Products, and refrigeration plants were accordingly installed in these two buildings. The apparatus was installed by exhibitors, but was operated at the expense of the Exposition, which in turn sold refrigeration service to various exhibitors and participants. Subsequent events, however, proved that the use of small individual plants, or ice, would have been more satisfactory than a central plant and distribution system, because the needs of the consumers were nominal.

72 In the Palace of Horticulture the central plant is an exhibit of the Automatic Refrigerating Company. The principal feature of this installation is its automatic control, a thermostatic switch starting and stopping the motor according to the changes in temperature of the refrigerated space. The water for the condenser is automatically regulated to meet load requirements and is shut off entirely when the motor stops. Should the water supply fail the motor will stop automatically and will start again when the water supply is resumed. The plant has a rated capacity of 3½ tons of refrigeration per day. The piping is 1000 ft. long, 1¼-in. diameter, and is covered with cork insulation. Although this plant receives but little attention the results are most satisfactory.

73 The central plant in the Palace of Food Products was installed as a working exhibit by the York-California Construction Company and consists of two double-cylinder, double-acting, vertical compressors —one of 20 tons and the other of 4 tons refrigerating capacity, both belt driven by motors. Brine is chilled and circulated through the distribution piping to the various exhibits. This piping is not covered as it is cheaper to waste power during the short period of the Exposition than to provide insulation. To prevent waste of water an inexpensive cooling tower was built 4 ft. by 8 ft. in section and 25 ft. high. Spray heads were installed half way from the top of the tower and the air for cooling purposes taken from beneath the building floor and blown through the tower by a 30-in. disc fan. The discharge of air from the tower directly into the building has not proven objectionable.

74 *Repair Shops.* Early in the progress of construction work machine and blacksmith shops were fitted up to care for general repair work and later for the manufacture of electric standards and fixtures. All of the iron work used in the construction of the gas and electric standards was fabricated in the exposition's shops. It was found that the work could be gotten out more quickly and at less cost in these shops than by sending to outside shops. At times as many as 12 house-smiths and 8 machinists, exclusive of helpers, were employed on this class of work alone.

75 In conjunction with the operation of the machine shop, and in a separate building, is an automobile repair shop, in which as many as 7 machinists have been kept busy on the maintenance of the motor transportation equipment. While there can be no question but that the use of motor-driven vehicles was a large factor in completing the Exposition and opening it on time, the maintenance of these vehicles has been a large item of expense.

ELECTRICAL FEATURES

76 One of the earliest problems for settlement was that of securing a supply of electrical energy for the construction and operation of the Exposition. Owing to the fact that there were three electrical companies, with steam stations within the city and connections to extensive hydroelectric systems, the Exposition was in a favorable position to negotiate for the purchase of electric energy.

77 In order to have a basis on which to solicit bids, a preliminary estimate was made of the probable load and the amount of energy required during the Exposition and during the periods previous to and

subsequent to the Exposition. At the time this estimate was made, although the general plan of the Exposition had been fixed, no details had been decided upon, nor had any scheme of illumination been considered. To arrive at a figure sufficiently close for the purpose of making a contract for electric service, the areas of the buildings, courts, gardens and avenues were computed and various light intensities assumed for each. Tungsten units were figured for use in the buildings and courts, and magnetite arc lamps for the gardens and avenues. The load in the concessions area was assumed as being 80 per cent of the connected load in the concessions area of the St. Louis exposition. The amount of load for exhibitors was guessed at as there was no information available on this point.

78 These rough calculations indicated that the average peak would be 8500 kw., with a maximum of 11,500 kw. during the winter season when the day load would overlap the evening load. It was decided to compare proposals on the basis of a 12,000-kw. maximum load and a load factor of 30 per cent. In asking for bids on electric service, the various companies were requested to indicate what distribution apparatus and cable they were willing to loan to the Exposition. It was realized that the purchase of a distribution equipment by the Exposition would entail not only a large initial investment, the funds for which might more advantageously be spent in buildings and decorative features, but also a large loss in having to dispose of the apparatus quickly afterwards. It was thought that the company which supplied service would be in a position to absorb the equipment into its system after the close of the Exposition with no loss other than that incident to anticipating its usual annual purchases and the subsequent storage until the equipment was required.

79 On Feb. 28, 1913, a contract for supplying electric service was entered into with the Pacific Gas and Electric Company, under the terms of which the Exposition was to pay 2 cents per kw-hr. during the pre- and post-exposition periods, and during the actual period of the Exposition a fixed charge of $18,000 per month, plus an energy charge of 0.6 cents per kw-hr. For the fixed charge named the Exposition was entitled to take loads up to 15,000 kw., but was to pay $2.00 per month per kw. for the excess demand over and above this amount. At the time this agreement was signed, there was every reason for assuming that the maximum load would be 12,000 kw. or more, but owing to the period of financial depression which followed, and the European war as a climax, a large number of prospective foreign and domestic exhibits were withdrawn, and the maximum load

realized has not exceeded 8100 kw., with an average maximum of 7880 kw. On the basis of this average maximum and the charge for energy, the cost per kw-hr. has averaged 1.7 cents.

80 Following out the idea of renting the electrical apparatus from the company supplying electric service, an agreement was entered into with the Pacific Gas and Electric Company, whereby it was to furnish the Exposition, for use during the Exposition and during the construction and dismantling periods, all of the electrical apparatus, instruments, materials and appliances which might be required, except building wiring, searchlight projectors and other special apparatus not in general use by power companies. For the use of this apparatus the Exposition agreed to pay 5 per cent of the cost of the apparatus to the Pacific Gas and Electric Company as a rental, independent of the length of time the apparatus had been in use, and to pay outright for all apparatus not returnable.

81 The following are a few of the most important items secured under the terms of this contract: Forty miles of lead covered primary, 101 miles of weatherproof and rubber covered wire, 1076 ornamental magnetite arc lamps, 2139 meters, 76 motors aggregating 1603 h.p., two 1000-kw. motor-generator sets, two 250-kw. motor-generator sets, 139 oil switches, 517 transformers having a total capacity of 15,577 kw., and 418 poles. The total value of this apparatus and material is $450,000, and it is safe to say that had the Exposition been obliged to invest this amount a number of decorative features which have contributed to the beauty of the Exposition and possibly a building or two, for instance the Tower of Jewels, would of necessity have been omitted.

82 Included in the contract for electric service was the requirement that the power company should install a sub-station on the grounds and a 5000-kw. steam standby plant. This requirement was later modified by an agreement between the Pacific Gas and Electric Company and the Sierra and San Francisco Power Company under the terms of which the Pacific Gas and Electric Company installed its sub-station in the west end of the Sierra and San Francisco Power Company's power house, and received standby service from its 18,000-kw. turbine plant. As the power house mentioned is within the grounds and the plant was to be kept running during the period of the Exposition, it was felt that the protection of service thus secured was even greater than under the terms of the original contract.

83 In fulfilment of the terms of the contract, the 60-cycle, 3-phase energy used by the Exposition is generated at Station A,

which is in the southern part of the city. This station has a capacity of 52,000 kw. in steam prime movers and, in case of breakdown or overload, takes electric energy from the power company's 60,000-volt transmission lines which terminate at Martin Station, at the southern boundary of the city, where the voltage is reduced to 11,000 volts. To transmit this energy three 4/0, 3-conductor, 11,000-volt cables run underground from Station A direct to the sub-station on the grounds, known as Station F, and a tie-line connects this sub-station to the power company's system of city sub-stations. Six 1500-kw. transformers are installed in Station F which reduce the voltage from 11,000 to approximately 4100, at which voltage the Exposition takes all energy except the 11,000-volt service for its motor-generator sets. The magnetite arcs used by the Exposition are served by six 75-light rectifier sets in Station F and 11 similar units in an adjacent building. The design of Station F is a striking example of the realization, in a temporary installation, of all of the protection obtained by standard equipment, without that expense and complication so frequently identified with installations of a permanent character.

84 *Primary Distribution System.* Power from Station F is distributed throughout the grounds by means of fifteen 3-conductor, 3-phase, 4100-volt feeders and two 11,000-volt, 3-conductor, 3-phase feeders. With the exception of the overhead lines in the concessions district, and the extreme westerly end of the grounds, the distribution is from an underground system. The primary cable system includes 20 miles of cable and forms a ring around the main group of buildings, with a tie-line across certain courts, and loops into the States and Foreign sites. The ring, ties and loops are all operated open, the feeders being independent of one another. The various cables terminate at oil switches in transformer vaults, and in case of cable failure the load affected can be quickly transferred to another cable, or divided among several other cables through the tie connections.

85 The cable system and somewhat extensive arrangements for switching were adopted more to provide flexibility of operation than for protection against cable failure. It was realized that the requirements of participants for electric service could not be ascertained in advance, so estimates were made of the probable load in each building, and a cable layout based thereon which would handle the known load due to the Exposition's lighting and take care of the loads assumed. It so happened that the actual loads were not far from those assumed and no changes in the scheme of feeder loading were necessary, but

had a few isolated loads of any magnitude materialized, the flexibility in switching features would have taken care of the situation.

86 *Secondary Distribution.* With the exception of the areas mentioned, where overhead lines are installed, alternating current at secondary voltages for light and power purposes is taken from transformer vaults which are built into the various buildings throughout the grounds. These vaults are of concrete and each has a ventilating flue extending to the building roof, and a screen door opening to the outside of the building. All vaults are equipped with sliding, fireproof doors held open by fusible links. These vaults are in reality minor sub-stations in which the primary cables terminate and the high-tension switch gear and transformers are installed. There are no secondary fuses or switches in the vaults, the distribution boards being usually either on the top or on one of the sides of the vaults. All buildings are furnished with energy for light from 3-phase, 4-wire mains connected to the distribution boards on the transformer vaults. This arrangement provides 115 volts from any conductor to neutral, and consumers are served from one, two or three of the phases, depending upon the size of their load. Power is. supplied from separate transformers and the distribution is at 230 volts from 3-phase, 3-wire mains.

87 *Building Wiring.* A general system of lighting mains was installed beneath the floors of the eight buildings in the main group. Power mains were installed only as required. The wiring for the general and decorative lighting in the main buildings consisted of 3-phase, 4-wire risers from the lighting mains beneath the floors to the roof trusses, with 2-wire branches to the outlets. The risers, for a distance of 15 ft., were placed in conduit for mechanical protection against damage during construction of exhibitors' booths. The building wiring for the eight buildings, exclusive of the mains, without lamps and reflectors, cost about $4.00 per outlet, and the special reflectors $3.50 each.

88 For supplying the searchlight projectors on the roofs of the various buildings 3-wire mains were run beneath the floors, with four risers to a building. The size of wire used for the mains varied from No. 6 to 500,000 c.m. In the Palace of Machinery the lighting mains were located in the bracing trusses, between columns, and connected to the transformer vaults by feeders run in ducts which had been laid during the construction of the building. The copper in the crane trolley wires was used to advantage for the A-C power mains, taps

being taken off at intervals which fed auxiliary buses to which the consumers were attached.

89 For the assistance of exhibitors printed specifications were issued covering the wiring for booths. These contained clauses which would apply to any size lighting or motor load, together with general clauses covering the requirements of the Exposition, and protected the exhibitor. These forms were distributed to all interested parties and their use avoided the controversies which so often involve the architect, owner and contractor.

90 Consumers in the main exhibit buildings were required to run their service wires in conduit from their meter boxes to the nearest main and there install a metal-lined fuse box with fuses of 25 per cent greater capacity than the main fuses on their switchboard. Connection to the consumer was made within the metal box and the box sealed. The service fuses are not accessible to the consumer and were installed to prevent overloading and the use of coppered fuses on the consumer's board. Under normal conditions a fault or overload affects the fuses on the consumer's board only, so that when the service fuses fail it is an indication to the Exposition that something is wrong with the consumer's installation. The consumer's service terminates in a metal, or metal-lined, meter box provided with a glazed door or opening to permit the meter being read without breaking the seal on the box. This arrangement prevents tampering by the consumers or unauthorized employees of the Exposition and has proven quite effective.

91 *Electric Rates.* Electric service is furnished at Flat, Meter, and Special rates, as follows:

Flat Rate: Electric installations having a connected load to any given service of less than 2 kw. are charged at the rate of $15 per month per kw. of connected load, with a minimum of $2.50 per month where the load is 166 watts or less. This rate is based on the understanding that the entire connected load will not be used more than 240 hours per month.

Meter Rate: Consumers having connected loads in excess of 2 kw. are provided with meters, and electric service is charged for on the basis of $2.50 per month per kw. of connected load, and an energy charge of 5 cents per kw-hr., with a reduction of 0.1 cent for each 1000 kw-hr. used per month up to 20,000 kw-hr. and 3 cents per kw-hr. for the balance.

Special Rate: All exhibitors in the main palaces were given lighting service at rates 25 per cent less than the foregoing, and power service at a rate of 3 cents per kw-hr. The connected load was estimated from the manufacturer's rating of the apparatus and in most instances was the maximum demand.

92 An exception to these rates was made in order to encourage participants in the States and Foreign sites to illuminate their buildings and no fixed charge was made on lights installed for exterior decorative purposes.

93 *Direct Current System.* Direct current for the operation of searchlights and projectors is generated by two 1000-kw., 250-volt generator sets and two 250-kw., 125-250-volt balancer sets. These units are installed at the various load centers in order to reduce to a minimum the distribution copper required, but are all operated in parallel. These machines are operated only at night, with the exception of one of the balancer sets, which provides day service in conjunction with some of the exhibited prime movers in the Palace of Machinery. On account of the large amount of copper required for the direct current distribution system and the large areas involved, direct current service for exhibitors is available only in the Machinery, Mines, Transportation and Manufactures buildings.

94 *Underground Conduit System.* The underground conduit system is approximately 30,000 ft. in length, and comprises 300,000 ft. of duct. The short life of the Exposition necessitated an economical design, and the constant settlement of the filled ground required that the design have an element of elasticity. Following a number of studies and experiments, a type of conduit construction was adopted based on the use of fibre duct with 1/8-in. walls (Linaduct), and wooden manholes. The ducts were laid on a box or trough extending between manholes which varied in size to suit the number of ducts. The space between the ducts and the box was filled with sand sluiced into position with water, except at the entrance to the manholes, where concrete was used for a distance of 4 ft. to prevent the ducts being displaced by cable pulling operations.

95 The boxes were made up in the mill and delivered in 14-ft. and 20-ft. lengths, which were made continuous by splice pieces nailed to the sides and bottoms. The load on the bottom boards was transferred to the sides by means of 1-in. by 4-in. strips fastened to the bottom and sides and spaced 4 ft. apart, the corners held by iron straps. After the boxes were filled flush with sand, pieces were nailed across the top. This form of construction proved to have great mechanical strength, and there were many instances where building materials were piled upon exposed lines, and spans of 10 ft. or more

were left unsupported as a result of undermining for other utility mains, without damage to the conduit. The conduit system was laid before the streets were paved but was not damaged by the teaming or road rolling. In fact, a damaged or misplaced duct has never been discovered.

96 The cost of the conduit system, exclusive of manholes,

FIG. 25 BANNER TYPE LIGHTING STANDARD

averaged 10 cents per duct foot. Most of the runs were in a sandy soil which was easy to excavate and backfill. The manholes were of timber and a considerable saving was made by the use of entrance boxes for the conduit, so designed as to permit starting the cable bends outside of the manhole proper. By this arrangement the manholes were made much smaller than usual. In certain sections where the ground water level was near the surface a shallow type of manhole was used, the cover being the full size of the manhole. Manholes of the shallow type cost, installed, an average of $25 and those of the deep section type $50. Transformers were not placed in any of the manholes but were installed in fireproof vaults as hereinbefore noted.

97 The ducts used for running between arc light standards and for services in the underground system were made up of 2-in. by 4-in. pine pieces with a half round groove, held together by special staples. One end of each piece of duct was turned and the other counterbored to provide a suitable connection. This class of duct cost about 15 cents per foot laid.

98 *Lighting Standards.* The design of the foundations for arc standards was given careful study as most of the standards were to be set in a soft dredger fill. The standards fitted with banners, in particular, were liable to blow over in high winds, as the exposed area was large and the arrangement ideal for catching the wind. These poles are 40 ft. high, 13½ in. in diameter at the ground line, 4 in. in diameter at the top, and carry from 5 to 7 arc lamps which weigh 100 lb. each. The banners have an area of 50 sq. ft. with the center of pressure from 25 ft. to 35 ft. above the ground line. The other design of arc standard used has a staff decoration on the top which, owing to its size, shape and weight, presented almost as severe a problem as the banner type.

99 Considering that there are 200 of these standards, the foundations for which involved considerable expenditure, experiments were carried out with various forms of foundations to determine the least expensive type of construction which would meet the rather unusual conditions. The result of these experiments was the adoption of a reinforced concrete slab placed 6 in. below the ground surface and supported by piling. The piling consisted of 2-in. by 6-in. pieces which were jetted into place in advance of the concrete work. Spikes were driven near the tops of the piling, which were embedded into the concrete, and served to transmit to the piling any uplifting forces due to the turning moment developed by the wind. The concrete slab was made in the form of a cross to insure a sufficient spread with the minimum of material, and a hole was left in the center to receive the pole, which was grouted in after setting.

100 These foundations cost $35 each, and although storms have been experienced which tore away the banners in some instances, none of the foundations has failed. With some 300 gas and electric standards to install, it was desirable to have a form of foundation which would eliminate the delay due to the usual plan of setting a standard in a hole, then plumbing it and backfilling. These standards were

15 ft. long, and the staff was applied to a 4-in. pipe which was continued the full length of the standard. The bottom end of this pipe was fitted with a rough cast-iron sleeve, which was held rigid by lead poured into the annular space between the pipe and the sleeve. The lower end of the sleeve was fitted with 8 set screws. The foundation was made of a block of concrete into which was set a short piece of 4-in. pipe.

101 The standards were brought from the staff shop by a storage battery crane and landed on the pipe stubs set in the foundations. They were then quickly plumbed by means of the set screws. It was

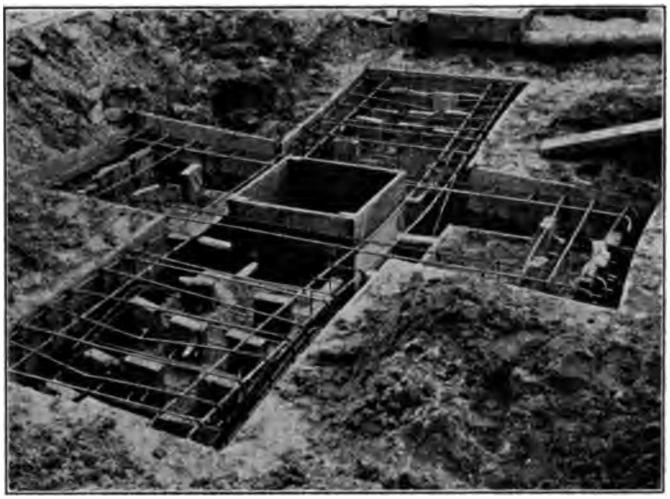


FIG. 26 FOOTING FOR LAMP STANDARD, SHOWING REINFORCEMENT AND ANCHORAGE

not an uncommon performance to set as many as 20 standards in 8 hours, and it is believed that the plan used not only expedited the work but was actually cheaper than the usual method.

102 *Illumination.* At the outset, it was realized that a departure should be made in the exposition lighting and at the same time it was decided that the lighting should not be an independent feature but rather a means to be used for enhancing the beauty of the architectural and decorative features. With the realization that illumination had become a branch of engineering, a specialist in that field was employed to design the general scheme for lighting the Exposition, which was later adopted and made the basis for detailed plans and estimates.

As the actual work progressed, changes were made to bring the scheme into harmony with the architectural and decorative features and provide additional light for the avenues, practical demonstrations having shown that such changes were essential to the success of the effect.

103 The perfection of the illumination produced is largely the result of an appreciation by the engineers of the aesthetic qualities of the work of the architects and artists, and it was therefore possible to secure their coöperation, which was of the greatest value, particularly in the settlement of problems relating to color, and the location, scale, mass and decoration of the lighting standards.

104 The general scheme of illumination adopted was to use flood lighting from hidden projectors and massed lighting from arc standards, exposed sources being used in gardens and along roadways where the treatment admitted of incandescent standards. As the illumination was described in detail by the writer in the Feb. 13, 1915 issue of the *Electrical World,* only the means employed and the apparatus will be touched upon here.

105 The flood of light on the various towers is produced by searchlight projectors hidden on the roofs of buildings and outlying structures. For the flood lighting of the 6 towers, the Palace of Fine Arts and the large groups of statuary, thirteen 30-in. and two hundred 18-in. searchlight projectors are used. Most of these are fitted with dispersion doors which spread the beams anywhere from 10 to 40 deg., depending upon the design of the strip lenses in the door. In some instances double doors are used, one of which spreads the beam horizontally and the other vertically.

106 The searchlight projectors are of the manual control type and most of them are equipped with parabolic mirrors, while the balance have Mangin mirrors. The latter type of mirror is not suitable for lighting building surfaces as it produces a spectrum at the circumference of the field illuminated, but for lighting groups which occupy only a portion of the field this type of mirror is satisfactory. Gelatine color screens are employed in connection with the searchlight projectors and by their use the color tone of the Exposition can be readily changed.

107 The strong light from the projectors would cause deep and sharp shadows, and to neutralize this effect the shadows are illuminated by concealed incandescent lamps which are dipped an orange color. On the towers these lamps are located behind the columns, at the floor levels. To have wiped out the shadows entirely would have resulted in a flat effect, and no feature has contributed more to the

charm of the lighting than the treatment of the shadows with color.

108 A decorative feature is the use of imitation jewels in connection with the illumination of the upper portion of the main tower, where over 100,000 are used, while 25,000 are used elsewhere. The jewels are made of glass and vary in size from 21 mm. to 47 mm. They are cut with facets similar to those of a diamond, and a small mirror is mounted behind each to increase the number of flashes obtainable. These jewels were made in Austria and their cost ranged from 33 cents for the 47-mm. size to 6 cents for the 21-mm. size. Although the jewels are of various colors, most of them are amber. They are mounted in holders which are attached to supporting hooks, allowing them freedom to swing in the air currents. Under the powerful rays of the searchlights the jewels sparkle and glitter and contribute an element of action and color which is most striking. The jewels have never been cleaned and the difference in appearance after 5 months' use is not noticeable.

109 The flood lighting feature for the building wall surfaces is provided by magnetite arc lamps set on ornamental standards 30 ft. high and spaced from 60 ft. to 75 ft. apart. Two types of standards are used, one having a large curved shield of staff with three openings fitted with orange colored translucent fabric; the other has a combination of three canvas banners painted with heraldic emblems. The design of both types is such that the lamps are not in the line of sight but allow a small amount of colored light to pass. Lamps were used on these standards in combinations of 2, 3, 5 and 7, depending upon the distance of the standards from the building walls. Some light was reflected back from the building walls but not enough to illuminate the adjacent avenues, and additional lighting was provided by means of single-light electroliers made of staff and equipped with 250-watt Mazda lamps and 18-in. glass globes.

110 Similar electroliers, carrying from 1 to 21 globes, lighted the South Gardens, Court of Palms and the entrances to some of the other courts. Special globes were used which, while they contributed a mellow diffused light at night, produced a discordant effect during the day owing to their whiteness as compared with the color tone of the travertine finish of the buildings. A lacquer, into which color was mixed, was used to overcome the objectionable appearance of the globes.

111 The globes were all in position at the time it was decided to spray them, and for this purpose a portable rig was made consisting of an air compressor and four potter's wheels, which was moved

around from standard to standard, the globes placed on the wheels and sprayed with the lacquer by means of an air brush. The lacquer coating caused some loss of light but not enough to make a noticeable difference in the illumination.

112 The Court of the Universe, an area of 24,700 sq. ft., is lighted from the columns on the two fountains. Each column is 30 ft. high, the outside being made of glass strips curved to represent the flutes of the column. In each column are ninety-six 1500-watt gas-filled lamps, the light from which is diffused by means of sand-blasted glass screens placed between the lamps and the outer glass. A motor-driven fan is located in the base of each column and blows air through the column to prevent the lamps from overheating. Provision is made so that in the event the air fails the lights will be turned off automatically.

113 An element of life is given to the night picture of the Exposition by the buildings appearing to be lighted up inside, which effect is produced by Mazda lamps fitted with tin reflectors and suspended back of the various building openings. The color effects in some locations are obtained by using dipped lamps and in others by painting the glazing.

114 The kaleidoscopic colors and shapes which appear in motion on the glass dome of the Horticulture building are produced by twelve 30-in. searchlight projectors placed beneath the center of the dome and arranged to project their beams through revolving lenses and color screens. Eight color screens and twenty lenses are used in connection with an opaque sector disc. These three elements are mounted on a vertical shaft and slowly revolved at slightly different speeds. Various combinations of shapes and colors are thus projected upon the dome, but the effects are rather thin, owing to the fact that the quality of glass used does not intercept a sufficient amount of light.

115 Without doubt the most spectacular feature of the illumination of the Exposition is the so-called scintillator, located on the Yacht Harbor. The equipment consists of forty-eight 36-in. searchlight projectors of the manual control type and is used in connection with pyrotechnic displays and steam effects, in addition to being used nightly in the form of a great fan as a background to the illumination. The color screens used in connection with these lights are wooden frames with gelatine sheets held between chicken netting. The gelatine sheets are waterproofed with spar varnish for protection against moisture. The searchlights are operated by 52 U. S. marines under a commissioned officer.

116 The illumination of the exposition buildings and grounds, not including the States and Foreign section, produces a load of 5216 kw., of which 1700 kw. are used for the searchlights, 450 kw. for the arc lights and 3066 kw. for the incandescent lighting. The cost for electric service chargeable to illumination is $630 per night, and for labor, repairs and maintenance $125 per night. The lighting of the emergency gas lights costs about $35 per night, and the gas lighting in the States and Foreign sites about $40 per night. In round numbers the cost to the Exposition for lighting is $830 per night, which sum includes all operating costs but no capital charges.

GAS DISTRIBUTION

117 Gas for the use of the Exposition and its participants is purchased wholesale from the Pacific Gas and Electric Company and is retailed by the Exposition through a system of high pressure distribution mains which were installed by the company for this purpose. The gas used is manufactured from oil at the company's Potrero plant and is transmitted across the city at a pressure of 80 lb. Two Thomas electric gas meters measure the high pressure gas supplied to the Exposition. The distribution mains extend throughout the grounds, an 8-in. ring surrounding the main group, with an 8-in. loop into the States and Foreign sites, and a 4-in. loop into the Zone. Other areas are supplied from 4-in., 3-in. and 2-in. mains laid to form loops or rings. Lap welded pipe was used for the 8-in. and 4-in. mains and standard pipe for the smaller sizes. In all there were 100,000 ft. of pipe laid. All joints and connections to the mains were welded by the oxy-acetylene process and the connections were tested with high pressure gas before backfilling.

118 The various palaces were equipped with low pressure 4-in. and 6-in. ring mains which were fed at one or more points by 3-in. service connections from the high pressure mains. Pressure governors within the buildings reduced the gas pressure to 6-in. water gauge, and oil seals were installed close to the governors for protection against an increase in pressure in the domestic distribution system due to a failure of the governor. Consumers located within the Exposition buildings are supplied from the ring mains, while those in the States and Foreign sites are connected to the high pressure mains, with a regulator set on their premises to reduce the pressure. In the States and Foreign sites, and on top of all kiosks, high pressure gas lamps are installed. Each lamp has a small regulator which reduces the pressure to 3 lb., but oil seals are not used in connection with these

installations. Some 257 of these lamps are in use, with excellent results. Low pressure gas arcs are installed on poles along the Zone for emergency lighting. The distribution of gas lamps throughout the grounds is such that in the event of a failure of the electric service people could get about without inconvenience.

119 With a view to public safety gas lamps were installed for all emergency and exit lighting. Gas is sold to consumers at the rate of $1 per 1000 for the first 50,000 cu. ft. used in any one month, and 80 cents per 1000 cu. ft. for a consumption in excess of this amount up to 300,000 cu. ft., at which point a sliding scale is applied which reduces the price to about 70 cents per 1000 cu. ft. for some of the large consumers.

120 Gas is the only fuel allowed on the grounds, with the exception of plants which are sufficiently large to warrant the employment of an engineer. In such installations the use of oil-fuel is permitted.

121 The average quantity of gas used on the grounds is from 12,000,000 to 15,000,000 cu. ft. per month.

WATER SUPPLY

122 Preliminary estimates indicated that the average consumption of water would be about 1,258,000 gal. per day with a possible maximum of 2,525,000 gal. per day. In the absence of any exact data for guidance, and in order to be on the safe side, the pipe sizes of the distribution system were figured on the assumption that the average consumption would be 2,000,000 gal. per day and the maximum 3,500,000 gal. per day. The maximum rate of flow was assumed to be 1½ times the maximum daily consumption.

123 *Sources of Supply.* The domestic water distribution system was planned and installed in the belief that the water needed would be obtained from the local water company, and negotiations were under way whereby the Exposition was to share the expense of laying the additional mains which would be required to bring the water into the city. No doubt this proposed arrangement would have been put into effect had not the Board of Supervisors of San Francisco started an agitation having in view the acquisition of the water company's plant by condemnation proceedings. Under these circumstances the water company did not care to make investment for the additional mains, and the Exposition found itself confronted with the serious problem of procuring a supply of water from other sources.

124 A survey of the situation indicated the possibility of securing

about 800,000 gal. per day from the water supply system of the Presidio and the balance by developing the underground storage waters in and near Golden Gate Park, with a small additional amount of water from wells on the grounds.

125 *Lobos Creek Supply.* The Presidio takes its supply from Lobos Creek, which forms the southern boundary of the Presidio and drains a portion of the city. The ownership of the water in this stream was a matter of dispute, but the Exposition was able to secure the permission of all parties concerned to utilize water otherwise going to waste, and an arrangement was entered into whereby the Presidio agreed to operate its filter and pumping plant continuously to its full capacity and sell the Exposition all water in excess of its own requirements.

126 The Presidio allowed the Exposition the use of one of the compartments of its storage reservoir and the arrangement was such that when the Presidio's compartment was full the excess would spill into the compartment allotted to the Exposition. This reservoir is at an elevation of 384 ft. and the capacity of one compartment is 3,000,000 gal. As soon as this contract was closed, the U. S. Government allowed the Exposition to tap its 8-in. line to Fort Mason, which passed through the grounds, thus giving the Exposition immediate relief from a threatened shortage of water during the construction period.

127 *Golden Gate Park Supply.* No time was lost in making tests for water in the vicinity of Golden Gate Park, for it was generally known that a considerable quantity of water was to be found in that district due to the absorption of the rainfall by the sandy soil. Seepage downward is prevented by a strata of clay at a depth of from 60 to 90 ft. from the ground surface, but there is a flow toward the ocean at a rate of from 10 to 30 ft. per day. As the water shed was similar in character to that of Lobos Creek, the yield for the district was predicated on the records of the flow in Lobos Creek.

128 From a careful study of the situation, it was estimated that a supply of 1,000,000 gal. per day could be relied upon during the period of the Exposition, which estimate was to a degree verified during the investigations by the work of a sewer contractor in the vicinity of the proposed plant, who found it necessary to pump 1,000,000 gal. per day to unwater his trench. The first step taken was to construct in the westerly end of Golden Gate Park an infiltration sump similar to that used by the Park Commission for their supply of irrigation water. This sump was made 200 ft. long, 14 ft.

wide and 30 ft. deep. Tongue and groove sheet piling was used to keep out the surface water, while removable screens placed near the bottom collected the water of desired quality. This sump developed a yield which indicated it could be depended upon for 350,000 gal. per day, but owing to the high cost of construction for the output secured, it was decided to try sinking wells, although it was brought to the notice of the engineers that wells had been tried before in that territory and abandoned on account of inability to keep out the sand.

FIG. 27 FILTRATION PLANT. FILTER HOUSE IN FOREGROUND

129 The desperate situation justified taking a chance and a test-well was bored just outside the park, which was brought up to a capacity of 368,000 gal. per day without encountering any difficulty with sand. This well was 80 ft. deep with an outside casing 20 in. in diameter and an inside casing 16 in. in diameter. Both casings were perforated with slots having 1/16-in. openings, and the annular space between casings filled with gravel ranging from rice to pea in size. Encouraged by the performance of this well, 4 more of the same size and type were drilled in the same vicinity, varying in depth from 70 to 87 ft. The material encountered was a fine beach sand having an effective size of 0.18 mm. and a uniformity coefficient of 1.55. The wells were all located within an area of 650 ft. square, and

it was assumed that while one well would probably yield 260,000 gal. per day during the exposition period, mutual interference would bring their combined capacity to about 2¾ times the capacity of one well, or 650,000 gal. per day for the five wells.

130 *Filtration Plant.* Considering shallowness of the wells and the possibility of contamination due to the water shed being inhabited, it was decided to filter the water and treat it with chlorine gas. Realizing that the plant would probably end its usefulnes with the Exposition, a design was adopted in which first cost was the prime consideration. The elements of the plant consist of a measuring chamber into which the water is discharged from the wells and sump, a raw water reservoir where the coagulant is introduced, three rapid sand filters and a clear water reservoir. A uniform flume type construction was used for all elements, the adoption of which simplified the framing, foundations and erection. A bulkhead placed near one of the raw water reservoirs forms a measuring chamber 12 ft. 6 in. by 9 ft. 6 in. by 9 ft. 3 in. deep. Two 4-in. by 12-in. submerged orifices are located in this bulkhead. The supply lines from the sump and wells discharge into this chamber and are fitted with control valves actuated by floats located in the raw water reservoir. A differential gauge indicates the difference in levels between the water surfaces in the measuring chamber and raw water reservoir, from which the rate of flow is determined.

131 The raw water reservoir is 64 ft. long by 12 ft. 6 in. wide and 8 ft. deep, its size having been determined by the period of time required for coagulation with 4 filters operating at maximum rate. The treated water flows from this reservoir to the filters through a 2-ft. by 3-ft. flume located between the row of filters. The supply to each filter is controlled by a shear valve located in the side of the flume. At the far end of the flume is an overflow weir which comes into use in case the water level in the raw water reservoir exceeds the predetermined height.

132 Three filters, each rated at 500,000 gal. per day are in operation, but space and pipe connections are available for a fourth unit should the necessity arise. The units are 14 ft. 8 in. long, 12 ft. 6 in. wide and 8 ft. deep. The collector system employed is a departure from the usual types and has been most successful in operation. The design was developed by Prof. Charles Gilman Hyde, and in addition to its effectiveness has the merit of low first cost. The collector in each filter consists of a header, rectangular in section, bolted to the bottom of the filter. The effluent pipe is run through the

bottom of the filter to the center of the header from which point the header is tapered so that the loss in head is fairly uniform throughout its length.

133 On the top of the header, openings are provided for the 3-in. nipples which make the connection to the 2½-in. lateral pipe. These

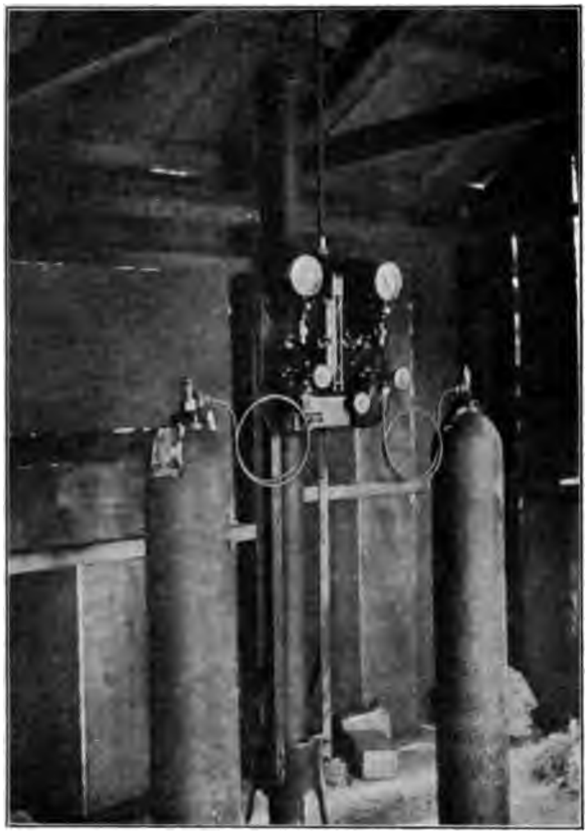


FIG. 28 APPARATUS FOR APPLYING CHLORINE TREATMENT TO WATER SUPPLY

laterals are placed on 6-in. centers and extend the full width of the filter. Holes 5/16 in. in diameter and 3 in. apart are located on the under side of the laterals. The loss of head through each of these small holes being the same, and great as compared with the losses elsewhere, insures a uniform collection of the effluent. Coarse gravel was placed in the bottom of the filter to a point 2 in. above the laterals, then three layers of graded gravel, each 3 in. thick, followed by 26 in.

of sand having an effective size of 0.35 mm. and a uniform coefficient of 1.5.

134 The depth of water over the top of the sand is 30 in. The effluent flows from each filter through an 8-in. riveted steel pipe to the clear water reservoir, where an effluent controller of simple design is installed. The controller is set in a compartment located within the clear water reservoir with water connection thereto through a submerged weir. The level of the water in this compartment is kept at a constant height by a float actuating the controller, which insures a fairly constant rate of flow. Should the water in the clear water reservoir reach the overflow point, an auxiliary float closes the effluent controller. Wooden floats are used throughout and the simplicity of the effluent controller may be realized by the fact that three of them, 8 in. in size, cost less than $50 to make. All of the measuring or controlling devices used in this installation are home made, and while some of the equipment may appear to be crude it does the work with all of the accuracy essential to operation.

135 The washing of the filters is done by the reverse current method, the water being forced through the collector system at a high velocity. A 10-in. centrifugal pump is used for this purpose and the rate of washing is 2-ft. vertical rise per min. in the filter, which is equivalent to 15 gal. per min. per sq. ft. of sand surface. Washing at this rate raises the sand bodily about 12 in. and keeps it in motion. Two wooden V-troughs in each filter collect the wash water. No appreciable disturbance of the gravel has been noted although no means have been provided to prevent it. It takes about 5 min. to wash a filter, the wash water being applied for 2 min. Owing to the water being practically free from turbidity, it is only necessary to wash the filters every 10 hours.

136 *Clear Water Reservoir.* The clear water reservoir is 96 ft. long, 12 ft. 6 in. wide and 7 ft. 6 in. deep and holds 60,000 gal. of water. From the bottom of this reservoir, and located on the side directly opposite that on which the effluent controllers are situated, three 6-in. suction pipes are run, one to each of the three units in the main pumping plant. Over that part of the reservoir where the controllers are installed is a house which contains the sterilizing equipment.

137 This equipment is the exhibit of the Electro Bleaching Gas Company, of New York, and consists of a gauge board, shut-off valves, gauges and a pressure reducing valve. The quantity of gas applied is indicated by the position of a small glass float located within a tube.

138 The apparatus is easy of adjustment and does not require a skilled attendant. In order to keep a check on the quantity of chlorine used, the cylinder in use is kept on a platform scale, and it has been found that the quantity determined from the gage readings checks within 1 per cent of the weighed quantity. The chlorine is applied at the rate of 3 lb. per 1,000,000 gal. of water, and bacteriological tests have demonstrated that the water is at all times perfectly safe for drinking purposes.

139 *Main Pumping Plant.* The water is taken from the clear water reservoir and pumped to the Presidio reservoir by three 5-stage 6-in. centrifugal pumps, each rated at 500 gal. per min. against a pressure of 220 lb. These pumps are belt driven by 125-h.p. induction motors. The pump house is located alongside of and about 5 ft. lower than the clear water reservoir so that the pump suction lines are under a pressure head at all times. The main pumps force the water through a 12-in. main to the Presidio reservoir, which is 19,300 ft. from and 365 ft. above the pumping plant.

140 *Trouble with Wells.* While no trouble has ever been experienced with the test-well mentioned heretofore, the other four wells gave trouble from the start. It is hard to explain this difference as the wells were all of identical construction and the formation encountered was the same for all. The pumps used in these five wells are vertical, 3-stage, centrifugal pumps, belt-driven by 20-h.p. motors, and operate at 1400 r.p.m. against a 75-ft. head. A foot valve was placed below the pump, but as it was the cause of the pump becoming choked with sand it was discarded and a check valve placed in the discharge line from the pump. Each of the wells in turn started to sand up as soon as any considerable amount of water was pumped, with the result that the pump soon choked and it became necessary to withdraw the pump and sand-pump the well.

141 Although the pumps were supposed to be adapted for handling water containing sand, the bronze bearing bushings were in some instances cut out with one day's operation. This trouble was largely overcome by discarding the tubes which enclosed the shafting, and using leather rings above and below each bushing, securely fastened thereto and fitting snugly on the shafting. By operating the pumps at capacities between 60 and 100 gal. per min., depending upon the well, continous flow could be obtained, but a certain amount of sand would be discharged and the wells soon fill up with sand. In order to avoid the expense and delay incident to removing the pumps and sand-bailing the wells, an air ejector was made which could be

lowered into a well without disturbing the pump. By this means 50 ft. of sand was taken out in 3 hr., a large part of the time being consumed in putting down and removing the piping. The removal of such large quantities of sand soon caused the ground around the casing to cave, with resulting disturbance to the foundations of the building and equipment.

142 It became evident that the wells as they stood would never give satisfaction, and it was decided that their failure was due to the size of the gravel which had been placed in the annular space between the outside and inside casings. In the belief that a suitable material could be found which would keep the sand out and at the same time allow the water to flow through freely, it was decided to experiment with various mixtures of sand and gravel and different perforations for casings.

143 Flat test plates 8 in. by 12 in. were prepared, each having a different size of perforation. These were tested by placing them in a wooden frame box with trial mixtures of sand and gravel placed between the plates. Above the plates a quantity of the sand was placed that had been taken out of the wells, and water pressure applied. The flow of water was measured by meter and the pressures taken for different rates of flow. The result of these experiments was the selection of a naturally graded beach sand which had an effective size of 0.42 mm. and a uniformity coefficient of 2.18. It was determined that with the use of this, sand plates could be used having a perforation of 1/32 in. in width and that the fine material from the wells would be held back even when the velocity through the perforations was as high as 0.75 ft. per sec. As the maximum velocity through the perforations under operating conditions would never exceed 0.5 ft. per second, it was considered safe to use this material.

144 As the wells, except No. 1, were not giving more than half the yield contemplated, it was decided to put down additional wells, making use of the data which resulted from the experiments. The contractor's price for the old wells was $7.50 per ft. and it appeared that new wells could be put down cheaper with the Exposition's force by using a hydraulic ejector instead of a well driller's standard equipment. The site chosen for the trial well was close to the main pipe line where water would be available at 175-lb. pressure. The hydraulic ejector or sand-pump was made with standard fittings but had a machined brass throat piece. The well drilling rig and ejector are shown in the illustrations.

145 In starting, the bottom section was made up of a 5-ft. length

of inner and outer casing, the inner casing being tapered to the diameter of the outer at the bottom and both riveted to a forged steel shoe. A hole was dug 5 ft. deep at which point water was encountered, and the bottom section set in place. The hydraulic ejector was placed inside and excavated the material down to the shoe. The first 15 ft. of casing went down of its own weight very rapidly, but for the balance of the distance it had to be forced down by means of the long lever shown. The maximum force applied was about 5 tons and the average rate of travel of the casing was 1 in. per min.

146 After 25 ft. of casing had been placed it was found that the

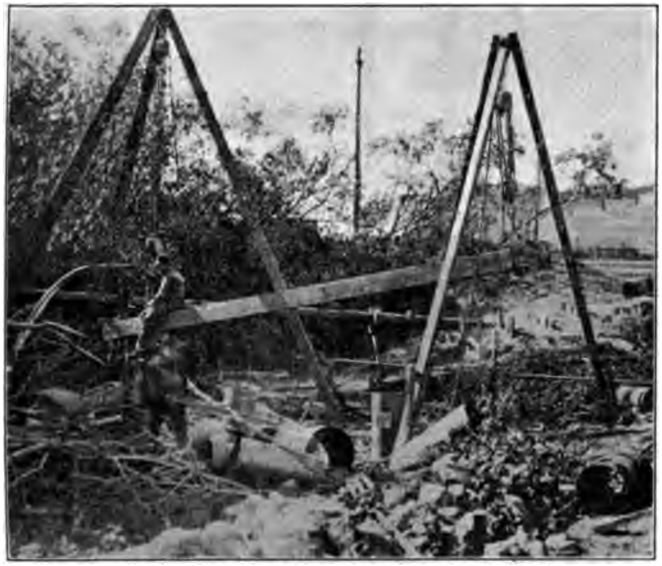


FIG. 29 RIG USED FOR DRILLING NO. 6 WELL

well was badly out of line due to the removal of too much material from one side. The casing was straightened up by putting a jet down one side and keeping a strain on the top of the casing by means of block and tackle. Guides were put on the ejector suction to keep it centrally located and no further trouble was experienced with the casing getting out of line.

147 It required 5 days to complete the well to a depth of 60 ft. The outer casing used for this well was 22 in. in diameter, No. 12 gauge steel, with collar joints, each section being 30 in. long, with perforations 1/16 in. wide by 1¾ in. long. The inner casing was

16 in. in diameter, of similar construction but with perforations 1/32 in. wide. The area of the perforations was estimated at 0.0329 sq. ft. per running foot of pipe. Perforated casing was used throughout the depth of the well except for the bottom section and the upper 5 sections. The material used in the annular space between the inner

FIG. 30 PERFORATED WELL CASING AND HYDRAULIC EJECTOR

and outer casings was that determined by experiment, and was put in place as the casing was lowered.

148 The well, on test, developed a capacity of 200 gal. per min. without discharging sand. Instead of drilling additional wells of this type, which would take time and involve additional pumping equipment, it was decided to reconstruct all of the other wells, except No. 1.

149 Efforts to remove the inside casing met with no success.

Fortunately the diameter of the deep well pumps was such as would allow the placing of a 13½-in. casing inside the well. One of the wells was equipped with this size of perforated casing, and the space between it and the 16-in. casing was filled with the same grade material as that used in the No. 6 well. The installation of this casing was a comparatively simple matter and required about 8 hr. work for an 80-ft. well. Under test the remodeled well operated continously at a rate of 243 gal. per min. and no sand appeared in the discharge. The use of the additional casing and the sand increased the resistance to the flow of the water, and for the same yield the water surface had to be drawn down about 30 per cent lower than before. The rest of the wells were reconstructed in a similar way, and have all been brought up to capacity, without trouble from sand.

150 This subject has been discussed at some length in the belief that it indicates the possibility of overcoming troubles which are quite prevalent where wells are drilled in sandy soil, by adopting the type of construction described and selecting a material on the basis of similar experiments. It was thought that the fine sand from outside the well might eventually plug the voids in the coarse sand between the casings but so far there has been no evidence of this action. Should plugging occur it would be a simple matter to remove the coarse sand from between the casings by means of a hydraulic ejector or air lift, and either replace the sand with new material or wash the fine material out of the old sand.

151 *Wells in the Exposition Grounds.* In October, 1913, three wells were sunk in the grounds, each of which produced a yield of from 50 to 150 gal. per min. The water exhibited turbidity at times and the performance of the wells was not generally satisfactory, and they were abandoned as soon as the Golden Gate supply was brought in. These wells were of considerable use during the construction period and helped out during the shortage of water around opening day. Owing to the cost of operating the wells and their small output, together with the questionable character of the water from a sanitary viewpoint, these wells are no longer used except for filling pools.

152 *Spring Valley Supply.* The only water taken from the Spring Valley Water Company at present is for the Fisheries exhibits and to supply an occasional shortage of water. On opening day, owing to the large amount of water which had been used for filling pools and fountains and that used incident to cleaning up the Grounds, there was a temporary shortage of water, and to meet this critical condition the Exposition made connection to the fire hydrants outside

of the Exposition grounds and pumped water from the Spring Valley system into the Exposition's mains, and obtained further relief by making a connection to the City's high pressure system. This situation continued off and on for several days until the Presidio reservoir was replenished by the output from Lobos Creek. In this connection it should be said that, had the Golden Gate Park System been completed on opening day, no such shortage would have occurred.

153 *Amounts of Water Used.* Below is given a table showing the quantities of water used and the sources of supply from Feb. 20th to July 31st, inclusive:

Month	Government water	Park water	Spring Valley water	Zone wells	Total (gal.)
Feb. 20–28	723,937	....	226,775	204,000	1,337,500
March	733,523	347,290	106,990	34,703	1,228,965
April	724,270	591,760	29,593	....	1,345,617
May	612,887	564,636	19,116	....	1,205,636
June	473,057	1,057,730	54,910	....	1,585,694
July	398,460	1,060,094	138,696	....	1,597,250

154 It will be noted that the supply obtained from the Presidio fell far short of the estimated amount, 800,000 gal. per day, and that the Golden Gate Park supply has been increased materially over the assumed yield of 1,000,000 gal. per day. Additional wells in the vicinity of the park are being developed which will probably contribute 500,000 gal. per day to care for the increased consumption to be expected in August and September, and offset a possible decrease in the yield of the old wells.

155 The maximum draft noted at any time shows that water was being used for short periods at the rate of 4,250,000 gal. per day. On average week days 60 per cent of the total consumption is used between the hours of 8.00 A. M. and 4.00 P. M., and on Sundays, (when there is no irrigation) about 47 per cent during these hours.

156 *Cost of Water.* The cost of water per 1000 gal. from the Golden Gate Park plant is 1.517 cents for pumping from the deep wells to the filters, 0.974 cents for filtering and treating with chlorine, and 3.935 cents for pumping to the Presidio reservoir, making a total cost of 6.426 cents per 1000 gal. delivered in storage. These costs cover operation and maintenance only. The water obtained from the Presidio is sold to the Exposition for 7½ cents per 1000 gal., while that taken from the Spring Valley Company averages 19½ cents per 1000 gal.

157 *Quality of Water.* Tests of the water from Golden Gate

Park are made weekly by the Federal Laboratory and in no instance has the treated water failed to meet the U. S. Government requirements for potable water. The following is a representative test:

Raw Water..............	Gas in 10 c.c., colon types count, 100 per c.c.
Effluent..................	Gas in none of the specimens, count, 12 per c.c.

158 *Construction Costs.* The filtration plant, including the pump houses for the main pumps and wells, together with the pipe line to the wells, cost $14,575. The cost of the first five wells was $3,075 and that of the No. 6 well $255. The cost of the sump was $14,979. The pumps, which included three 5-stage main pumps, one 4-in. vertical centrifugal sump pump, five 6-in. deep well pumps, and one 10-in. wash water pump, cost $4,962. The electrical equipment was rented. The cost of the pipe line from the pumping plant to the Presidio reservoir, and from the Presidio reservoir to the Exposition grounds, was $37,150. The total expenditure to develop a water supply and deliver it to the exposition grounds was $77,000.

159 *Distribution System.* Converse pipe was adopted for the domestic water distribution system, on account of its low initial cost and high salvage value. Approximately 59,000 ft. of this pipe was laid in 4-in., 6-in., 8-in. and 10-in. sizes, amounting to about 850 tons. All pipe was dipped locally and all valves and fittings protected by two coats of paint having an asphaltum base.

160 Mains were run in all important roadways and courts. Each exhibit building had a ring main installed about 50 ft. inside the building perimeter which was fed from each side of the building by 4-in. service pipes. In most of the buildings the mains were suspended from the floor joists, but in the Palace of Machinery they were run overhead in the bracing trusses. Connections to consumers were made to the ring mains in the buildings, and in other locations the services were run to the street mains. All water sold is metered and the charges range from 30 cents per 100 cu. ft. for amounts used per month up to 5000 cu. ft., to 11 cents per 100 cu. ft. for a consumption of 80,000 cu. ft. or more per month.

161 In order to reduce the demand on the domestic water supply, salt water is used in the Palace of Machinery for cooling purposes and at the Race Track for sprinkling.

TELEPHONE SYSTEM

162 One of the important activities of the Exposition is the operation of the telephone system within the grounds. An arrangement was made with the local telephone company, whereby the necessary equipment was rented to the Exposition and the latter installed the distribution system. An 18-position switchboard was installed in the Food Products building, at which point the trunk lines from the local company were terminated. In laying out the distribution system, the grounds were divided into sections and cables run to the approximate center of each. Cross connection boxes were installed at these points, from which feeder cables were run where required. The conduit system installed for the electric distribution was also used for the underground telephone cables.

163 In the buildings twisted pairs were run in rings along roof trusses or under the floors, while in the Zone district the subscribers were connected to a cable fastened to the fences in the rear of the concessions. In the States and Foreign sites cables were run to a few of the large buildings, and feeder cables from these points to other buildings. The underground distribution system required the installation of 47,700 ft. of cable (9,474,800 ft. of wire), of which 4300 ft. was 400-pair cable. Over 1,100,000 ft. of duplex wire was used in the overhead distribution.

164 With the view of facilitating the work of the exposition's forces, private branch exchanges were installed in a number of the large departments. These exchanges had interconnecting lines in addition to direct lines to the main board and to one of the exchanges of the telephone company, the latter line being used only for outgoing calls. Fifteen attended pay stations with 100 booths were installed, one in each of the main buildings, three on the Zone and two at the entrances. These were later changed to non-attended stations as the income did not warrant the expense of attendants.

165 In addition to the foregoing, 53 telephones with coin collectors were installed at convenient points around the grounds. The Exposition installed 732 telephones for its own use and 1244 for the needs of its subscribers and pay stations. Subscribers are charged on the basis of the equipment required, the charge for a 1-party station being $6 per month. All services are metered and the rates for calls terminating within the City vary from 5 cents to 3½ cents each, depending upon the number used in any one month.

166 No one element of the organization contributed more to the

successful opening of the Exposition on time than did the telephone department. The demands upon the telephone system around opening day were unprecedented, the number of calls handled per day reaching 49,500.

167 The satisfactory service rendered was due largely to the installation of an adequate plant and a flexible distribution system, to the coöperation of the local telephone company in handling traffic and to the employes recruited from the telephone company and selected for their special fitness. The full operating crew was trained for a month in advance of opening day so they were thoroughly familiar with the plant when the peak load came.

CONCLUSION

168 The president of the Exposition, Mr. Chas. C. Moore, has been identified with many successful engineering enterprises, and when he undertook the responsibility of this great project selected an engineer for the position of Director of Works. At previous expositions the director of works had always been an architect, and there was much criticism over the appointment of an engineer for this position, but the choice has been fully justified by the artistic results obtained and the completion of the Exposition on time and within the amount appropriated, a performance not usually associated with the construction of expositions or other monumental groups of buildings. There was some fear that with engineers in charge, whose efforts might be expected to be in the direction of keeping the costs within the allowances, there would be decisions which would seriously affect the aesthetic value of the various decorative features, and would stifle the zeal of the architects, sculptors and artists. But while the designers had their heads in the air, the engineers had their feet on the ground and many features were changed or eliminated to keep the expenditures within limits. These changes were made without friction, as the engineers were appreciative of the efforts of the architects, sculptors and artists and thereby secured their assistance toward the desired end.

169 The Exposition engineers have had an unusual opportunity not only to carry out their ideas in matters of construction, but to manage the operation of a great enterprise. Investigation will show, in this instance, the versatility of the engineer and his fitness by training and temperament to analyze a situation and to foresee difficulties and provide against them.

No. 1491

MECHANICAL ENGINEERING AT THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION

By George W. Dickie, San Francisco, Cal.
Vice-President of the Society

Before taking up the subject of this paper, I would like to say something regarding the Exposition itself. It would be impossible to portray, either by word or illustration, an adequate conception of the wonderful external beauty of this, especially when seen by any of the varied effects of light and shade so common to the gateway between the ocean and the great Bay of San Francisco. It may always remain a doubtful point whether this "city" of many colors which stood for one year by the Golden Gate appeared more beautiful when emerging from the mist of early morning in all its dazzling wealth of color; when standing in all its richness of detail under the cloudless mid-day sky; or when fading away in the twilight to burst into new and brighter splendor under a deluge of electric beams of light with new and wonderful effects from tinted towers and domes. From the hills which form the background of this wonderful creation, the scene is that of an ever-shifting panorama, the view from each point being a picture in itself.

2 Those who have been fortunate enough to see only for a day or two this wonderful exposition of the sculptor's art, the architect's science and the builder's skill, will never forget the charm of its setting; while those of us who have seen it through all the changing seasons of a California year, taking on new glories with the changing blossoms of the flowers out of which it springs, will find our thoughts taking new and better forms from this experience on through all the thinking days of our lives.

3 In some respects the Panama-Pacific International Exposition

Presented at the Panama-Pacific International Exposition Meeting, San Francisco, September, 1915, of The American Society of Mechanical Engineers.

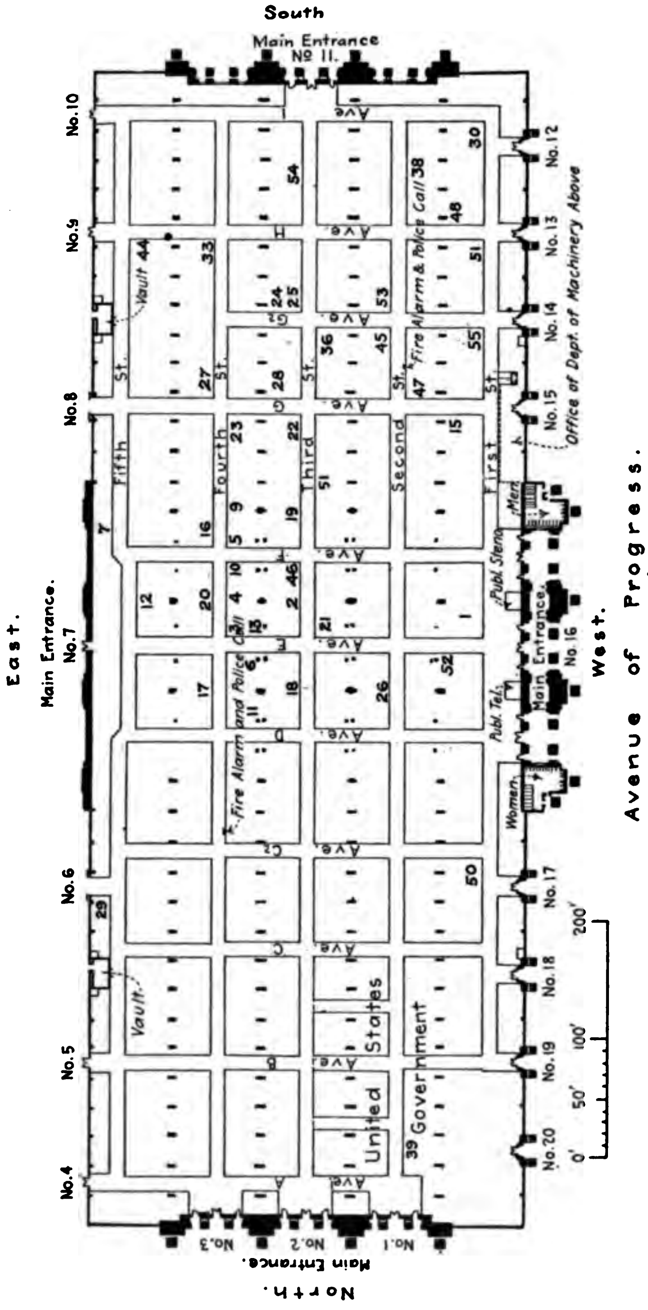


FIG. 1 FLOOR PLAN OF PALACE OF MACHINERY. NUMBERS ON PLAN INDICATE LOCATION OF EXHIBITS, SEE TABLE 1.

TABLE 1 EXHIBITS AT THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION MENTIONED IN THIS PAPER

No.	Exhibitor	Location of Exhibit
1	Standard Gas Engine Company	Palace of Machinery
2	Union Gas Engine Company	Palace of Machinery
3	Gas Engine and Power Company	Palace of Machinery
4	C. L. Seabury Company	Palace of Machinery
5	Van Blerk Motor Company	Palace of Machinery
6	Imperial Gas Engine Company	Palace of Machinery
7	Buffalo Gasoline Motor Company	Palace of Machinery
8	Wisconsin Machinery and Mfg. Co	Palace of Transportation
9	Loew-Victor Engine Co	Palace of Machinery
10	Waterman Marine Motor Co	Palace of Machinery
11	August Mietz	Palace of Machinery
12	New London Ship & Engine Co	Palace of Machinery
13	Fulton Mfg. Co	Palace of Machinery
14	International Harvester Co	Palace of Agriculture
15	Western Gas Engine Corporation	Palace of Machinery
16	Doak Gas Engine Co	Palace of Machinery
17	Bessemer Gas Engine Co	Palace of Machinery
18	Busch-Sulzer Bros. Diesel Engine Co	Palace of Machinery
19	McIntosh & Seymour Corporation	Palace of Machinery
20	Pelton Water Wheel Co	Palace of Machinery
21	Tinius Olsen Testing Machine Co	Palace of Machinery
22	Warner & Swasey Co	Palace of Machinery
23	Morton Mfg. Co	Palace of Machinery
24	Fred. Ward & Sons	Palace of Machinery
25	Gould & Eberhardt	Palace of Machinery
26	Crane Co	Palace of Machinery
27	Landis Tool Co	Palace of Machinery
28	Wm. S. Doig Co	Palace of Machinery
29	Geo. D. Parker	Palace of Machinery
30	Max Ams Machine Co	Palace of Machinery
31	E. W. Bliss Co	Palace of Machinery
32	U. S. Army Ordnance Department	Palace of Machinery
33	Carborundum Co	Palace of Machinery
34	Davis-Bournonville Co	Palace of Manufactures
35	Henry Disston & Sons, Inc	Palace of Manufactures
36	Dreis & Krump Mfg. Co	Palace of Machinery
37	Geometric Tool Co	Palace of Varied Industries
38	Hydraulic Press Mfg. Co	Palace of Varied Industries
39	Henry G. Thompson & Sons	Palace of Varied Industries
40	Larsen Ice Machine Co	Palace of Food Products
41	York Mfg. Co	Palace of Food Products
42	Vulcan Iron Works	Palace of Agriculture
43	Automatic Refrigerating Co	Palace of Horticulture
44	Dodge Mfg. Co	Palace of Machinery
45	Union Oil Co. of Cal	Palace of Machinery
46	Standard Oil Co	Palace of Machinery
47	A. P. Smith Mfg. Co	Palace of Machinery
48	Shepherd Electric Crane & Hoist Co	Palace of Machinery
49	General Electric Co	Palace of Manufactures
50	Neptune Meter Co	Palace of Machinery
51	National Meter Co	Palace of Machinery
52	Luitweiler Pumping Engine Co	Palace of Machinery
53	Layne & Bowler Corporation	Palace of Machinery
54	Krogh Mfg. Co	Palace of Machinery
55	American Well Works	Palace of Machinery

is not equal to some of the great international exhibitions of the past, yet it shows very clearly the changes which have taken place during the past 22 years, or since the Columbian Exposition, particularly in mechanical engineering. At the lake front of the Chicago Exposition, there was a full-sized model of the first three battleships then building for the U. S. Navy. This model was built of bricks and concrete on piles and in it the Navy Department had a very instructive exhibit. Today the actual battleship, thus portrayed on Lake Michigan, lies in front of our Exposition as an illustration of a type now passing away, but with a twenty years' history to think of, of which the model on the lake shore gave no intimation.

4 The first thing that the mechanical engineer observes on entering the Palace of Machinery at the Panama-Pacific International Exposition is the entire absence of the steam engine. There is not a steam engine of any kind in operation, nor a steam boiler under steam. This is the first international exposition where the steam engine, as a prime mover, has been so conspicuous by its absence. On first thought the engineer might take this as evidence that the steam engine, after a century or more of development and a service to industry beyond reckoning, had suddenly reached a stage of innocuous desuetude and had ceased to rank as the greatest machine that the mechanical arts had produced. On second thought, however, he would have to admit that, of all the power developed by the combustion of fuel, more than 90 per cent is developed in the cylinders of steam engines or the rotors of steam turbines. His third thought would probably bring him to the conclusion that the steam engine, in all its varied forms, has reached that stage of development when no striking improvements can be looked for; in fact, the steam engine has in a sense reached its prime, and may be expected to continue doing a large proportion of the work rendered possible by the combustion of fuel, either liquid or solid. While the steam engine was advancing to the place it now occupies and many improvements were being made, the builder with something new, something that he considered an advance, was ready to show his best work at an exposition. This great prime mover has now reached maturity, and will only take second place when the internal combustion engine has reached such a robust maturity as to enable it to do on a much smaller diet what our great steam engines are doing.

5 The fact that California is a great fuel oil producing state may be one reason why the steam engine forms no part of the machinery

exhibits; yet the great bulk of California oil used for fuel is consumed under steam boilers. There is an impression among engineers, and especially among the younger members of the profession, that the steam engine is nearing the close of a great career, and on that account is too old a thing to attract much attention at an exposition.

6 The separate generator and the separate condenser, the crowning achievements in the early days of steam engines, are, in these days of quick returns, considered antiquated; and, acting on the principle that there is no time to be lost in dealing with heat, we are now advised to burn our fuel as rapidly as possible in the engine

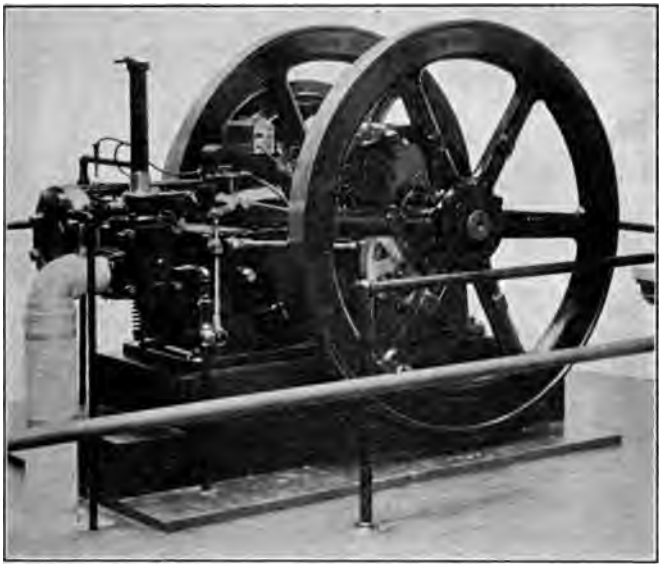


FIG. 2 STATIONARY GAS ENGINE EXHIBITED BY WESTERN GAS ENGINE CORPORATION, CYLINDER 12½ IN. BY 20 IN. 235 R.P.M. 60 H.P.

cylinder, to get the power out of it instantly and let it go—hence the internal combustion engine.

7 There is quite a large display of gas engines in the Palace of Machinery, the Transportation Palace, and the Palace of Agriculture. These engines may be classed as:

8 (*a*) *Marine engines, Otto type, constant volume class,* for distillate fuels such as gasoline and kerosene. In this class there are some fine exhibits, such as the Standard Gas Engine Company's exhibit, showing a good line of well designed and carefully finished

engines from 100 h.p. down to small sizes. The Union gas engine, shown cut open to illustrate the movements of piston and valve gear, is a fine type of marine gas engine. The Gas Engine and Power Co. and C. L. Seabury Company show interesting examples of engines.

9 The Van Blerck Motor Co. exhibit some handsome engines of attractive design, which present new features. The Imperial Gas Engine Co. have an attractive exhibit of substantial-looking engines; while others in the same class, such as the Buffalo Gasoline Motor Co., the Wisconsin Machinery and Manufacturing Co., the Loew-Victor Engine Co. and the Waterman Marine Co. are all well worthy of study. This type of marine gas engine is well represented at the Exposition.

10 (*b*) *Marine engines, Otto type, constant volume class,* with injection for heavy oils. Of this type there is but one example, that of August Mietz. It is a well designed and strongly built engine.

11 (*c*) *Marine Diesel oil engines.* In this class there are two exhibits—that of the New London Ship and Engine Co. and that of the Fulton Manufacturing Co. The first-named works under a load of about 80 per cent of full power, the normal power being 200 horsepower. These engines illustrate the present state of the art in Diesel engines of moderate power.

12 (*d*) *Stationary engines, Otto type, constant volume class.* for distillate fuels such as gasoline and kerosene. Of this class a great variety is shown, suitable for a wide range of work. Mention might be made of the exhibit of the International Harvester Co., which shows these engines designed for all purposes to which motors can be applied on the farm and all other rural work. There is a rich field in this exhibit, which gives information as to the best way to hitch up a gas engine to almost any kind of work.

13 The Western Gas Engine Corporation exhibits stationary gas engines which present some features worthy of study. These engines have an unusually long stroke for gas engines, and were the first to introduce water into the carburetor. The valve gear is very simple, with free-moving parts operated by a rod from a single eccentric.

14 The Doak Gas Engine Co. and the Bessemer Gas Engine Co. show some good work under this class. The Standard Gas Engine Co.'s exhibit is also well worthy of attention.

15 (*e*) *Stationary engines, Otto type, constant volume class,* with injection for heavy oils. Under this type some very good exhibits

are shown, notably those by August Mietz and the Bessemer Gas Engine Co.

16 (*f*) *Stationary engines, Otto type, constant volume class, for gaseous fuels.* In this class there are two exhibitors, the Standard Gas Engine Co. and the Western Gas Engine Corporation. As the arrangements of valve gear, etc., of these engines are quite different in design and function, they offer a good opportunity for comparison.

17 (*g*) *Stationary Diesel oil engines.* In this class there are

FIG. 3 500-H.P. DIESEL ENGINE EXHIBITED BY BUSCH-SULZER BROS. DIESEL ENGINE CO.

two exhibitors, the Busch-Sulzer Bros. Diesel Engine Co. and the McIntosh and Seymour Corporation. Both show fine examples of vertical engines of 500 h.p. each and both illustrate the best that these makers produce. Each engine is operating with Star distillate, which is probably the nearest to California crude oil with which it is safe to work these engines. (It is expected that an exhaustive

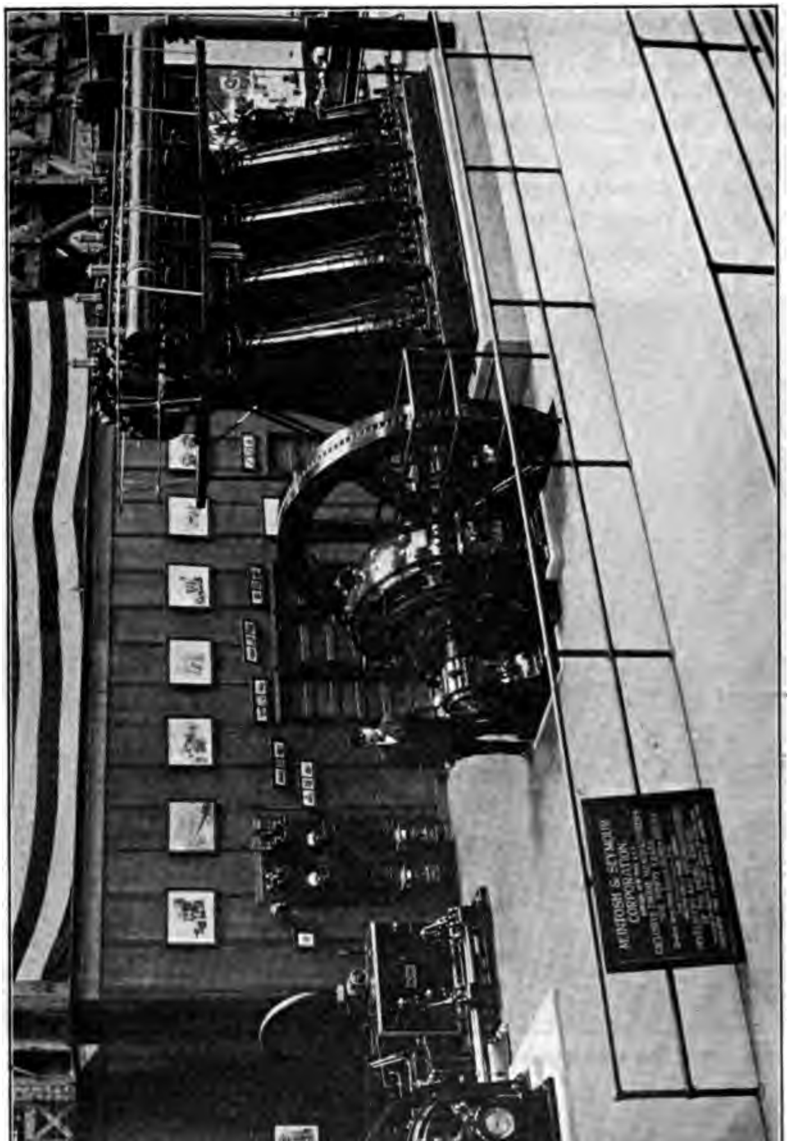


Fig. 4 500-h.p. Diesel Engine Exhibited by McIntosh and Seymour Corporation test will be made of these two engines to determine which should receive the highest award.)

18 On the whole, the exhibit of gas and oil internal combustion engines is complete and illustrative of the present condition of the art, and the mechanical engineer interested in this form of prime mover finds a rich field for study at this Exposition.

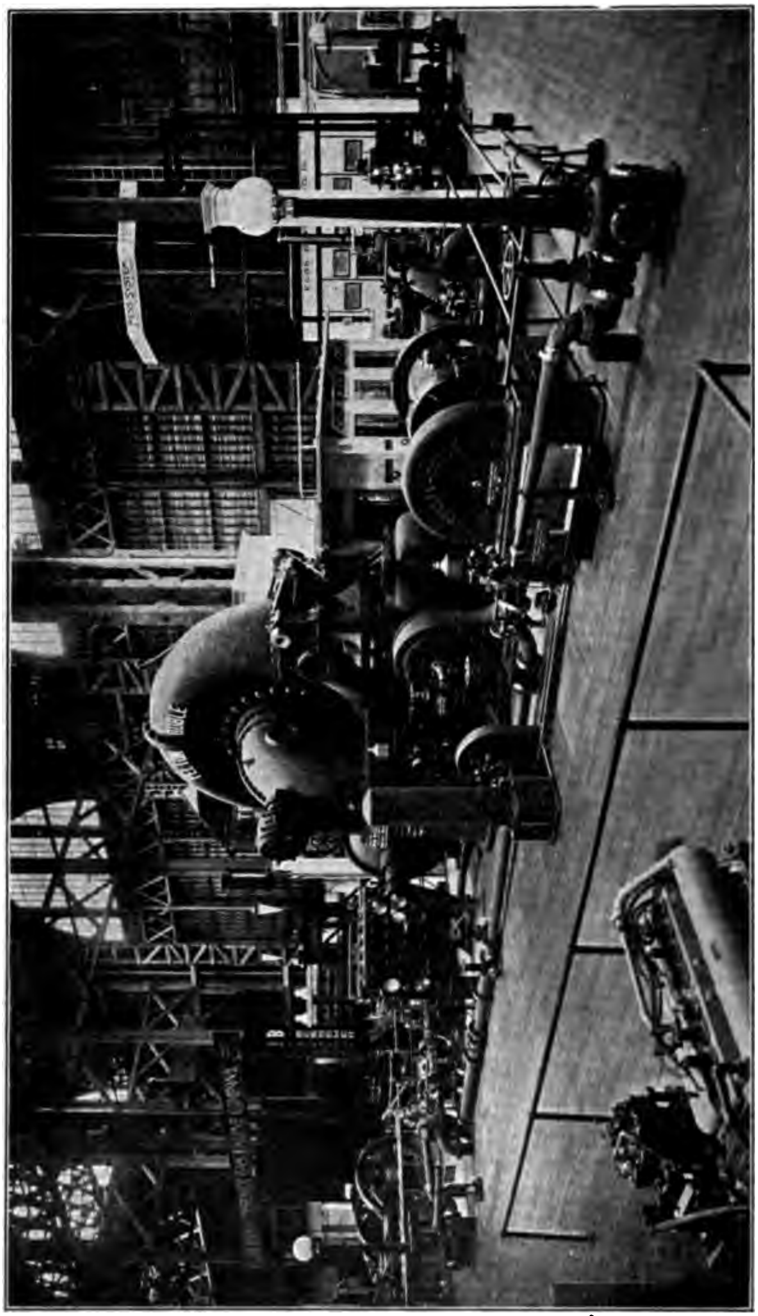
19 In the special types of gas engines or motors shown in the Transportation building, a large and varied display illustrating their application to automobiles, motor trucks and motor cycles, gives ample opportunity for study in this interesting field. The beautiful workmanship on many of these show the taste and skill that has been brought to bear upon the development of this immense industry.

20 California is not only rich in liquid fuel, making it an ideal home for the internal combustion engine, but the whole Pacific Coast is a land of liquid power, ready to be used without the necessity of burning anything to produce it. The winter snows on the high Sierras and other mountain ranges forming the eastern boundary of the Pacific slope is the source of this vast accumulation of stored-up energy, which, in the past thirty years or so, has been more and more converted into electrical energy and carried for hundreds of miles to the points where it is needed, to operate factories and street cars, light cities, warm houses and cook food.

21 It would be surprising, indeed, should there be nothing in this Exposition which is more of an electrical display than any of its predecessors, and that would give the engineer an adequate conception of how the snows of the mountains were utilized in furnishing light, heat and power to the people who live in the valleys and on the plains below. It is not surprising, therefore, that the most impressive exhibit in the Machinery Palace should be that of the Pelton Water Wheel Co. The writer felt his inability to do justice to this splendid example and asked help from W. A. Doble, chief engineer of the Pelton Water Wheel Co. and the designer of much of the display made, who kindly furnished a description from which the following is taken:

22 Hydraulic machinery has played such an important role in the development of the West, that one's expectations are raised quite high. The exhibit of the Pelton Water Wheel Co., covering every phase of rotative hydraulics and hydraulic control, can be divided into two parts: *Operating division,* and *display division.*

23 It being impossible to secure water, either in sufficient quantity or under pressure desirable for driving hydraulic prime movers,



an effective combination of turbine pumps, drawing water from a common sump, and discharging, through control devices and pipe lines of the types customarily employed in hydroelectric practice, into water wheels of both the tangential and turbine types using the sump as a tail race, has been worked out.

24 The operating division consists of two sections:

a A high head pumping project combined with a high head hydroelectric development where water economy is important, and employing tangential water wheels.

b A deep well pumping project combined with a medium head hydroelectric development, employing mixed flow turbines.

25 The high head pumping plant consists of a Pelton-Doble turbine pump, driven through a flexible coupling and herringbone speed-increasing gears by an internal combustion engine. The high speed shaft of these gears turns at 1800 r.p.m., the efficiency of transmission at full rated load of 150 h.p. being in excess of 98 per cent.

26 The engine is a 6-cylinder vertical type, developing 180 h.p. at 350 r.p.m. It uses heavy oil up to 18 deg. Beaumé for fuel and operates on the 4-stroke cycle with maximum compression not exceeding 450 lb. per sq. in. No ignition difficulties are experienced with this low compression.

27 The pump is a single stage uni-diffusion type, having a capacity of 1100 gal. per min. against a head of 300 ft. The impeller, of the closed non-overload type, hydraulic-balanced and made of bronze, is overhung. The discharge line, after passing through an 8 and 6-in. venturi ring, connects to a Pelton-Doble tangential unit, rated at 100 h.p. under 300 ft. head and operating at 300 r.p.m. This unit is of the single overhung, two-bearing type, with the armature of a 75-kw., 250-volt engine-type direct current generator carried between the bearings, and a flywheel overhung on the shaft end opposite the tangential runner.

28 The entire unit is mounted on a high pedestal sub-base, fitted at the water wheel end with a large window, which permits examination of the jet and bucket action.

29 The water wheel disc is steel, with ellipsoidal bronze buckets, saddle-mounted. Water from the high-pressure pump is applied through a needle nozzle, governing being accomplished by an oil pressure type governor mounted on the nozzle casting, and actuating the needle directly through suitable link and rockshaft connections. An auxiliary relief needle nozzle avoids excessive pressure rise in the

pipe line and minimizes water waste during governing cycle. T
needle of this nozzle is connected to a spring-controlled oil cylinc

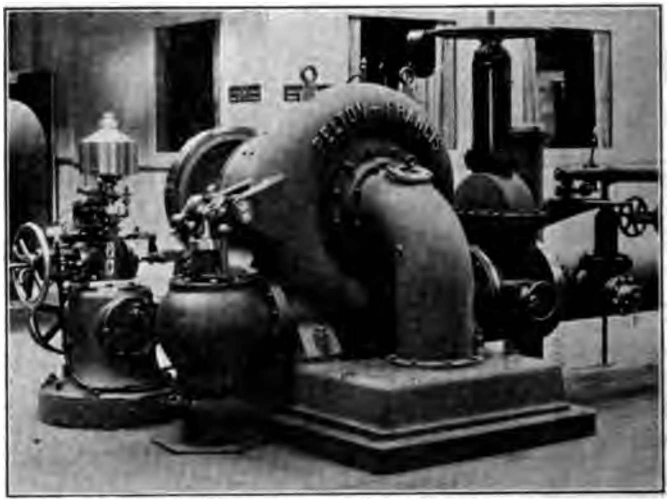


FIG. 6 50-H.P. PELTON-FRANCIS WATER WHEEL. 1400 R.P.M. PELTON WA
WHEEL CO.'S EXHIBIT

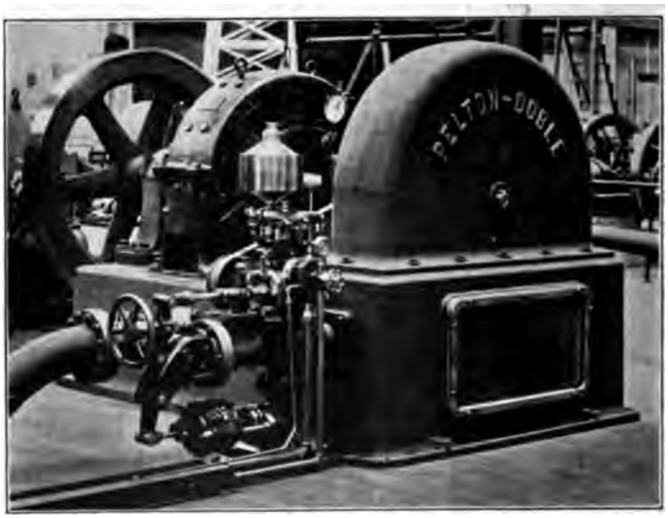


FIG. 7 100-H.P. PELTON-DOBLE WATER WHEEL. 300 FT. HEAD. PEL
WATER WHEEL CO.'S EXHIBIT

known as the cascade cylinder. Ports, adjustable from the outsid
the cylinder, determine the time element of piston travel. The s

link and rockshaft system that actuates the main needle nozzle is connected to the piston of the cascade cylinder, the relative motion (and thus the nozzle area) being equal and opposite in direction when the time element of the cascade cylinder is zero. With this setting, the auxiliary nozzle merely acts as a synchronous bypass, and the water quantity is constant. Where water economy is of importance, the time element of the cascade cylinder is so adjusted that a rate of closing of the main needle which does not raise the surge pressure to a dangerous point will not open the auxiliary needle. Higher closing rate opens the auxiliary needle, which is then returned to closed position by the compression springs of the cylinder mounting at a

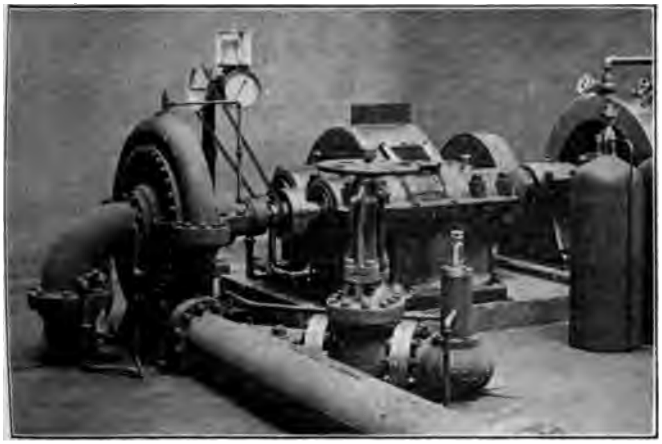


FIG. 8 CENTRIFUGAL PUMP AND HIGH SPEED TRANSMISSION. PELTON WATER WHEEL CO.'S EXHIBIT

rate that will not cause dangerous surge pressures. This is the position of maximum economy. A baffle plate, employing the vortex principle, is set in the casing and receives the impact of the relief stream, quenching its energy.

30 The deep well pumping plant consists of a Pelton-Doble borewell type turbine pump, with a capacity of 4500 gal. per min. against a head of 60 ft. and driven at 1760 r.p.m. by a vertical motor mounted on the pump pedestal. The maximum diameter of the pump is 22½ in., permitting its easy entrance into a 24-in. bored well.

31 The discharge orifice has a diameter of 14 in., and has, bolted directly to it, a riveted steel pipe line which leads to a Pelton-Francis turbine. A venturi ring, 18 in. in diameter, is set in this

line. This unit operates under a head of 50 ft. and develops 50 h.p. at 1400 r.p.m., an unusually high specific speed. The runner is balanced and is of the Francis type. The wicket gates are controlled by a self-contained governor, which actuates a water economizing cascade relief valve as well. The principle of operation is the same as the governor employed with the tangential unit, except that an oil reservoir and belt driven pump are placed in the governor base.

32 The entire operating division is arranged for laboratory investigations, and more than 800 tests have been conducted by independent engineers and the staff of the Pelton Water Wheel Co. During the autumn semester of the local universities, 1915, additional tests will be conducted by senior students of the engineering departments.

33 The display division may be divided into four sections: *Francis turbines, tangential, governor* and *miscellaneous.*

34 The central feature of the entire exhibit is a Pelton-Doble single, overhung-runner, single-discharge turbine, rated at 20,000 h.p. under 500 ft. head, at 360 r.p.m. The entrance valve, casing, draft tube and governor are assembled complete; the runner, of bronze with steel hub and forged wearing rings, and fitted with leakage evacuators, is displayed on a separate pedestal. The main valve, of the butterfly type, is 66 in. internal diameter, with body of annealed cast steel.

35 The tangential section may be divided into two groups: *Hydroelectric prime movers* and *water motors.*

36 Two runners, each rated at 10,000 h.p. under 1350 ft. head at 360 r.p.m., are displayed. These are of the chain connected type, with ellipsoidal cast steel buckets. The jets for these units are 7 in. diameter, and are projected from deflecting needle nozzles with motor operated manual controlled needles and governor controlled deflection. One of these nozzles, assembled complete with motor, and three needles, pedestal mounted, are displayed. A single bucket of the chain connected type, taken from a wheel rated at 14,000 h.p. under 870 ft. head at 200 r.p.m., is also displayed. A number of water motors of standard and special design complete the tangential section.

37 In the governor section are displayed a group of self-contained oil pressure type governors, of 5000 ft-lb. drawbar effort, and, in conjunction with a standard mounted tangential unit, an enclosed type of oil pressure governor having a drawbar effort of 80 lb.

38 The miscellaneous section includes a display of hydraulic giants of the types employed in placer mining and hydraulic fill construction.

39 The exhibits of machine tools are not very numerous and, out-

link and rockshaft system that actuates the main needle nozzle is connected to the piston of the cascade cylinder, the relative motion (and thus the nozzle area) being equal and opposite in direction when the time element of the cascade cylinder is zero. With this setting, the auxiliary nozzle merely acts as a synchronous bypass, and the water quantity is constant. Where water economy is of importance, the time element of the cascade cylinder is so adjusted that a rate of closing of the main needle which does not raise the surge pressure to a dangerous point will not open the auxiliary needle. Higher closing rate opens the auxiliary needle, which is then returned to closed position by the compression springs of the cylinder mounting at a

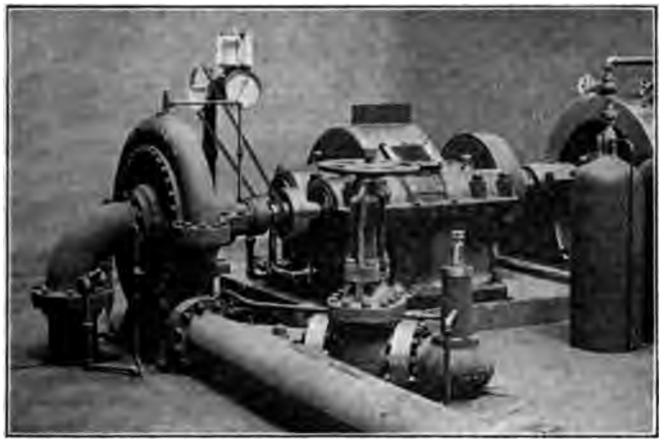


FIG. 8 CENTRIFUGAL PUMP AND HIGH SPEED TRANSMISSION. PELTON WATER WHEEL CO.'S EXHIBIT

rate that will not cause dangerous surge pressures. This is the position of maximum economy. A baffle plate, employing the vortex principle, is set in the casing and receives the impact of the relief stream, quenching its energy.

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46 A few years ago, Mr. Olsen was called upon to design a machine for testing the efficiency of drills, taps, and dies. About two

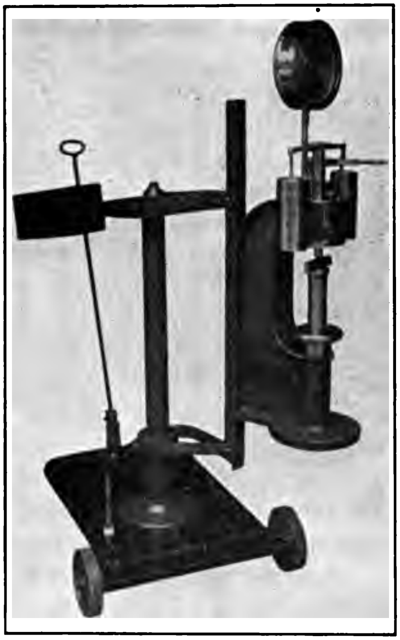

FIG. 10 BRINELL TYPE HYDRAULIC HARDNESS TESTER. TINIUS OLSEN EXHIBIT

years ago this was worked into a practical machine for the Philadelphia Navy Yard and has proved satisfactory, filling all the requirements for this purpose. Later, by additional attachments, this machine has been perfected so as to make a universal tool as well as a tool steel testing machine. This machine in its newest form is shown in the Olsen exhibit; it will test tool steel, lathe and planer tools, milling cutters, drills, taps, dies, rimers, files, and hack saws, thus forming a practical efficiency testing machine for determining the quality of most tools as well as the most effective way to make and use them.

right machine with which to do it. Later, more improved apparatus was made and used by instructors and writers on technical subjects for formulating tables of factors for strains to be used by engineers in the design of different structures. These more or less crude testing machines were used in various places in Europe, and from these the engineering texts and handbooks were first developed.

42 In the United States, prior to 1870, very little progress was made in apparatus for ascertaining the actual strength of materials. One device may be mentioned as used by the U. S. Board of Ordnance for testing the cast iron used in gun making. We know also of one hydraulic press used by Captain Eads for testing the members of the first bridge over the Mississippi River. About 1870, the U. S. Government established the Bureau of Boiler inspection in connection with the U. S. Steamboat Inspection Service. This bureau formulated rules which required that all boiler plates should be tested by having a cupon from each plate subjected to tensile strain for determining the breaking strain, as well as the yielding point or elastic limit, also the reduction of area at the breaking point and the elongation between two points. Hence came the inquiry for and the development of the commercial testing machine.

43 The first machine of 40,000 lb. capacity was designed and built by Mr. Olsen for boiler plate manufacturers. Soon after, about 1870, he built ten such machines for the different branches of the Boiler Inspection Service. Following this development, the technical and engineering schools began to install such machinery. One of the first—if not the very first—was bought from Mr. Olsen by Dr. R. H. Thurston for the Stevens Institute of Technology, Hoboken, N. J.

44 As time passed the testing machine was changed in design and improved and put to more constant and varied use by all manufacturers of structural materials, as well as large users, and also in technical schools, where its use developed laboratories with apparatus of larger and more diversified forms and functions to include investigation of the more varied character and direction of stress to which materials are subjected in modern design.

45 In group *b* are classed what are termed efficiency testing machines. These machines are of more recent development, although a simple machine of this class was shown at the Centennial Exposition, 1876, in a file manufacturer's exhibit; it was probably made by him to demonstrate the most efficient method of file cutting, as well as the most reliable material for file making and the best method of tempering. Nothing more was publicly shown in this line until a few years

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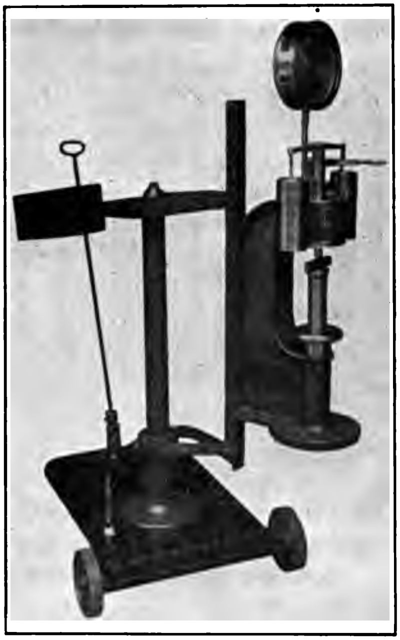

Fig. 10 Brinell Type Hydraulic Hardness Tester. Tinius Olsen Exhibit

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47 The Olsen exhibit is well worth a careful study by the mechanical engineer. Mr. Olsen himself is usually at the exhibit and he looks upon this fascinating collection of testing machines as his own children. As he knows all their little weaknesses as well as their good qualities, an hours' conversation with him is a liberal education in this subject. Here the observer will find the universal testing machine and instruments, spring testing apparatus, cement, concrete, and road material testing machinery; cloth, yarn, paper, rubber, and leather testing machines; oil testing machines, transverse testing machine, and special testing machines including impact, indentation, vibration, bending, hardness, endurance, torsion, and efficiency testing.

48 It is unfortunate that some of the largest machine tool makers of this country who had reserved a large amount of space in the Machinery Palace should have withdrawn from participation in the Exposition on the outbreak of war in Europe under the impression that the Exposition could not be a success under the conditions. This is to be regretted as it leaves a gap in a very important part of the machinery department that the mechanical engineer will not fail to observe. Nevertheless there are some notable exceptions whose presence makes this condition much less felt than it would otherwise have been.

49 Among these exceptions the engineer will not fail to notice the splendid collection of both automatic and semi-automatic tools exhibited by the Warner and Swasey Company. This company shows ten tools, several of them being of new design; the tools are all shown in operation and are attracting the favorable criticism of our best mechanics. Their new universal turret lathes are powerful tools and have some remarkable features. Because of their superior strength and rigidity and the fact that each carriage with its independent feeds is able to operate simultaneously, the lathes are enabled to produce a very large amount of work. They have equal facilities for both bar and chucking work. They have geared heads with splash lubrication. As many as eleven cutters may be used in the turrets and carriages of these lathes, all cutting at the same time and in one set up.

50 The other tools exhibited by this company belong to their line of screw machines and plain turret lathes which are so well known as the product of the Warner and Swasey Co. and are all characterized by the same scientific design and careful workmanship. The automatic boring and tapping machine used by all the large manufacturers of steam and water fittings for boring, facing and threading

their outlets, valves, and joints is also shown in operation. This machine finishes unions as fast as the operator can put them in the chuck and an average of 3500 one-inch unions are turned out in a day of nine hours by this machine.

51 The Warner and Swasey exhibit will be acknowledged by all mechanical engineers as the principal factor in redeeming the machine tool display at the Exposition from being classed as a very ordinary exhibit of machine tools, and its high character will command the admiration of the engineer.

52 Another exhibit that has given character to the display of

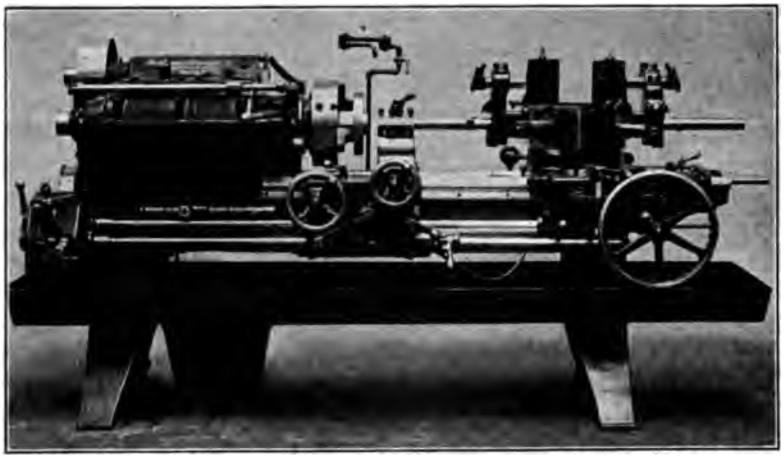


FIG. 11 UNIVERSAL HOLLOW HEXAGON TURRET LATHE EXHIBITED BY WARNER AND SWASEY CO.

machine tools at the Exposition is that of the Morton Manufacturing Co. This is an exhibit of draw-cut shapers, traveling head planers and key-seating machines. The draw-cut type has been developed by this company and lends itself to heavy cutting to a remarkable degree. The whole stress of cutting draws the work solidly and directly against the face of the main casting of the machine and eliminates the stresses from the table rail or the upper bolts holding the table as in the usual type of shaper. The tool arm or ram is under tensile stress in cutting, which tends to reduce or eliminate vibration; when the ram or arm is in compression, the heavier the cut the greater the danger of vibration. One feature of this type of shaper or planer is the facility it offers of shaping to lines, as the latter are on the outside in view of the operator and are not broken or destroyed by the tool as it leaves the work. The tool beginning its work on the face next the operator, the lines on that face can be worked to very accurately as they are not broken or destroyed until cut out by the tool itself. This feature is of great value to the operator as well as the owner of the tool. There is an adjustable back bearing which forms a stop or abutment to the end of the vice when planing parallel with the jaws. With this stop it is only necessary to clamp the work sufficient to hold it as the thrust of the cut comes against the back bearer or stop. As the drawing or pulling cut overcomes vibration, it is possible to make forming tools to be used in these machines at a moderate cost. Rounding tools, either concave or convex, can be made of various radii for producing correct curves in finishing parts of connecting rods, etc. Cutters can also be used for machining parallel openings cutting down both sides at once. Being the originators of the draw-cut principle as applied to this class of machines, the makers have consistently followed up every indication offered by their extended use leading to further improvement in design and operating function.

53 In the collective exhibit of Fred. Ward and Sons there are shown some very good machine tools, notably those by the American Tool Works. The engine lathes shown are well constructed modern tools with easily adjusted feeds and with speeds especially adapted and designed for electric drive. They are capable of caring for the service demanded by the use of high-speed tool steels and, as demonstrated at work, are good machines for modern manufacturing or general shop use. A planer and a shaper of good design and strongly built are also shown. The radial drilling machines shown by the same company are of good design and well built; they are also well balanced as shown by the absence of vibration when working at high speeds; the speed changes are readily made and the tapping attachment is very good.

54 Gould and Eberhardt exhibit a shaper of excellent design and workmanship adapted for heavy work. The arrangement for changing the length and position of stroke is very good. The motor that drives it has two speeds brought about by sliding the armature through the field of the motor which reduces the number of change wheels required in the gear box. The same firm shows a special gear cutter which cuts two blanks at the same time; this a strong, well built machine with several features that should be of interest.

55 In the extensive exhibit of products made by the Crane Co., is a motor-driven pipe-threading machine having two unusual at-

tachments—a quick centering rear chuck and compressed air cutting-off tools. The whole machine is well designed and strongly constructed and on account of the two unusual devices referred to is of

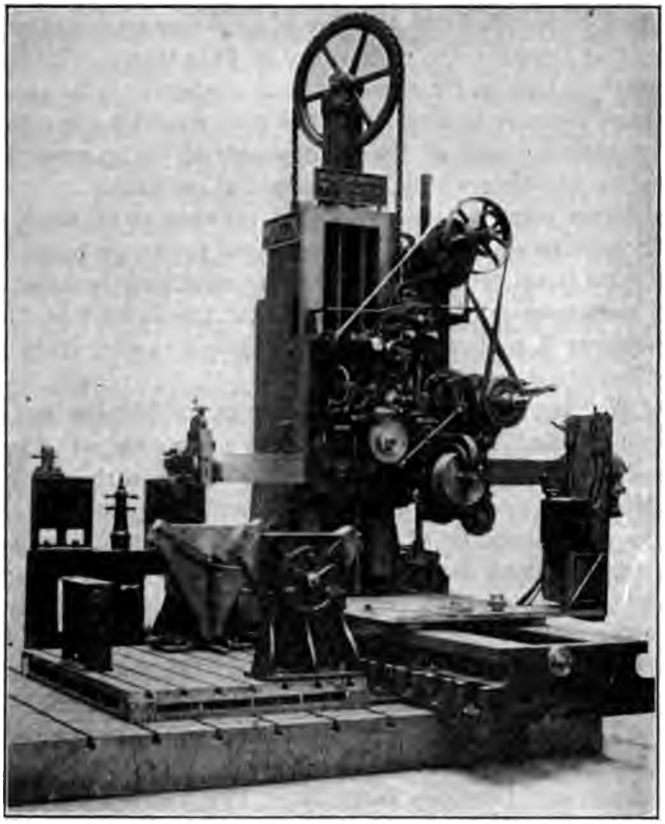


FIG. 12 CYLINDER PLANER EXHIBITED BY MORTON MANUFACTURING COMPANY

interest to those specializing in pipe work and fittings. This machine is made by the Crane Co. for its own establishments.

56 The Landis Tool Co. exhibits grinding machines of both universal and special application. The several machines are of high class design, massive in character and fitted to produce a wide range of work. The excellent quality of their work shows the care displayed in the building of the machines and the arrangement of the exhibit shows the wide and varied application of these machines in the mechanic arts.

57 There are a number of other metal cutting and shaping machines that attract attention in the Palace of Machinery and in the other buildings of this great Exposition; the engineer will discover them in places where he would never think of looking for such things, but on close inspection he will find that they are in some way connected with other things that belong in the place where he finds them. While the display of machine tools is, for reasons already explained, by no means complete, there is enough to dispel any anxiety in regard to the ability of the tool makers to meet all the requirements of the engineer for tools to produce anything within the compass of mechanics.

58 California being an extensive fruit growing state, much of that product must be shipped to other states and to foreign countries either in wooden boxes or in tin cans and the machinery required to make these containers is of great interest. At the Exposition, this class of machinery is well represented and merits a careful study by those interested in automatic machinery.

59 The William S. Doig Co. are pioneers in the manufacture of box making machines. Their exhibit contains a complete set of machines of wide range and application for labor saving in the making of boxes. The sides and ends of a box are nailed together in one machine; the proper number of nails the correct distance apart are fed from a nail box above the machine; the nails come down through tubes to their proper position where they take their places automatically in the nail driving device and are all driven by one stroke of the machine. A matter of four strokes drives the sides and ends together. The box is then passed to another machine and the bottom is nailed on all four sides at one stroke. Three men will nail up five thousand fruit boxes in one day on these machines. The mechanism is very ingenious and functions perfectly.

60 Another clever device is shown in this exhibit; it is a machine for fastening two or more pieces of wood together edge to edge. On this machine a number of reels contain corrugated steel ribbons sharp on one edge; if the pieces to be fastened together are long enough to require say four fastenings then there are four reels. The strips to be fastened together are drawn in at one end and discharged at the other. The corrugated ribbons are drawn in at one side at right angles to the seam and are cut off in lengths of about 1½ in. and driven into the wood across the seam making a solid joint. This machinery enables strips of any widths to be combined to make sides and ends, bottoms and tops of boxes, and saves much lumber that would otherwise be wasted. One machine with two boys will work up 4000 to

5000 feet of lumber in a day. The mechanical engineer will find much to interest him at this exhibit as the functions of the machine are readily followed.

61 At the George D. Parker exhibit two box making machines are shown, both new in design. One of these, the orange box machine, has now obtained a wide use in California. It is automatic and makes an orange box in four movements. First, the ends and middle blanks are pushed up into place; second, the slats forming the bottom are pushed forward from the back of the machine into place and nailed; third, the partly-formed box is turned on its side and the side slats pushed forward into place and nailed; fourth, the box is turned on the other side, the slats for that side pushed forward into place and nailed, and the finished box thrown out in front and the next movement brings up the ends and middle boards of the next box. Orange boxes come out of this machine at the rate of about 6 a minute, the whole process being automatic except the loading of the materials onto the tables. Another machine which has just been perfected after seven years' experimenting is shown. This completes a box at one stroke or revolution. The ends and sides are pushed up from below by a ram to the proper place while the bottom is pushed in from the back of the machine; the nailing mechanism acts simultaneously on the bottom and sides while a ram pushes the finished box out of the way of the material coming up for the next box. This universal machine is highly ingenious in design, entirely automatic in operation, of large capacity and has great economic value.

62 At the Max Ams Machine Co.'s exhibit is a complete set of can making machinery, arranged in the order usually occupied in a large can making establishment and being a complete outfit for making sanitary and open-top cans. Much ingenuity and good workmanship is displayed in this exhibit, but unfortunately the working out of the automatic operations has not been so coördinated in the display as to give great perfection of continuous operation. Still, these machines have a fine record on the Pacific Coast and a large proportion of the canning done here is in cans made by these machines.

63 In the E. W. Bliss Co.'s exhibit is a very complete operating exhibit of power metal working machinery illustrative of the high degree of perfection to which this class of machinery has attained. The sanitary can making equipment is almost entirely automatic. The design of the different presses and carriers shows great mechanical ability; the automatic actions are perfect in operation displaying fine construction and accurate workmanship. To this

equipment is attached an ingenious and effective can testing apparatus which charges each can with 40 lb. air pressure and passes it under water, immediately showing an air bubble if there is any leak; this testing is continuous and its speed equals that of the machine delivering the cans. The individual presses with semi-automatic attachments for the manufacture of various types of metal boxes and cans and the automatic threading machines show the same fine workmanship and precision of operation of the automatic working parts.

64 Also in the line of automatic machines, at the U. S. Army Ordnance Department exhibit is found a set of cartridge-making machinery in operation; this comprises a very interesting set of machines that show the result of the long process of perfecting required in machines for delicate operations. To such an extent has this perfecting been carried out that some of these machines are arranged to remind the operator of his omission should he neglect to do his part. This demonstration is very interesting and instructive.

65 As other instruments for cutting metals or other materials are coming into more extended use every day, the engineer will naturally look for some display of them and the work they do. He will find something worth while in this class at the exhibit of the Carborundum Co. in the Palace of Machinery. Here is shown an exceedingly high class exhibit of unusual educational value, extensive application of product, and instructive demonstration. The product of this company has revolutionized certain methods of manufacture, lowered costs and increased output. The application of the cutting wheels, cylinders, and discs extends from the cutting of the hardest crystals to the buffing of leather and cloth. This company has been the pioneer in this field, furnishing not only capital to establish a large business but also brains for original research. The engineer cannot fail to find this a most interesting exhibit.

66 Illustrative of cutting metals by a flame cutter, there are several exhibits of the oxy-acetylene cutting and welding equipments. The best exhibit is that of the Davis-Bournonville Co. in the Palace of Manufactures as it shows a complete oxy-acetylene equipment including the apparatus for producing oxygen. The generator for acetylene is of the latest design and the cutting torches are both hand and mechanically operated. The welding torches and gas regulators are especially worthy of the closest study and show ingenuity and a high development in this new art. The radiographs and oxygraphs are unique. The exhibit in general shows the high development

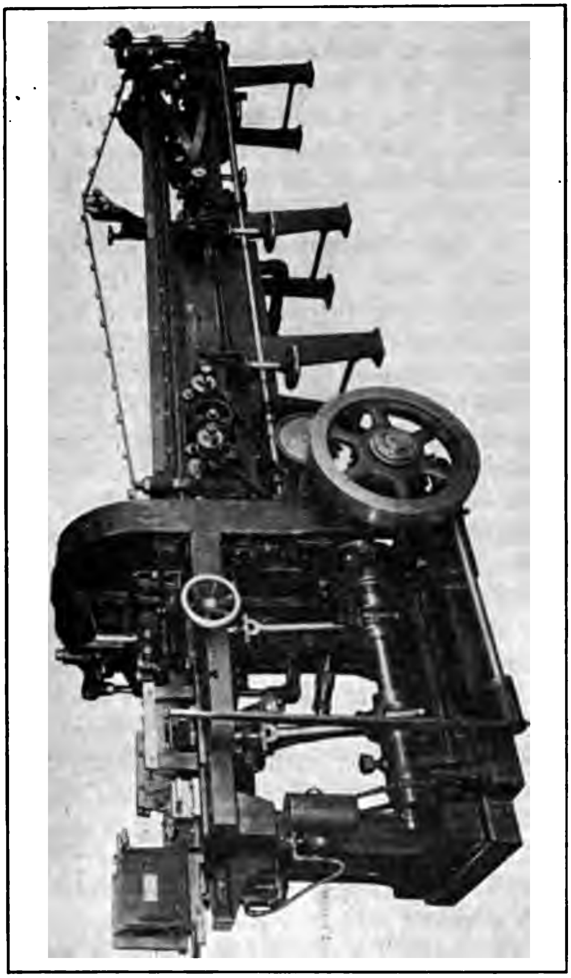


Fig. 13 E. W. Bliss Co.'s Automatic Can Body Maker. Capacity 150 Cans Per Minute

reached in the cutting and welding of metals by the oxy-acetylene flame.

67 The fact that the Pacific Coast is a good market for saws, especially wood cutting saws, brought some good exhibits of that class of cutting tool to the Exposition, the most notable being the exhibit of Henry Disston and Sons, Inc. This firm shows a splendid collection of saws both for wood and metal cutting. The new forms of metal cutting saws with inserted teeth show the most advanced practice, the teeth being made of the newest grade of tool steel for high speed cutting in hard metals. To a mechanical engineer seeking new developments in cutting tools this exhibit is of great interest. A very instructive moving picture exhibit, illustrating the methods of manufacture in their establishment and the quality and variety of their products, is shown by this company in the Pennsylvania Building. The many and varying grades of steel necessary for their productions are manufactured by the company itself and this specialization has undoubtedly been instrumental in building up an immense world-wide trade in Disston saws. It may be of interest to note that the Disston company has worked out and put in practical operation many interesting schemes for the welfare of its workmen.

68 As he works his way through the Palace of Machinery, the mechanical engineer, if he be at all interested in the shaping of sheet metal, will stand for a while at the exhibit of the Dreis and Krump Manufacturing Co., who show a line of very well thought out brakes for bending metal plates. Why these machines should be called "brakes" when their function is to bend not to break the materials they handle is a mystery to the writer. Three machines are shown, two being for light work, such as cornices or pilasters, adapted to all classes of straight line work, producing sheet metal boxes with right angle corners with great rapidity, and one for heavy steel plate, powerful in character and capable of bending to sharp angles plates $3/8$ in. thick and 12 ft. in length; this latter is a very original machine with well designed and strongly constructed steel working parts. This exhibit marks a decided advance in sheet metal working machines.

69 The Geometric Tool Co. in the Palace of Varied Industries exhibit tools for cutting external and internal threads, interesting on account of the fine and accurate workmanship and ingenious design displayed. The head for external threads opens automatically as soon as the length of thread for which it is set has been cut. For internal threads, the taps automatically collapse at the points for which they have been set. In each case, when the thread is

finished the chasers do not return over the threads on the reversing of the tool. These tools can be used in any screw machine or turret lathe and show a decided advance in tools of everyday use.

70 To the engineer interested in hydraulic press work, the exhibit of the Hydraulic Press Manufacturing Co. is especially interesting. This is an extensive display of presses including pumps, valves, etc., for the manufacture of cider, olive oil, etc., and for filtering the same. There are shown also presses for bending or straightening metal bars, pressing car wheels on to axles and other similar lines of work.

71 Among small tools are those shown by the Henry G. Thompson and Sons Co. in the Palace of Manufactures, whose exhibit consists of an interesting line of hack, power, and jig saws for cutting metals, and also some very clever tool holders. All the machines are shown in operation. The power hack saw is fitted with an ingenious pump for handling the lubricating compound and guide rollers for holding the saw blade in line. These improvements are a decided advance. The jig saw for cutting irregular contours in metal is an excellent machine. The saw blade at least 50 ft. in length is wound on a reel like a measuring tape; the free end does the cutting being operated from above; when the part cutting is used up, it is broken off and a new part brought up from the reel. The tool holders shown are novel, every variety of tool for one machine being carried by one holder.

72 The requirements of modern civilization have opened up many and varied fields for the engineer to cultivate for the good of humanity; this is very forcefully illustrated in the matter of food preservation and today there are large establishments devoted to the production of refrigerating machinery. There are four exhibits of refrigerating machinery at the Exposition that merit attention. Two of these are in the Palace of Food Products, one in the Horticultural Palace, and one in the Manufactures building.

73 The Larsen Ice Machine Co. have a very fine and interesting exhibit consisting of a complete plant for the freezing, storing, and hardening of ice cream, the daily capacity of the plant shown being 400 gal. This exhibit as a whole is of a very high character. The freezer cylinders and agitator are of German silver made without a seam, insuring perfect cleanliness; the arrangement and insulation of the hardening chambers are also well worthy the attention of anyone interested in this class of machinery.

74 Near this exhibit is that of the York Manufacturing Co., which is an exhibit of a complete refrigerating plant in operation,

keeping cold several chambers throughout the building. This company is one of the largest makers of this class of machinery in the United States and on that account the writer expected a finer display of their product.. What is shown, however, is good.

75 The exhibit of a complete refrigerating plant by the Vulcan Iron Works will be found very interesting. The refrigerating chambers are a part of the exhibit and, being of glass, show the splendid

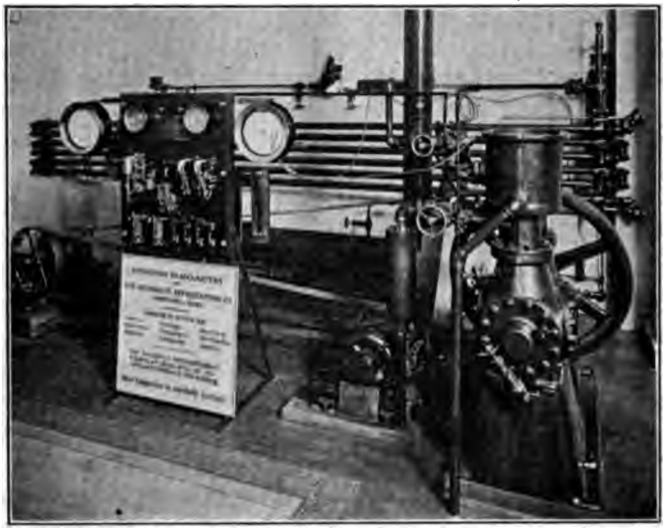


Fig. 14 Refrigerating Plant Exhibited by The Automatic Refrigerating Co.

condition of the foods inside, there being no sign of dampness either on the inside of the glass insulation or on the contained meats, butter, cheese, etc. This exhibit is a first-class illustration of the whole refrigeration process and affords valuable information on this subject.

76 The remaining exhibit of refrigerating machinery is by the Automatic Refrigerating Co. This is the only complete, automatically controlled refrigerating plant on exhibition. The automatic control is a decided step in advance on new ground; it automatically stops the machine when the temperature in the refrigerated chamber reaches a point one degree below the temperature for which the machine is set and starts it again when the temperature is one degree above that point. This automatic regulation also applies to any number of chambers working at different temperatures. The cham-

bers on which this machine is operating are used in connection with fruit exhibits, and during certain parts of the day they are opened very frequently. Cards taken from the pressure recorder on the ammonia pump show this very well and it can be noted that during the cooler part of the night and when the chambers are not opened the machinery runs about 4 min. and stands 15 min. very regularly.

77 There are many other engineering exhibits in the Palaces of Machinery, Transportation, and Manufactures. Among those worthy of mention are the Dodge Manufacturing Co.'s exhibit of general power transmission machinery and accessories, which includes a good example of the Dodge system of rope drives. There is also the Zimer vibrating screen, an ingenious combination of reciprocating and tossing action by which the material is screened and conveyed at the same time, designed especially for moving and segregating hot ores, bolts, rivets, ashes, slag, etc.

78 The mechanical engineer depends so much for the success of his machinery on the quality of the lubricant he uses that the oil question is always an interesting one. The exhibit of lubricating oils made by the Union Oil Co. of California shows real achievement in petroleum technology and the company deserves great credit for its scientific development of lubricants from California crude oils. As petroleum specialists, they have succeeded in developing oils suitable for all classes of lubrication. For many years the crude oils that do not contain asphaltum have been considered necessary to meet these conditions. The exhibit shows that through scientific development the Union Oil Co. has contributed much to the knowledge available on the production of lubricating oils from crude petroleum.

79 What has been stated in regard to the above-mentioned exhibit applies equally to that of the Standard Oil Co., which has shown great skill in the treatment of oils for the many and varied uses to which its product is applied. Beyond simply covering the field of lubrication, the California oils show certain superiority that merits special recognition. This refers largely to the low cold tests of heavy lubricating oils.

80 To the engineer interested in the distribution systems of water works, the exhibit of the A. P. Smith Manufacturing Co., is one of great interest; this is a collection of water gate valves, fire hydrants, water works machines and appliances. The most interesting feature is the exhibit of appliances used for placing a gate valve into a line

of main pipe while the latter is under pressure. This company is one of the oldest and best known in the manufacture of these specialties and their display is both instructive and attractive. In particular the tapping and valve inserting machines show great ingenuity.

81 The economical handling of materials in the shop is illustrated in the working exhibit of the Shephard Electric Crane and Hoist Co. This consists of electric power cranes and hoists for use from one or two overhead supports; also an electric winch, monorail track, and electric controllers and switches especially adapted for hoisting apparatus. The cranes and hoists are adapted especially for safe, convenient and rapid operation about shops, foundries, mills and warehouses. The mechanism is well protected against dust and grit. The monorail consists of two hard steel T-rails bolted to the lower flange of a standard I-beam without mutilating it; the strength is thus increased and wear on the supporting beam avoided.

82 The General Electric Co. shows a complete self-propelled electric truck on which is mounted an electric crane; this is demonstrated in operation and is an attractive and instructive exhibit.

83 The correct measurement of the quantities of liquids during their transmission through pipes has always been an interesting problem, and at the Exposition are two exhibits, that of the Neptune Meter Co. and that of the National Meter Co., illustrating apparatus for this purpose.

84 The Neptune Meter Co. shows a very complete and well arranged collection of various types of meters designed to measure and record different liquids in various units and quantities. They all show good design and excellent workmanship. Of special interest are the various sizes and valuable modifications of the old and well known disc type of meter and a meter known as the "Victor," of the oscillating piston type. A meter of the turbine type known as the "Crest" should be noted; it has two turbine wheels on the same shaft with the helical blades inclined in opposite directions, and is designed to measure large flows with slight loss of head and range in size from 1½ in. to 20 in. diameter. There are also shown various forms of compound meters, in which a meter of the disc type is combined with a meter of the turbine type. These are all arranged so that when either meter is in operation the other is shut off by a valve. They are designed to measure accurately flows where the variation of inflow is very great, the small flows being measured by the disc meter and the large by the turbine meter.

85 A special form of the compound meter known as the "Protectus" is shown; this is of new design and contains in combination a disc meter, a turbine meter, and a "Control Orifice Tube" with an automatic check valve at the delivery end of the tube. This valve is operated by excess of pressure in the tube when the flow from the discharge pipe of the compound meter exceeds 50 per cent of the capacity of the disc meter. The opening of the check valve acts to mechanically close the outlet valve of the disc meter and stop its action and open wide the passage through the orifice tube and the turbine meter. The turbine meter is proportioned to measure 25 per cent of the flow through the orifice tube and to register the combined flow of itself and the orifice tube. A special adaptation of the disc meter is shown in an apparatus for measuring and recording a predetermined quantity of liquid and automatically stopping the meter and the flow when that quantity has passed the meter.

86 The National Meter Co. shows a large variety of types and sizes of meters for measuring and recording the flow of liquids. One with the trade name of "Crown" has been in use since 1879 and is still largely used for muddy water. Another with the trade name "Nash" of the nutating disc type is the most extensively manufactured; the disc is conical and has an unconstrained rolling movement in its conical seat and this permits small, solid particles to pass through without damage to the meter. Type K is a very highly developed disc type, in which all working parts may be removed without removing the body from the pipe line and without removing bolts. Damage by freezing is prevented by simple cast-iron washers under the bolt nuts which break under undue strain. A meter of the turbine type named "Gem" is shown; this is adapted only to the measurement of large and rapid flows and is used either alone or in combination with one of the oscillating piston type called "Empire," to form the "Empire Compound." In this arrangement the passage through the "Empire" is always open while that through the "Gem" is controlled by an automatic differential check valve which opens under the difference of pressure produced by a large flow through the "Empire" meter. This combination is said to have the accuracy of the "Empire" for small flows and the freedom and accuracy of the "Gem" for large flows. Of special interest is an apparatus having the name of "Premier" to be placed in 30-in. water mains. It consists of one large and one small venturi tube in parallel relation. These tubes are so proportioned that the combined flow is 105 times that through the

small tube and the flow through the small tube is measured and recorded by an "Empire" meter.

87 Both these exhibits of meters are of great interest to the engineering profession, which is sufficient excuse for giving a somewhat lengthy description of them.

88 At the Exposition there is a large amount of pumping machinery for all the varied purposes for which pumps are used. While much of this is of the ordinary type which can hardly expect much attention, there are, however, several exhibits in this class that the engineer will find interesting and a short notice of some of them will not be out of place.

89 The Luitweiler Pumping Engine Co. has an exhibit consisting of Luitweiler pumps of two different types adapted to surface, deep well and hydraulic service. The special feature is the method of driving. In place of the usual crank shaft operating through connecting rods there are eccentric or heart-shaped cams with the face of the cam working on rollers to produce the motion for the pump ram or bracket. The motion transmitted to two or more pump rods may be so arranged as to give a non-pulsating constant discharge of water or other liquid. The surfaces of the cams are hardened as well as the rollers; they are both generous in size and little wear need take place.

90 The Layne and Bowler Corporation exhibits a turbine pump especially adapted to deep wells. Its special feature lies in the pump chamber being suspended from its upper end, as are the rotating parts. The shaft rotates inside a central pipe in which it has bearings at intervals, lubricated with oil or clear water injected at the top of the pipe; the shaft bearings are thus protected from contact with dirty water. The rotating parts are carried on roller bearings at the upper end of the shaft, and for heavy pressures these are supplemented by oil pressure applied between one or more pairs of discs. Each rotating disc fixed to the shaft is enclosed within a close fitting bronze ring which has free lateral movement. This is a well-designed pump marking progress in a class of machinery that has been in a stationary condition for some time past.

91 The Krogh Manufacturing Company exhibits a variety of centrifugal, turbine, and plunger pumps for various uses which are of good design. Among these are a vertical centrifugal mine sinking pump, cornish pumps, jack head pumps, multi-stage mine station pumps, single-stage motor driven horizontal centrifugal

pumps, multi-stage high pressure turbine pumps, and a long double-suction centrifugal pump with a capacity of 30,000 gal. per minute. The engineer finds several items for his note book in this exhibit.

92 The American Well Works has an exhibit consisting of a large variety of centrifugal, turbine, and deep well plunger pumps, of good design and well worthy of attention.

93 Closely connected with pumps are pipes, valves, and fittings. These are found in great variety and of all dimensions in the Crane Co.'s exhibit which fills a prominent place in the Palace of Machinery. Some of the main features of this exhibit are: One 72-in. wedge water gate in operation worked by a hydraulic lift and weighing 56,000 lb.; one 36-in. wedge water gate in operation worked by a motor; a complete line of steel water gates from 2 in. to 18 in.; one 12-in. high pressure water gate for 800 lb. pressure; a complete line of high pressure check valves for 1000 lb. pressure; a large assortment of water gate valves with and without indicators. This exhibit contains many more of the company's products than can be mentioned here and is one of the most notable features of the mechanical department.

94 There are many other exhibits that should have been mentioned if space permitted. In the Palace of Mines there is much that is interesting and new. This branch of engineering coupled with the advance in chemistry and metallurgy has made great progress in recent years and much of that progress can be traced in the exhibits in this building.

95 In the Palace of Transportation, the progress in that field of engineering can be traced through all its stages. In one corner is found an interesting old pioneer with the wagon in which he crossed the continent 65 years ago and from that point of beginning can be followed the progress in transportation up to the present day, when 60 miles an hour can be made on our improved highways with almost the comfort of a railroad coach. From the great exhibit of the Westinghouse Co. in this palace as a center, wherever one turns new wonders of mechanical genius continually arrest attention, and the skill that has conquered the land, the sea, and the air, all help to raise the engineer's estimate of the worth and dignity of the profession to which he belongs.

Standing

B. C. Bryan · F. R. Low · J. Hunter · C. A. McAllister · F. J. Frank · D. S. Watkins · W. H. Crosby · C. P. Poole · C. Hering · H. W. Bringhurst · T. Morrin · W. K. Kavanaugh · J. M. Smith · N. A. Bowers

Sitting

H. W. Hibbard · C. E. Lucke · J. T. Faig · J. C. Mengel · G. W. Dickie · G. W. Danforth · W. H. Onken, Jr. · C. M. Jansky · G. M. Brill · E. S. Fischer · C. W. Rice

Absent: A. Gianini, W. Meredith, C. F. Pemberton

FIG. 12 INTERNATIONAL JURY OF AWARDS FOR DEPARTMENT OF MACHINE EXHIBITS
Panama-Pacific International Exposition, 1915

No. 1492

THE DIESEL ENGINE AND ITS APPLICATIONS IN SOUTHERN CALIFORNIA

By Walter Holbrook Adams, Pasadena, Cal.

Member of the Society

Diesel secured his first patents in 1893, and brought out his first successful engine in 1897, at the Augsburg Works in Germany, and since the latter date the use of the Diesel engine has been increasing steadily, especially in Europe. Several well-known steam engine manufacturers in the United States today have begun the manufacture of Diesel engines, thus showing a growing demand in this country for such a prime mover. There are comparatively few Diesel engines in the United States at present, the total horsepower in use being just over 100,000, but the number is increasing rapidly every month.

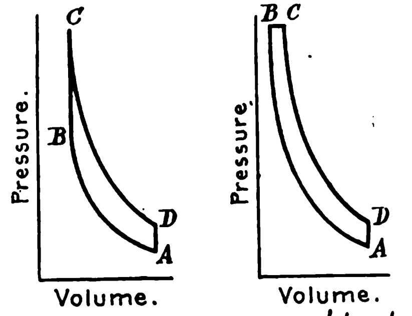
2 Diesel's original patent described the action of his engine as follows: (*a*) The highest temperature is that due to the compression of air only and this may be regulated by making the compression the desired amount. (*b*) Into this air is introduced the fuel, gradually, in a finely-divided state and in such quantity that the burning offsets the cooling due to the expansion as the piston moves forward.

3 This was the original idea of Diesel, namely, a supply of heat at constant temperature. Such a supply would fulfil one of our thermodynamic conditions for maximum efficiency—a supply of heat at a constant maximum temperature. Diesel's engine in practice did not give this desired result, so he modified his statement to cover an increase in temperature during the admission of the fuel, this increase in temperature taking place at constant pressure. This is the condition of the Diesel engine today, as closely as it is possible for the actual engine to meet the ideal conditions.

4 The difference between the Diesel engine cycle and that of

Presented at the Panama-Pacific International Exposition Meeting, San Francisco, September 1915, of The American Society of Mechanical Engineers.

the standard form of internal combustion engine (Otto) is shown in Fig. 1. In the Otto cycle there is compression from A to B; ignition and burning at constant volume from B to C; expansion from C to D; and rejection of heat to the exhaust, at constant volume, from D to A. (It makes no difference in the ideal diagram whether the engine is 2- or 4-cycle.) In the Diesel engine there is corresponding compression from A to B; then burning at constant pressure from B to C; expansion from C to D and exhaust at constant volume from D to A. In the Otto cycle there is an explosion while the volume remains constant, thus increasing the pressure and temperature; in



$$E = 1 - \left(\frac{V_B}{V_A}\right)^{k-1} \qquad E = 1 - \frac{\left(\frac{V_B}{V_A}\right)^{k-1}\left(\left(\frac{V_C}{V_B}\right)^{k} - 1\right)}{k\left(\frac{V_C}{V_B} - 1\right)}$$

OTTO DIESEL

Fig. 1 Comparison of Cycles and Thermal Efficiencies

the Diesel cycle there is burning at constant pressure while the volume and temperature increase.

5 The expressions for the thermal efficiency of the ideal cycles are also shown in Fig. 1, and curves plotted from these equations are given in Fig. 2. The most interesting thing to observe from these curves is that, for corresponding pressures at the end of compression, the Diesel engine has the lower thermal efficiency. This is offset by the fact that in the Otto engine the limit of compression pressures is 80 to 200 lb. per sq. in., while in the Diesel engine the compression may be carried as high as desired. The reasons for this are that in the Otto engine the fuel is compressed with the air, and pre-ignition will take place if the compression is carried too high, due to increase of temperature with increase of pressure. In the Diesel en-

gine, air only is compressed and the temperature may rise as high as desired without danger of pre-ignition. The temperatures due to compression are shown in Fig. 3, calculated for an adiabatic compression. At a pressure of 500 to 550 lb. per sq. in. the temperature is about 1000 deg. fahr., and if fuel in a finely-divided condition is introduced into air at this temperature, it will take fire and burn without any special ignition apparatus.

6 High compression is possible with the Diesel engine with

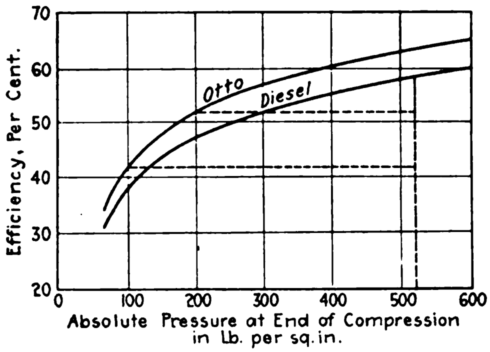


FIG. 2 COMPARISON OF EFFICIENCIES, OTTO AND DIESEL ENGINES

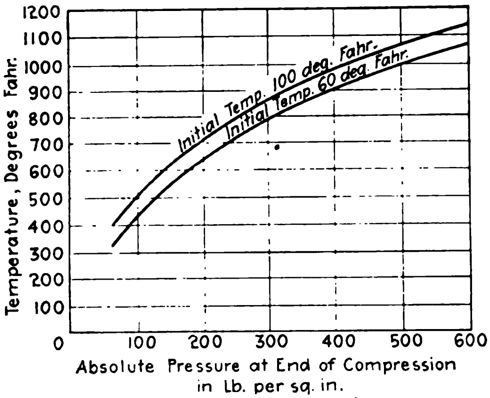


FIG. 3 DIESEL ENGINE COMPRESSION TEMPERATURES

corresponding gain in efficiency, and at the same time there is more complete combustion of the fuel because of the finely-divided state in which it is forced into the cylinder. The maximum possible efficiency for the Otto engine is 52 per cent for blast furnace gas and 44 per cent for a motor car engine. Against this there is 57 per cent efficiency for the Diesel engine, a gain of 29 per cent over the motor car engine and 11 per cent over the blast furnace gas engine. If the Otto engine could have the same compression, it would be a better engine that the Diesel engine.

7 This question may be asked: Why cannot the efficiency of the Diesel engine be further increased by higher compression than 500 to 550 lb. per sq. in.? This is possible in the ideal engine, but experience has shown that 600 lb. per sq. in. is the highest allowable pressure in the actual engine. For pressures above that amount, the increased size of the parts of the engine increases friction and cost above any gain in theoretical efficiency. Changes in the Diesel engine will have to be more along the line of reduced cost and simpler design rather than better economy.

TWO-CYCLE AND FOUR-CYCLE INTERNAL COMBUSTON ENGINES

8 The difference between the 2-cycle and 4-cycle types of internal combustion engine is in design and construction and not in theory.

9 In the 2-cycle engine there is expansion of the burnt gases until near the end of the stroke; then exhaust begins and almost simultaneously there is admission of either air and fuel or air alone at another part of the cylinder—this admission taking place under slight pressure. The burnt gases are forced out by the fresh charge, the piston moves back and compression takes place. This gives one working stroke for every revolution, just as in a single-acting steam engine. The necessary compression of the inlet air is supplied by the slight compression produced by the piston in an enclosed crank case (cheap engines) or is produced by a separate scavenging air compressor, as in all 2-cycle Diesel engines. In this type the pressure of the air is not carried to more than 4 to 8 lb. per sq. in. above atmospheric.

10 In the 4-cycle type there are ignition and expansion until the end of the stroke. Then the exhaust valve opens and remains open until the piston has moved back, thus allowing the piston to expel the burnt gases. The exhaust valve closes and the admission valve opens, remaining open while the piston moves forward to draw in a fresh charge. It then closes and compression takes place. This complete cycle requires two revolutions of the engine and in practice necessitates a large flywheel or a multiple-cylinder engine if the speed is to be kept constant.

11 There are advantages and disadvantages for both types applied to Diesel engines. The 2-cycle type gives almost twice as much

power for the same size of cylinder, as it has two working strokes for one in the 4-cycle. (Actual value is 170 to 180 per cent.) This means less weight, space and first cost. As usually constructed, the piston acts as its own valve and so air inlet and exhaust valves are not required. (This is not true of some of the better class of 2-cycle Diesel engines, as will be explained later.) In marine work the reduction in number of valves makes it easier to reverse a 2-cycle engine. The use of the 2-cycle type has also made large units possible, and 1200 h.p. per cylinder in a single-acting engine has been built.

12 On the other hand there is to be said for the 4-cycle type of Diesel engine:

a It is older than the 2-cycle type and so has become a more stable construction

b It gives better fuel economy, as expansion can be carried to the end of the stroke and no power is required for the scavenging pump. The gain is about 10 per cent

c The mean temperature is lower. There is more time to remove the heat and not so much heat to remove per unit of cylinder surface. (In a 2-cycle engine 90,000 B.t.u. per hour has to be removed for every square foot of cylinder surface. In 4-cycle engines the figure is 40,000 B.t.u. In an ordinary water-tube boiler working at 300 per cent of rating, it is 10,000 B.t.u.)

d The valve gear runs at one-half the speed of the main shaft

e In the high speed 2-cycle engine, it has been difficult to get the burnt gases out of the cylinder in the short time available, so that such engines have not been quite as successful as 4-cycle engines.

13 The tendency in this country and abroad is to use 4-cycle engines up to from 700 to 1000 h.p. and above that 2-cycle. This is due to the reduced first cost of the 2-cycle type in the large sizes and the excessive diameter of cylinder required in large 4-cycle engines. As progress is made in design, the 2-cycle type may supersede the 4-cycle, but this is not evident at present in the smaller sizes.

APPLICATIONS OF THE DIESEL ENGINE

14 The Diesel engine is in use today in almost all places where a steam engine or turbine might be used. Its starting torque is poor and it should run at a constant speed, although this may be varied to some extent. The rated load decreases as the altitude at which the engine operates is increased.

15 The engine is being employed for propelling ships of over 400 ft. in length and 9000 tons in cargo capacity, at a speed of about 11½ knots; such ships are twin-screw and have engines of 1600 b.h.p., developed in 6 cylinders. The engine has not been used for high speed passenger ships. It has been used in many sailing vessels to provide auxiliary power in calm weather. It has been used in submarines, and the Craig Shipbuilding Company is now building for the United States Government some submarines which are to be equipped with Busch-Sulzer Diesel engines.

16 One locomotive equipped with Diesel engines has been built in Europe, but it was large, clumsy and not very successful. Diesel locomotives will not compete with steam locomotives at present. For isolated plant or central station service the Diesel engine is well adapted, if several units are installed so that each unit will work near its rated load without a heavy overload under all conditions of load factor.

17 The Diesel engine gives excellent service when installed as a pumping engine. It can be used for all kinds of factory service just as well as the steam engine. The reasons why it is not adopted more extensively in this country are probably:

1 The availability of cheap fuel has prevented a demand for an expensive first cost prime mover that will give decreased operating cost
2 American manufacturers have been slow to take up the manufacture and introduction of these engines
3 Engines giving satisfactory service have only been made in Europe within the last five years
4 A Diesel engine requires extreme care in manufacture and in adjustment, particularly of the fuel valve
5 The engine cannot be operated without careful supervision when the cleaning and adjusting are going on
6 Some prejudice exists against all forms of internal combustion engines due to the multiplicity of causes that may prevent their starting
7 Oil must be used as fuel, and the cost may vary within wide limits
8 Innate conservativeness of the human race makes it slow to adopt a new method or machine until others have tried it.

DESIGN CHARACTERISTICS OF DIESEL ENGINES

18 It is my intention to discuss design characteristics in a gen-

eral way, summing up the present situation, rather than to describe details of the various types. (An excellent article[1] on the design of the Diesel engine has been published in The Journal.)

19 At present engines are manufactured in this country in both the horizontal and vertical types. One to four cylinders are used in the horizontal and one to six cylinders in the vertical engines. The favorite size of the latter is two to four cylinders.

20 The horsepower per cylinder ranges from 30 to 250, with size of cylinder varying from 12 to 21 in. The stroke bore ratio is about 1.25. (In December one manufacturer announced a 4-cylinder vertical engine of 2500 b.h.p.)

21 The smallest Diesel engine (dimensions of cylinder) that the author finds any record of is a 6¾ by 8⅝ in., 2-cycle, 4-cylinder engine, developing 110 b.h.p. at 550 r.p.m. The largest engine is a 32.2 by 39.4 in., 2-cycle, single cylinder engine, developing 1250 i.h.p. at 150 r.p.m., m.e.p. 106 lb. per sq. in. This latter is an experimental engine built by Carels Bros. in Belgium. If such an engine were built with 8 cylinders, it would have an output of 10,000 b.h.p., which would compare favorably with a steam turbine, if the space occupied be neglected.

22 The engines built in the United States are all comparatively slow speed, ranging from 150 to 300 r.p.m., with piston speed of 600 to 900 ft. per minute. A few high speed marine engines have speeds as high as 480 r.p.m. In Europe, before the war started, there was being developed a line of high speed engines with a speed of 550 r.p.m. for submarines and such work. The highest commercial speed in use at present is 350 to 400 r.p.m.

23 All engines, except one make, are single-acting. (One manufacturer reports the making of a double-acting engine but I should judge that this has not been tried out thoroughly.) In Europe several firms were experimenting on a double-acting engine; the trouble experienced in this type is in cooling the cylinder and piston and keeping the stuffing boxes tight.

24 All engines in this country, except two makes, embody trunk pistons without crossheads. In Europe several marine types use a crosshead, but all others employ the trunk piston. The crosshead takes the wear produced by the angular thrust of the connecting rod.

25 *Valves and Valve Gear.* There is necessarily considerable

[1] Recent Developments in the Manufacture of the Diesel Engine, H. R. Setz, Journal Am. Soc. M. E., Vol. 36, December, 1914, page 420.

difference in the design of valves and gear of the 2-cycle and 4-cycle types. As the piston acts as its own valve in the high speed 2-cycle type, there are only two valves in the cylinder head—the fuel valve and air starting valve. In the better slow speed 2-cycle engines there are 7 valves in the cylinder head; these are 4 scavenging valves, 1 fuel valve, 1 starting air valve and 1 safety valve. There are no American engines with 4 scavenging valves in the cylinder head.

26 The 4-cycle engine must have at least 4 valves, namely, suction, exhaust, fuel and starting valves. The safety valve is often combined with the air starting valve. In most engines the valves are all placed in the head of the cylinder, but in a few types the suction and exhaust valves are placed on its circumference.

27 The material used for cylinder walls, liners, and heads is cast iron. A few engines have been equipped with cast steel heads, but these did not prove satisfactory and cast iron has been substituted.

28 *Fuel Valve.* The fuel valve in the Diesel engine is the most delicate part of it. Even when the engine is fully loaded, this valve moves only a few hundredths of an inch. In an engine with a 21 by 30 in. cylinder, the amount of oil per stroke at rated load is only 0.4 cu. in. When it is remembered that this oil must be introduced in the form of a very fine spray in a time of about 1/40 sec., and that the regulation of the amount of oil is the only method of governing the engine, it is easy to imagine the troubles of the early operators and designers. The fuel valve must not clog or fill with gum and must always operate correctly. The designs of the manufacturers vary in detail, but the underlying principle is the same. The fuel is pumped into a chamber surrounding the valve by a pump whose stroke is controlled by the governor; this chamber forms a labyrinth passage. Air under a pressure of 800 to 1000 lb. per sq. in. is admitted in back of the oil and forces the latter into the cylinder in the form of a fine spray. The valve is controlled by a cam, opening about 1 per cent before the end of the compression stroke and remaining open from about 8 to 10 per cent of the working stroke.

29 In most of the present engines, the oil is forced into the valve passages against the air pressure, thus requiring a strong oil pump. Several manufacturers have adopted the method of pumping the oil into a restricted passage between the air valve and the cylinder during the suction stroke of the engine, where it remains until the air valve opens and it is forced into the cylinder. This arrangement reduces the work of pumping. The relative merits of the two types are under

discussion today. So far as the author can find out the low pressure type seems to give the best satisfaction with California oils. At least the manufacturers using this type say they can use any grade of oil, while the manufacturers of the high pressure type like to specify a minimum grade of oil that they can use.

30 The remainder of the design of the Diesel engine follows gas engine design quite closely, with generally more massive and careful construction. It is discussed fully in the paper quoted above.

31 *Air Compressor.* The air used for spraying the oil into the cylinder is supplied by a 2- or 3-stage air compressor. The pressure required for the spray is 800 to 1100 lb. per sq. in., depending mainly upon the kind of oil used, but also to some extent on the load under which the engine is working. The amount of this air is estimated to be from 16 to 34 cu. ft. of free air per b.h.p. per hour. The power required for operating the compressor is about 4 to 7 per cent of the total power developed by the engine. The compressor is usually made an integral part of the engine, and is driven by a crank forged on the crank shaft, or in a few cases by a belt from the engine or by a motor.

32 *Scavenging Pump.* In the 2-cycle engines a special scavenging pump is used for driving the burnt gases out of the cylinder. This is usually made the low pressure stage of the air compressor. As previously mentioned, the scavenging air is controlled by the piston or by scavenging valves in the cylinder head. This latter type gives the best scavenging, as the air sweeps through the cylinder from the head end, and passes out through ports placed around the circumference at the crank end. There is a large gain in economy and power due to using this scavenging air in the 2-cycle engine, because of better combustion, but it is not necessary in the 4-cycle engine.

33 The air is supplied at a pressure of 4 to 8 lb. per sq. in. above atmosphere. The volume of this air is 1.2 to 1.8 the cylinder volume. The power required for the pump is about 4 per cent of the output of the engine.

34 *Governing.* The governing is by a governor which regulates the amount of fuel supplied to the engine. The governor holds the suction valve of the fuel pump open for a portion of the forcing stroke or regulates the length of the stroke or varies the clearance of the pump. Each manufacturer employs a different method for governing, but all methods in use seem to give a close regulation. If the engine has more than one cylinder, each cylinder must have its own pump and all pumps must be under the control of the

governor. As far as the author knows, no one has attempted to distribute the oil to the various cylinders after it leaves a common fuel pump.

35 The regulation is well under 3 per cent in all types, and, if necessary, some manufacturers are willing to guarantee much closer regulation. All engines, except the single cylinder types, will give close enough regulation for operation of all electrical machinery. The overload capacity of the Diesel engine is small when compared to turbines as it is only about 10 to 15 per cent.

36 *Water Cooling.* The cylinders and cylinder heads of all Diesel engines must be watercooled, and in the larger sizes the pistons must be cooled also. The amount of water required is about 3 to 9 gal. per b.h.p. per hour, depending upon the temperature rise which is allowed. With a temperature rise of 70 deg. fahr. the amount of water will be about 3 to 4 gallons. The maximum temperature of the cooling water is kept about 130 to 140 deg. fahr., although it may rise to as high as 180 deg. fahr. if the water contains no impurities that will precipitate at this temperature. The heat carried away in the cooling water is about 2500 to 3000 b.t.u. per b.h.p. per hour.

FUELS

37 Any fuel that will burn without leaving an ash or residue, either due to incomplete combustion or due to unburnable material in the fuel, may be used in a Diesel engine. Attempts have been made to introduced pulverized coal into the cylinder of the engine but these have not as yet been successful. Gasoline, kerosene and the light distillates need not be considered as fuel for the Diesel engine as they can be used to better advantage elsewhere.

38 There is left crude oil, low-grade distillates and the coal tar products. The last have not been used to any great extent in this country.

39 Crude oil which is free from sand and water can be used as fuel, even if it contains as much as 50 per cent asphaltum. Owing to the scarcity of gasoline, today practically all crude oil has the gasoline content removed before it is sold for fuel, so that all fuel oil is "topped" oil.

40 On the Pacific Coast we are not interested in the Eastern oils, but I would like to say that when used in Diesel engines these have given better satisfaction than the Western oils. The California oils have been tried on the manufacturers' testing floors in the Eastern states, and all reports that the author has received indicate that they

have been satisfactory. But, at the same time, the statement is made that the tests have not been continued for more than 6 to 7 days as the supply of the special oil becomes exhausted. After such tests the condition of the engine is always reported to be excellent. The objections that the author has heard against the Western oils are these: Viscous and sluggish, high sulphur content, high water content, and high in ash.

41 A viscous and sluggish oil can be heated by the cooling water as it leaves the engine, or by the hot exhaust gases, until it becomes fluid. It can then be pumped as well as any oil. When such an oil is employed kerosene should be used to start up and for a few minutes before shutting down, so as to clean all the heavy oil from the piping and pumps.

42 The high sulphur content oil is more dangerous, as it burns to sulphur dioxide which tends to cause corrosion of the piston and cylinder, the valves and valve seats and the exhaust pipe. The maximum amount of sulphur that can be allowed seems to be about 2 to 4 per cent.

43 Water in the oil will decrease the heating value and cut down the amount of fuel delivered to the cylinder. If the water comes in "slugs" it will cause the engine to run irregularly. "Topped" oil will not contain much water, as it will be removed during the topping process. Most manufacturers specify an oil containing less than 1 to 2 per cent of water.

44 The ash is of considerable importance as it tends to remain in the cylinder, causing cutting of the walls, the valves and the valve seats. This makes the maintenance charges high. The Eastern paraffine base oils can be cleaned much easier than the Western asphaltum base oils. In the latter the asphalt collects around the sand particles, and it is impossible to separate them except by heating the oil and straining it while hot.

45 Some engines are working today on Western oils being sold in the market as boiler fuel oil. The Diesel engine located on the Jameson ranch at Corona, Cal., is now running on 24 to 26 deg. B. Santa Fé tops bought for the smudge pots. The oil costs 69 cents per barrel, F.O.B. Los Angeles. This engine has been run for more than 2 months without a shut-down. About a quart of coal oil was supplied twice a day to clean out the pump. The engine was run for 10 days on 18 deg. fuel oil then in use on a steam road roller in

Corona, and the run was stopped at the end of that period, as the supply of that particular fuel was exhausted.

46 The Lyons-Atlas Company sold a 600 b.h.p. engine to the Hawaiian Commercial and Sugar Company on the guarantee of 710 hours operation out of 720 per month on 14 to 18 deg. California oil, then in use under the Company's boilers in the Hawaiian Islands. The engine was tested at the factory under the supervision of the company's engineer, was paid for, and then shipped through the Panama Canal to the Islands. The test consisted of a 48-hour pre-

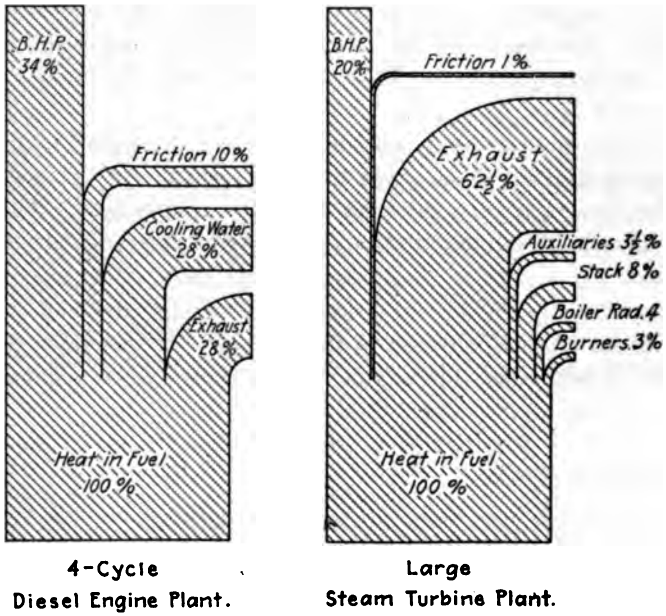


FIG. 4 COMPARISON OF HEAT BALANCES

liminary run on Eastern oil and then a 144-hour (6 day) continuous run at rated load on the California oil. At the end of the run, the valves and the heads were examined and no evidence of any deposit was found. A 120 h.p. 4-cycle engine has been running for 6 months, 24 hours a day, without a stop, in San Antonio, Texas. The fuel was a 20-deg. Texas oil. There are numerous other examples of engines on this coast but not using California oils. Many engines are in operation in Texas, New Mexico and Arizona using Texas and Mexican oils.

ECONOMY AND EFFICIENCY

47 The economy of the Diesel engine is the best of all present engines. Fig. 4 shows the heat balance for a 4-cycle Diesel engine, and also for one of the latest large steam turbine plants. In preparing this heat balance for the Diesel engine, a mechanical efficiency of 77 per cent was assumed and the thermal efficiency of an actual engine, as shown by test, was used as a basis for the remainder. The distribution of waste heat between exhaust and cooling water for this engine varies, so that an equal distribution was assumed.

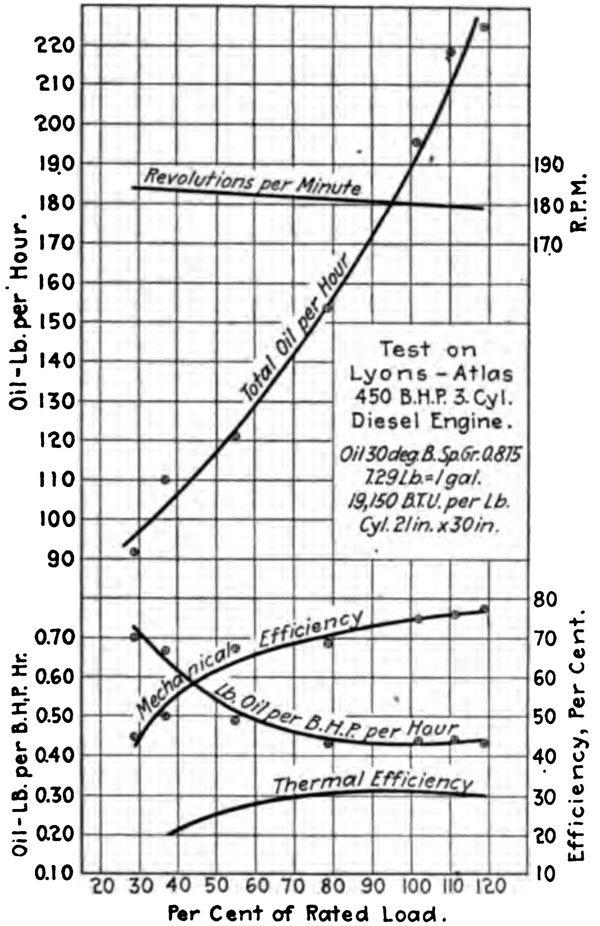
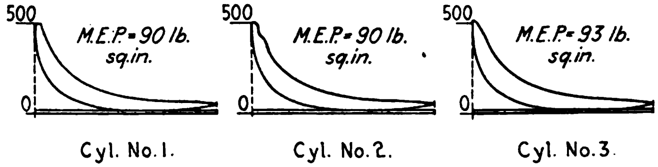


FIG. 5 RESULTS OF TESTS ON 3-CYL. LYONS-ATLAS DIESEL ENGINE

48 The heat balance of the turbine plant is a composite heat balance based upon an oil-fired boiler and a turbine generator unit, using the best steam figure that I have record of. 95 per cent is allowed for the efficiency of the generator and 95 per cent for the mechanical efficiency of the turbine.

49 The author's idea in making this comparison is to show the best thermal efficiency in both types of prime movers, thus indicating the superiority of the Diesel engine as far as thermal efficiency is concerned. So far as is known, however, no turbine plant is operating today with an over-all thermal efficiency quite as high as the 20 per cent shown.

50 The efficiency of the Diesel engine may be still further increased by utilizing the heat in the exhaust for making steam to run



Scale of spring, 400 lb.
Injection Air Pressure, 1045 lb. per sq. in. abs.

FIG. 6 INDICATOR CARDS FROM 3-CYL. LYONS-ATLAS DIESEL ENGINE

a steam turbine. Experiments are now being carried out in this direction, but the results are not yet good enough to indicate that this can be done in all cases. Experiments are also being made along the line of increasing the temperature of the jacket water, so that this may be converted into steam which may be used.

51 Figs. 5 and 6 show respectively efficiency curves and indicator cards for a 3-cylinder Lyons-Atlas Diesel engine, using an Eastern oil. Fig. 5 shows clearly that from about 60 per cent to 120 per cent of the rated load the economy and thermal efficiency remain nearly constant. The mechanical efficiency tends to increase as the load increases. This factor is about 75 per cent at full load for a 4-cycle engine and 70 per cent for a 2-cycle engine (This is due to the power required for the air compressor for the scavenging air).

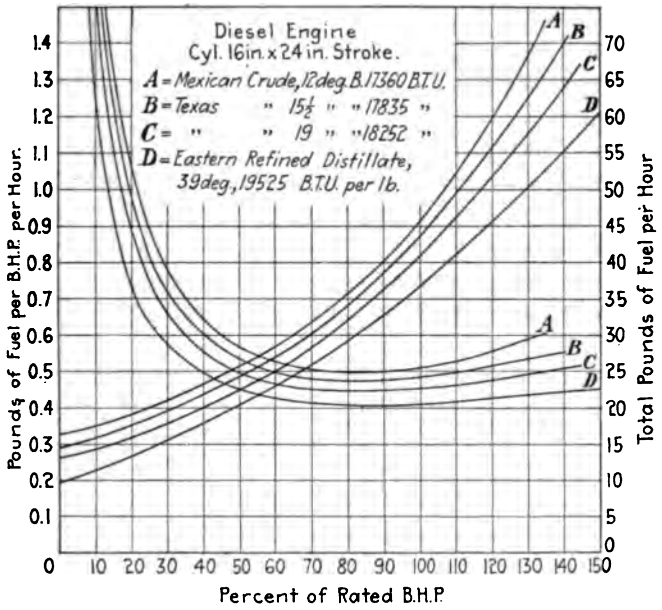
52 Fig. 7 shows the results of a series of tests on an engine, using various fuels. When the best grade of fuel indicated is used, the amount is about 0.4 lb. per b.h.p.-hr., while it does not exceed 0.5 lb. with the poorest grade at rated load.

53 The Diesel engine will work nearer to test conditions at all

times than any other type of prime mover, as it is more independent of the operator and requires a good compression for ignition. The efficiency depends upon the compression; if this latter drops due to valve trouble the engine will show it at once, as there will not be heat enough to ignite the fuel.

OPERATING AND DEPRECIATION

54 The amount of lubricating oil is stated to be about 0.01 pint per b.h.p. per hour, based on the rated load. The engine at Corona



Curves starting lower left hand corner show total fuel

FIG. 7 DIESEL ENGINE, COMPARISON OF FUELS

uses 3 quarts of oil every 24 hours to supply the loss. The load is about 35 h.p. while the rating of the engine is 65 h.p. This is at the rate of 0.008 pint per h.p. per hour.

55 One engineer can handle 1000 to 1500 horsepower per shift. The attendance consists in keeping the engine supplied with both fuel and lubricating oil and the minor work that there always is around a power plant. Operators and manufacturers say that it is necessary to examine the fuel valve and the exhaust valves periodically and clean them. The time interval for this depends upon the kind of fuel in use—it may be a week or several months.

56 The question of maintenance and depreciation is still an open one. The maintenance charges per year seem to average about 1 per cent of the first cost of the engine. Very few manufacturers have yet had engines in service for any length of time, so that the life of the engine is still uncertain, although it is claimed to be longer than that of a steam engine. The Busch-Sulzer Diesel Engine Company have two 225 b.h.p. engines installed in Texas which were put in over nine years ago; these have been operating on an average of 18 hours a day. The cylinders have never been rebored and now show very little wear and are as smooth and bright as glass. The same company also has a 225 b.h.p. engine in Illinois which has been working for 24 hours a day, 6¾ days a week, for over 2½ years, with only two minor shut downs.

57 The following facts seem to be fairly well established:

1 The Diesel engine can operate continuously for 6½ or more days out of seven
2 This can be kept up for long periods if a short interval is allowed for overhauling and minor repairs
3 The exhaust valves may give trouble by burning if the load is too large
4 The air compressor may give more trouble than the engine if it is not watched
5 Dirt in the oil will give trouble
6 Water in the oil will give trouble
7 The engine will not carry much more than its rated load for any length of time.

WEIGHTS AND COSTS

58 The weight of the Diesel engine per horsepower varies considerably, even for the same size of engine. Data supplied by the manufacturers in this country show that the weight per b.h.p. varies from 250 to 500 lb., with no uniformity, except that the higher-priced engines are the heavier. In an issue of London Engineering during 1914, the statement was made that in European practice the weight had been reduced to 62 lb., but this was an exception.

59 The cost of the engine is hard to determine, as it varies so much and manufacturers do not like to supply cost data. The price increases directly with size of engine from $6000 for a 75 h.p. engine to about $48,000 for a 1000 h.p. engine. The price of a small Diesel engine is prohibitive and that for large engines of several thousand horsepower does not go much below $45 per horsepower. Cost curves

for the Pacific Coast plotted from data supplied by several manufacturers are given in Fig. 8.

COST OF A SMALL PLANT

60 Table I shows a comparison of the cost of steam turbine and Diesel engine plants, 600 kw. The plants are suppositious, but the cost figures given can be considered as approximately correct. They show that the Diesel may enter into serious competition with the steam plant when the load factor is better than 25 per cent. The Diesel

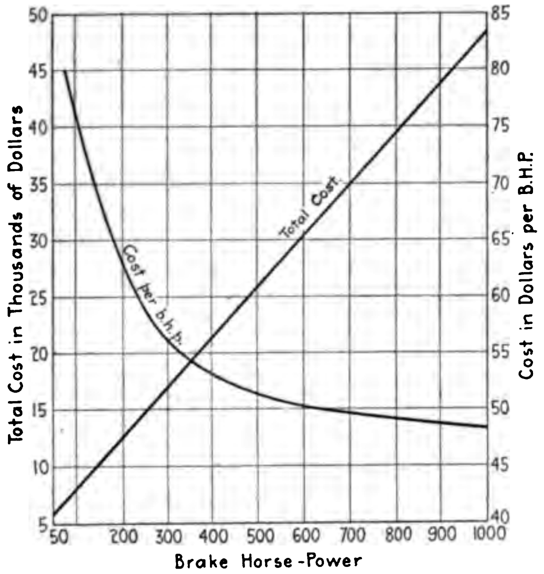


FIG. 8 COST OF DIESEL ENGINES, F. O. B. LOS ANGELES, CAL.

engine will not replace the steam plant until much more definite figures are secured regarding the life of the former.

61 As the yearly load factor is increased the Diesel engine will show a saving in total operating cost due to the saving in fuel. In the steam plant the fuel item is the largest single item of expense, with the fixed charges next. In the Diesel plant the fixed charges are the largest single item, with the remainder of the operating cost about equally divided between fuel and labor, water, etc.

62 The items under first cost of the two plants are only approximately correct, as the author had no personal data available and had to depend upon published results which did not check with one

TABLE 1 COMPARISON COSTS, STEAM TURBINE AND DIESEL ENGINE, 600 KW. MUNICIPAL POWER PLANT

ASSUMPTIONS

Load Factor = 25 per cent Maximum load = rated output

(This gives turbines slight advantage in overload capacity.)

Turbines operated condensing, using jet condenser and cooling tower. Oil Fuel. Crude oil, 95 cents bbl. Distilled oil, $1.50 bbl.

Turbine Plant develops 140 kw-hr. per bbl.
Diesel Plant develops 447 kw-hr. per bbl.

FIRST COST

TURBINE PLANT		DIESEL ENGINE PLANT			
1-200 kw., 1-400kw. Units		3-200 kw. Units		1-200 kw., 1-400 kw. Units	
Boilers and Settings	$ 6,200	Engines	$51,000	Engines	$47,500
Pumps	250	Erecting	5,000	Erecting	5,000
Piping	500	Piping	1,400	Piping	1,400
Stack and Flues	2,950	Oil tanks	1,000	Oil tanks	1,000
Heaters	500	Water Cooling Apparatus	1,000	Water Cooling Apparatus	1,000
Turbines	12,500				
Generators, etc.	11,400	Generators	11,400	Generators	11,400
Condensers	2,400	Building	6,000	Building	6,000
Cooling tower	3,500				
Building	10,000				
Total	$50,200	Total	$76,800	Total	$73,300

OPERATING COSTS
1,314,000 kw-hr. per Year

TURBINE PLANT		DIESEL ENGINE PLANT	
Wages	$3,000	Wages	$3,000
Lubrication	500	Lubrication	500
Miscellaneous	100	Miscellaneous	100
Maintenance	400	Maintenance	400
Water	250	Water	50
	$4,250		$4,050

TURBINE PLANT		3 Engines		2 Engines	
		Fuel		Fuel	
		95 cents bbl.	$1.50 bbl.	95 cents bbl.	$1.50 bbl.
Fuel at 95 cents per bbl.	$8,910	Fuel . . . . $ 2,790	$ 4,410	$ 2,790	$ 4,410
Fixed charges 14 per cent	7,030	Fixed Charges 14 per cent 10,780	10,780	10,280	10,280
Total	$20,190	Total . . $17,620	$19,240	$17,120	$18,740

DISCUSSION OF THESE VALUES

Difference in first costs = $73,300 - $50,200 = $23,100.

Diesel Engine Plant costs 46 per cent more.

Difference in operating costs.

(*a*) $20,190 - $17,120 = $3,070. Net saving per year = $3,070

(*b*) $20,190 - $18,740 = $1,450. Net saving per year = $1,450

Conclusion: Difference in yearly cost is so small that no definite conclusion can be drawn Each plant should be investigated carefully before type of equipment is decided.

another. The output per barrel of oil is based upon published yearly reports of both Diesel and steam plants. The steam plant is located in California while the Diesel plant is in Texas. The distilled oil cost was purposely placed high, so as to indicate the showing that

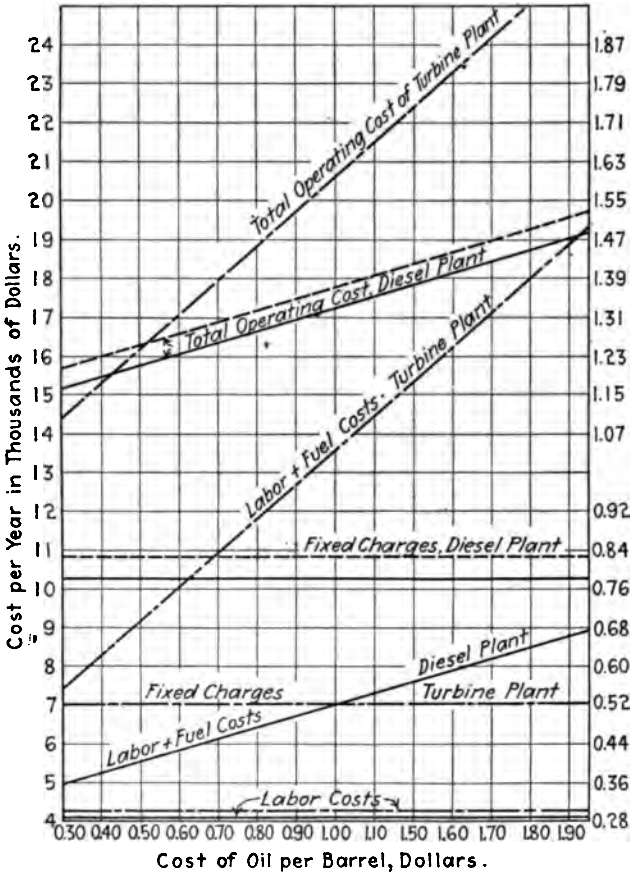


FIG. 9 COMPARISON COST CURVES, STEAM TURBINE AND DIESEL ENGINE, 600 KW. MUNICIPAL LIGHTING PLANT

a Diesel engine could make in competition with a steam plant, even if it was handicapped with a high price of suitable oil. As we are not interested in comparison of coal and oil as fuels, no attempt has been made to show such. The items under operating cost were taken from a published report of a Diesel plant in Texas, where wages are low in comparison to those in California. In the turbine plant about the same conditions of operation have been assumed, as the total of the items listed under lubrication and miscellaneous are small.

63 In Fig. 9 showing cost curves are plotted the yearly costs for the plants against price of oil per barrel. This shows graphically the effect of the price of oil on the total operating cost. By projecting from the total operating cost curve of the steam turbine plant to the corresponding curve for the Diesel plant, a comparison may be secured for any price of oil. For example, assume the price of boiler fuel oil to be 75 cents and engine oil to be 90 cents per barrel. Then the turbine plant will cost $18,400 per year, and the Diesel plant $17,500 per year. When the price of oil is 53 cents per barrel, the yearly cost will be the same for both plants.

64 On account of the lack of data relative to the life of the Diesel engine, the same fixed charges, namely, 6 per cent depreciation, 6 per cent interest and 2 per cent insurance and taxes, have been assumed for both plants for comparison.

65 The item of life of the Diesel engine is open for discussion, but no one can yet say definitely what the life of the Diesel engine, when properly taken care of, is going to be as none of our successful plants have been in operation long enough to give the answer.

DISCUSSION

H. R. Setz said that to answer the question of life of the Diesel engine we will have to look to Europe, because there the Diesel engine had its conception, and there it has been perfected to the highest stage. Experience there has proven that all the parts which are subject to wear, at least in a well-designed engine, can be replaced at a cost of about 35 to 40 per cent of the initial cost of the engine. These parts include pistons and cylinder liners, for instance, as well as valves, bushings, and all those parts subjected to natural wear of the engine. European practice has brought out that these parts will stand up easily from eight to ten years.

He said that in 1901 he took part in a test on a two-cylinder engine in which after 12 years of operation the pistons and one cylinder liner have been replaced, but all the other parts are practically the same as in the original engine contained.

Regarding the fuel question, he said there are many different kinds of fuel oils, and a thing which ought to be done in this country, and probably by a committee of members of the Society, is to make a systematic study of fuel oils. In Switzerland, a very comprehensive study was made by actually running a Diesel engine for two or three weeks at a time on various fuels. Unfortunately, the fuels that were tried were all, with one or two exceptions, European oils, from Galicia, Roumania and Baku. There were only two American samples used; one was an Oklahoma oil and the other a California oil, both asphalt base oils. The latter should be investigated.

Referring to a few examples of his experience with asphalt base oils, he said there is no standard definition for asphalt. Asphalt is defined sometimes as the residue which remains after heating the oil for a certain number of hours at a certain temperature in the open-air. But the residue may contain many other things than asphalt, so that is not a conclusive test at all. Another method of determining asphalt, is by a chemical analysis, and even in that there is no accepted standard. When he started the first Diesel engine in the Southwest, he was told the oil contained 32 per cent asphalt, and he thought the engine would never run on that fuel. Yet to his great surprise, the engine ran and is running today with nothing else than that particular grade of oil. It is a California oil of 14.6 gravity, Baumé, and the engine can even be started, and is regularly started, on that kind of oil—a thing which no engineer in Europe would believe possible unless he saw it.

The usual method of designating an oil in regard to its suitability for operation in Diesel engines is to give the gravity, either Baumé or specific gravity. From a number of California oils, it has been found that the heat values range from 18,000 to almost 20,000 B.t.u. per pound, and that the percentage of asphalt contents varies as much as 25 per cent. This indicates that specific gravity is not the only criterion for determining the value of fuel oil.

Another matter referred to in the paper is the percentage of water. This, he said, is not in itself enough to designate the oil as to its suitability for Diesel engine use. For instance, an engine operated on a Mexican crude oil containing over 2 per cent water ran for quite a while very nicely. All at once it began to run jerkily, irregularly, and finally shut itself down. After investigating it was found that this Mexican crude oil was so heavy that the water remained in it suspended in very small drops, and separated out but very little. These very small particles of water had been pumped in gradually with the oil into the injection valve—not into the cylinder but into the injection valve on the atomizer plates.

This valve serves the purpose of disintegrating the fuel before it is injected, and the atomizer plates have a temperature high enough to evaporate the water. This water vapor rose inside the injection valve space to the upper end of the valve, and there gradually condensed and formed a drop big enough so that afterwards, when it was blown into the cylinder, it shut the engine down.

Another oil which contained about 5 per cent of water did not cause any trouble at all, because the oil was of such a consistency that the water was easily separated and could be drawn off; no water came into the cylinder at all.

A feature of some California oils is the salt contents. These salts may under unfavorable conditions form deposits that might cause excessive wear. This is another thing which ought to be investigated.

R. L. Rowley asked Mr. Setz what was the smallest size of four and six-cylinder engines, and also the weight now adopted for Diesel work in European practice?

H. R. Setz replied the question of weight is entirely dependent upon the speed of the engine. Up to a short time ago, in order to meet requirements of smaller submarine construction, speeds were increased and weights reduced beyond what is considered desirable limits today. Later practice is tending towards heavier weights and more rigidity in the engine, so as to insure longer life.

Roger D. De Wolf (written). The data used by the author in compiling Table 1 do not agree with data possessed by the writer, and I would like to submit a revised comparison.

My turbine data are based upon quotations and guarantees made about two years ago. Fig. 10 shows the relation between the cost per kw. and kw. capacity of turbines ranging in size from 100 to 750 kw. This curve is based upon actual quotations, and includes the cost of the turbine and generator. It shows the cost of a 200-kw. unit to be $31.00 per kw. and a 400-kw. unit to be $21.50 per kw., or an average cost per kw. for these two units including generators, of $24.75. The cost which the author gives for turbines and generators is $39.80 per kw. It should be noted that he has assumed that the generator would cost the same in the turbine plant as in the Diesel engine plant; this is obviously wrong, since the generator for the former would be very much higher in speed than that for the latter.

Fig. 11 shows the water rates which the manufacturers were willing to grant on these turbines, based on 175 lb. gage pressure and 2 in. absolute vacuum; and Fig. 12 shows the total steam required per hour under different loads. Based on these curves, the steam consumption for the plant would be as follows:

200 kw. at 24.5 lb.	4900 lb.
400 kw. at 20.5 lb.	8200 lb.
Total to turbine	13,100 lb.
Additional 10% for auxiliaries	1,310 lb.
Total	14,410 lb.

To furnish this amount of steam, assuming that the boilers are operated at 200 per cent of rating, or, roughly, 60 lb. evaporation per rated boiler hp., the total boiler hp. required would be $14{,}410 \div 60 = 240$ hp.

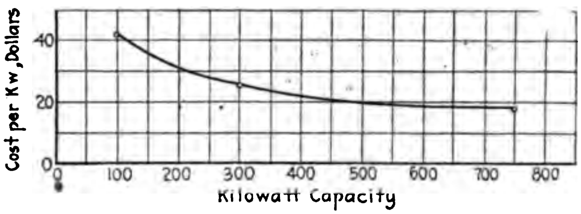


FIG. 10 RELATION BETWEEN TURBINE COST AND CAPACITY

The proceedings of the National Electric Light Association at the Chicago convention in 1913, include a table[1] giving the heating value of different California oils as from 17,600 to 18,878 B.t.u. per lb. I have assumed an average value of 18,500. The weight of one bbl. of oil is given as 336 lb., and the B.t.u. per bbl. would then be 6,216,000.

Numerous boiler tests using oil as fuel have shown an efficiency considerably better than 80 per cent. In a modern plant there should be no great difficulty in getting an efficiency of 80 per cent, which would mean that there would be 4,972,800 B.t.u. available per bbl. of oil.

Referring again to Fig. 12, it will be seen that for the average load of 150 kw., based on an assumed load factor of 25 per cent the 400-kw. machine would use 36,000 lb. of steam per hour, or 24 lb. per kw-hr., which is less than the consumption of the smaller unit for the same load. Adding 10 per cent for auxiliaries, the steam con-

[1]Page 466 of the Technical Volume.

sumption per kw-hr. would be 26.4 lb. In this connection it should be noted that the steam consumption per kw-hr. is, under maximum load conditions, only 24 lb. per kw-hr., including auxiliaries, so that this figure of 26.4 per kw-hr. as an average is conservative.

Steam at 175 lb. gage pressure contains approximately 1197 B.t.u. per lb.; assuming the temperature of the condensate to be 82 deg. fahr., the net heat per lb. of steam is $1197-50=1147$ B.t.u. The heat required per kw-hr. is then $1147 \times 26.4 = 30,300$, and the kw-hr. per bbl. of oil will be $4,972,800 \div 30,000 = 164$. A generation of 1,314,000 kw-hr. per year will then require 8,000 bbl. of oil which, at $ .95 per bbl., comes to $7,600.

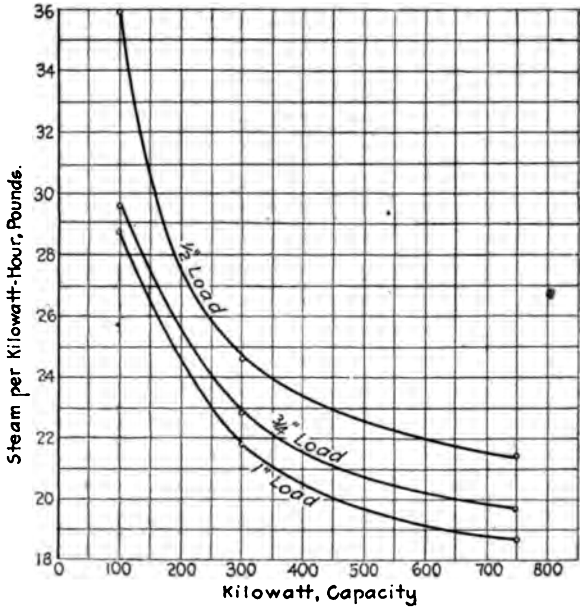


FIG. 11 TURBINE WATER RATES. 175 LB. GAGE. 2-IN. ABSOLUTE VACUUM

COST OF TURBINE PLANT

Boilers, 240 h.p. @ $18.00	$ 4,320
Pumps	250
Piping	500
Stacks and flues	2,950
Heaters	500
Turbines and generators	14,800
Switchboard	1,200
Condensers	2,400
Cooling tower	3,500
Building	10,000
Total	$40,420

OPERATING COSTS

Wages	$3,000
Lubrication	500
Miscellaneous	100
Maintenance	400
Water	250
Fuel	7,600
Fixed charges @ 14%	5,658.80
Total	$17,508.80

In the operating costs of the Diesel engine plant, the author assumes the cost of lubrication to be the same as in the case of the turbine plant. I believe there is no question but this cost would be at least 50 per cent greater in the case of the Diesel engine plant.

In the table the author also gives the cost of maintenance at $400, whereas in Par. 56 he states that the average maintenance cost per year is about 1 per cent of the first cost of the engine. I have therefore revised his operating costs by adding $250 to the cost of lubrication, and $110 to the cost of maintenance for the 3-unit plant, and $75 to the cost of maintenance for the 2-unit plant. These costs would then be as follows:

OPERATING COSTS, DIESEL ENGINE PLANT

Wages	$3,000
Lubrication	750
Miscellaneous	100
Water	50
Total	$3,900

3-ENGINE PLANT

WITH $ .95 OIL		WITH $1.50 OIL	
Total from above	$3,900	Total from above	$3,900
Maintenance	510	Maintenance	510
Fuel	2,790	Fuel	4,410
Fixed charges @ 14%	10,780	Fixed charges @ 14%	10,780
Total	$17,980	Total	$19,600

2-ENGINE PLANT

WITH $ .95 OIL		WITH $1.50 OIL	
Total from above	$3,900	Total from above	$3,900
Maintenance	475	Maintenance	475
Fuel	2,790	Fuel	4,410
Fixed charges @ 14%	10,280	Fixed charges @ 14%	10,280
Total	$17,445	Total	$19,065

This shows the turbine as cheaper in all cases except in the case of the 2-engine plant using $ .95 oil, where the Diesel engine shows a saving of $63.80 per year.

The difference between the first cost of the turbine plant and the 2-unit Diesel engine plant is $32,880, or 81 per cent more than the cost of the turbine plant.

It should be further noted in the above that no credit is given in the turbine plant for the heat given up to the feedwater by the steam used by the auxiliaries. Evidently the theoretical saving with the Diesel engine plant of $63.80 is very problematical.

In many cases it would be unnecessary with the turbine plant to erect a cooling tower, so that the investment required for the turbine plant could be further reduced to $36,920.

George W. Dickie said he was sorry that the paper did not go a little more into the question of the marine Diesel engine, as this was a subject of particular interest.

He related how about five years ago a prominent shipowner in San Francisco, who had a boat building for his lumber business in the Pacific Coast shipping service, wanted to try the Diesel engine. The owner wanted an engine of 1200 h.p., and told him that he could spend $100,000 on it if necessary, so he wrote to a firm of

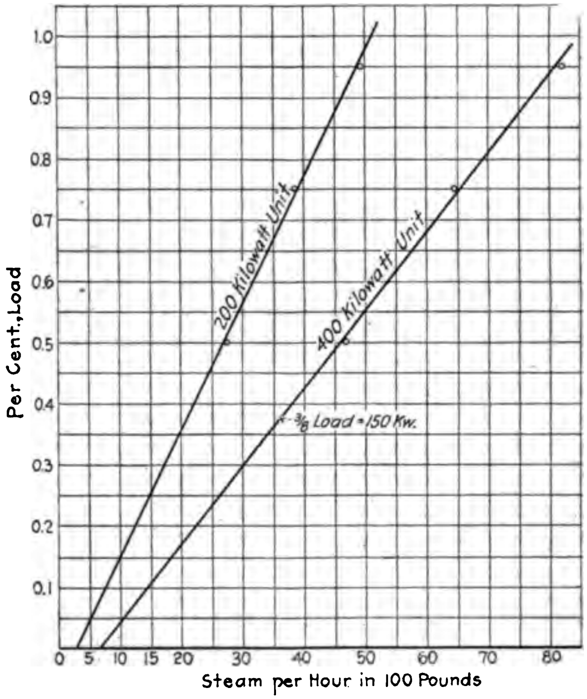


Fig. 12 Total Steam per Hour Required with Turbines

engine builders whom he knew very well and could depend upon. They asked for samples of the fuel to be used, and he sent them some twenty samples of California oil just as it came from the wells, but they would not undertake to build any Diesel engine to run and be depended upon with that fuel. On that account the Diesel engine was not used, and so far as he knew no attempt had as yet been made to put the Diesel engine in service on that Coast.

He thought, however, that Diesel engines might come to be used in the Coast service if they could be depended upon with California fuel. With the Diesel engine, the operating expenses are relatively high and the engine room force must consist of all skilled men, both of which factors must be taken into account in considering putting in an engine of this type.

REGINALD W. CROWLEY[1] (written). The author is not safe in venturing a prophecy concerning the outcome of the battle between the 2 and 4-cycle types of engine. Design, it is true, produces occasional advances in technics, but it is to metallurgical science that we must eventually turn for assistance in striving to overcome the difficulties presented by the large Diesel engine.

Referring to the statement that in the 2-cycle engine air inlet and exhaust valves are not required, it is to be noted that it is more usual to find the flow of scavenging air controlled by valves. In only three designs are exhaust valves and air admission valves absent. Between the extremes there occur two interesting intermediate types. The trend is markedly towards the control of scavenging air by valves.

The statement that in marine work the reduction in number of valves makes it easier to reverse a 2-cycle engine is ambiguous. What has been the least simple reversing mechanism applied to any Diesel engine has been that fitted to all Carels 2-stroke sets; on the other hand the small Polar 2-cycle sets have a simpler reversing device than any other engine. The most that can be said in favor of the 2-cycle engine in this regard is that it *can be adapted* to have a simpler reversing mechanism than the 4-cycle.

Literally it is correct that the 2-cycle engine has made large units possible, and that 1200 h.p. per cylinder in a single-acting engine of this type has been built. But without the proper qualifications to this statement, would it not be inferred that the large 2-cycle engine is a commercial realization and not merely the experimental production it is at present?

As a matter of fact, Sulzer's have been making trials with a single-acting unit which develops about 2000 h.p., and in the double acting unit, which it seems appropriate to mention here, over 2000 h.p. per cylinder has been attained in experimental sets in the Krupp shipyard and the Nuremburg works. Yet, of all this costly effort no practical commercial result has been born.

[1]Jas. Craig Eng. and Mach. Wks., Jersey City, N. J.

Not having at hand references from which to quote cylinder dimensions of the smallest Diesel engine that has been marketed, I can amend the author's citation only by the bald assertion that the Delaunay-Belleville establishment, of Paris, exhibited at the Turin Exposition, 1911, a single-cylinder 4-cycle unit developing 5 h.p. at about 500 r.p.m. Louis Nobel furnished from his Petrograd factory to the Russian Imperial Navy, 50 h.p. four cylinder 4-cycle sets to propel its diminutive submarines of 50 tons displacement commissioned about 1911 or 1912. Other examples might be cited.

The higher limit is reached certainly in the 2000 h.p. single-cylinder built by Sulzer for trial; this, being operated on the 2-stroke cycle, has a cylinder diameter of probably about 44 in. In all likelihood, the big Krupp & M.A.N. double acting sets with about 2000 h.p. per crank have a larger cylinder diameter than that quoted by Mr. Adams for the Carels engine. This latter is given as developing 1250 i.h.p. and its mean effective pressure as 106 lb. per sq. in. It may also be observed that to obtain even this indicated power a piston speed of nearly 1000 ft. per min. is utilized and the mean pressure is higher than Carels have ever succeeded in maintaining in a 2-cycle cylinder.

THE AUTHOR. My attention has been called to several statements in the paper which do not check with data as given at the present time. This is due partly to the fact that the Diesel engine is being developed rapidly and conditions are changing from day to day.

The curves of Fig. 3 have been criticised because they are based upon the adiabatic curve $PV^{1.4} = \text{constant}$. The actual exponent of the curve is a variable for different engines. The adiabatic is a standard curve and gives approximately the same temperatures. The actual temperatures are slightly lower. My curves check the temperatures in the cylinder at the end of compression, as I used a lower temperature, at the beginning of compression, than the actual temperature.

Table 1 has caused considerable discussion. The steam turbine engine engineers maintain that I have not treated the turbine fairly, and the Diesel engineers say the same about the Diesel engine. Under such circumstances I feel that I have kept in the middle ground, which is what I hoped to do.

My answer to Mr. DeWolf's costs is that his figures are for the Eastern States, while mine are for California. My total cost for the turbine plant has been accepted as approximately correct by turbine engineers in Los Angeles. (It is only fair to say that they do not agree with me on the individual items.)

I do not agree with Mr. DeWolf's figure of 80 per cent for the efficiency of oil-fired boilers. Such boilers on test give this figure, but there are very few, if any, boilers on the Pacific Coast operating at this efficiency in everyday service. Certainly there would be none in the plant I selected.

Mr. DeWolf's figure of 164 kw-hr. per bbl. of oil is too high for such a plant. I can supply published data of a very successful plant in Southern California, having a load factor of over 30 per cent, which produces under 140 kw-hr. per bbl. of oil. It is a larger plant than the one under discussion. Some of the larger plants may have such an output as mentioned by Mr. DeWolf but I have no record of any.

I do not feel that the Diesel engine can displace the turbine as a prime mover in plants with a small load factor, as the depreciation forms such a large part of the production cost, and the first cost of the Diesel plant is 50 to 100 per cent more than the first cost of a steam plant. As the load factor is increased the unit production cost is decreased by using the Diesel engine. This is clearly shown by the figures in Table 2, taken from an article by R. H. Burdick, in the *Electrical World,* March 11, 1916. These data show the analysis of the costs of a plant now in operation. The load factor is 50 per cent. Comparison is made with the operating costs of a plant with a load factor of 28.5 per cent.

Referring to Fig. 9, in the original paper, the cost of a kw-hr. is 1.44 cents. This checks quite closely with Table 2.

Mr. Crowley's criticism has added considerable information to that contained in the paper, but I do not agree with some of his statements. The cost of the Diesel engine prohibits its use in this country in the smaller sizes, except for very special purposes. It is possible to make double acting internal combustion engines, but most of our engines of this class are single acting. This shows the tendency now; I do not care to prophecy about the future. I know of no manufacturer in the United States who is making a double acting Diesel engine.

The 2-stroke cycle is used in the large engines. Most of our smaller engines use the 4-stroke cycle. I know of one manufacturer who has given up the manufacture of small 2-stroke cycle Diesel

TABLE 2 COMPARATIVE ESTIMATES FOR DIESEL AND STEAM EQUIPMENT FOR PARIS (TEXAS) STATION

	Estimated for Diesel Station						Estimated for Steam Station					
	Quantity	Unit Used	Unit Cost	Cost per kw.	Per cent Total	Total	Quantity	Unit Used	Unit Cost	Cost per kw.	Per cent Total	Total
Station building	134000	cu. ft.	0.10	$12.76	8.8	$13,395	108000	cu. ft.	$0.12	$11.78	11.7	$12,960
Oil engine equipment	1050	kw.			45.3	69,485						
Boiler equipment							900	b.h.p	21.40	17.50	17.4	19,260
Turbo generator equip.							1100	kw.	24.00	24.00	23.9	26.400
Electrical equipment	1050	kw.	27.30	27.30	18.7	28,620	1100	kw.	16.50	16.50	16.4	18,150
Transformer yard	1275	k.v.a.	4.15	5.03	3.5	5,280	1275	k v a.	4.15	4.80	4.8	5.280
General station equip.		Job		4.56	3.1	4.785		Job		5.20	5.2	5.720
Improvements to grounds		Job		0.29	0.2	300		Job		0.27	0.3	300
Construction plant				2.32	1.6	2,440				1.60	1.6	1.760
Overhead expense				27.39	18.8	28,695				18.78	18.7	20,660
TOTAL COST	1050	kw.	145.75		100	$153,000	1100	kw.	100.44		100	$110.490

PLANT DATA	For Diesel Station	For Steam Station
Size of plant	1050 kw.	1100 kw.
Number of units	3	2
Station factor	50 %	48 %
Yearly output	4,600,000 kw-hr.	4,600,000 kw-hr.
Fixed charges	15 % = $22950	12 % = $13260
Interest	7 %	7 %
Depreciation	8 %	5 %
OUTPUT COSTS		
Production costs	4.66 mils per kw-hr.	8.56 mils per kw-hr.
Fixed charges	4.85 mils per kw-hr.	2.88 mils per kw-hr.
Total output cost	9.51 mils per kw-hr.	11.44 mils per kw-hr.

Estimated annual saving—Diesel over Steam—$8880

Oil @ 3 c. gallon Coal $2.50 per ton

ACTUAL UNIT PRODUCTION COSTS, PARIS AND TYLER DIESEL STATIONS, SEPT. 1 to DEC. 1, 1915

DATA	PARIS	TYLER
Station output in kw-hr.	1,565,000	499,000
Rating of plant—kw.	1,050	600
Station factor in per cent	51	28.5
Total fuel oil in gallons	149,072	78,455
Pounds of oil per kw-hr.	0.672	1,100
B. t. u. per kw-hr.	13,100	21,400
Kw-hr. per bbl. of oil	441	267

PRODUCTION COSTS IN MILS PER KW-HR.	PARIS	TYLER
All labor	1.44	2.24
Fuel oil	3.07	5.18
Water	0.09	0.19
Lubricants and waste	0.04	0.56
Miscellaneous supplies	0.10	0.29
Maintenance of engines	0.04	4.48
Maintenance of buildings	0.05	0.05
All other maintenance	0.15	0.61
Total production cost	4.98	13.60

engines, as they were not found satisfactory. Large 2-stroke cycle engines appear to be satisfactory. What the future holds in store no one knows.

No. 1493

THE HEAVY OIL ENGINE, ITS PRESENT STATUS AND FUTURE DEVELOPMENT

By

A. H. Goldingham, New York, N. Y.

Member of the Society

Since the oil engine was invented, about 1870, rapid progress has been made with it. This is best demonstrated by reference to the indicator diagrams, Figs. 1 to 8, taken from engines built during the past twenty-seven years.

2 The first diagram was taken from a Priestman oil engine, 1888 type, cylinder 10¾ in. diameter, stroke 14 in., 160 r.p.m. The initial pressure was 125 lb. per sq. in., exhaust pressure 24 lb., compression pressure 20 lb. and m.e.p. 44 lb. The b.h.p. developed was 8.4, using fuel with a heating value of 19,000 B.t.u. and specific gravity 0.853. The fuel consumption was 1.05 lb. per b.h.p.-hr. and the thermal efficiency 12.8 per cent. All these figures represent average conditions. The fuel was sprayed into an external vaporizer (separate from the cylinder) heated by the exhaust gases, the ignition being effected by an electric ignitor.

3 Fig. 2 is a card from a Hornsby-Akroyd engine, 1890 type, with hot-surface vaporizer constructed without the partial water jacket of later engines, and with very low compression pressure. The engine represented developed 6 b.h.p. at 216 r.p.m., with fuel having a heating value of 19,000 B.t.u. and specific gravity 0.8410. Fuel consumption 1.0 lb. per b.h.p.-hr. and thermal efficiency 13.5 per cent. The initial pressure was 120 lb. per sq. in., exhaust pressure 20 lb., compression pressure 40 lb. and m.e.p. 35 lb.

4 In subsequent types of the same engine, the compression pressure was increased with consequent increase in thermal efficiency.

The distillate engine largely in operation in the Pacific Coast states is not discussed in this paper. Only engines using heavy fuels are referred to.

Presented at the Panama-Pacific International Exposition Meeting, San Francisco, September, 1915, of The American Society of Mechanical Engineers.

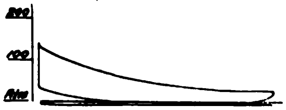


Fig. 1

Fig. 2

Fig. 3

Fig. 4

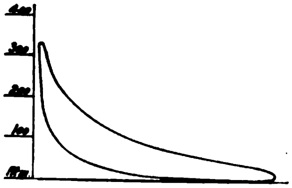


Fig. 5

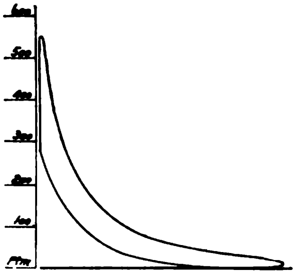


Fig. 6

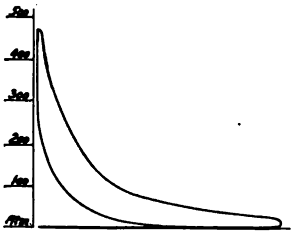


Fig. 7

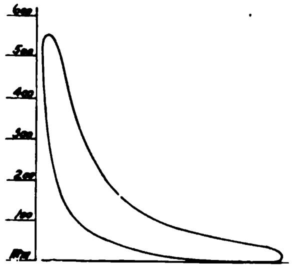


Fig. 8

Indicator Diagrams showing Progress in Oil Engines since 1888

This is shown in Figs. 3 and 4, which are diagrams from 1893 and

1905 types Hornsby-Akroyd engines respectively. Fig. 3 represents an 8.02-in. by 14-in. engine developing 5 b.h.p. at 214 r.p.m.; initial pressure 120 lb. per sq. in., exhaust pressure 22 lb., compression pressure 46 lb., m.e.p. 37.5 lb.: heating value of fuel 18,600 B.t.u., specific gravity 0.824, fuel consumption 0.99 lb. per b.h.p.-hr. Thermal efficiency 13.8 per cent.

5 Fig. 4 is taken from an engine with cylinder diameter 14.5 in. and stroke 17 in.; r.p.m. 202; initial pressure 168 lb. per sq. in., exhaust pressure 30 lb., compression pressure 60 lb., and m.e.p. 48 lb. B.h.p. 27. Heating value of fuel 19,000 B.t.u., specific gravity 0.825. Fuel consumption 0.74 lb. per b.h.p.-hr. Thermal efficiency 18 per cent.

6 Fig. 5 is taken from a De La Vergne oil engine, type DH, built in 1913. Cylinder diameter 14 in., stroke 24 in. Initial pressure 325 lb. per sq. in., exhaust pressure 25 lb., compression pressure 169 lb. and m.e.p. 75 lb. Heating value of fuel 18,500 B.t.u., and fuel consumption 0.543 lb. per b.h.p.-hr. Brake horsepower 60 at 210 r.p.m. Thermal efficiency 25 per cent.

7 Fig. 6 is a diagram from a Ruston Proctor engine, type 1913. Initial pressure 550 lb. per sq. in. approximately, exhaust pressure 25 lb., compression pressure 280 lb. and m.e.p. 70 lb. Heating value of fuel 19,000 B.t.u. Fuel consumption 0.46 lb. per b.h.p.-hr. Thermal efficiency 29 per cent.

8 A diagram from a De La Vergne engine, type FH, 1915, is shown in Fig. 7. Cylinder diameter of this engine 17 in. and stroke 27½ in. Initial pressure 475 lb. per sq. in., exhaust pressure 30 lb., compression pressure 260 lb. and m.e.p. 82 lb. Engine develops 100 b.h.p. at 200 r.p.m. Fuel consumption 0.450 lb. per b.h.p.-hr. of California crude oil, having a heating value of 18,500 B.t.u. Thermal efficiency 30.5 per cent.

9 Fig. 8 is taken from a Diesel engine, type 1915, cylinder diameter 18.875 in. and stroke 28.375 in. Initial pressure 550 lb. sq. in., exhaust pressure 40 lb., compression pressure 550 lb. and m.e.p. 95 lb. Heating value of fuel 19,266 B.t.u., and fuel consumption 0.407 lb. per b.h.p.-hr. Thermal efficiency 32.5 per cent.

10 It will be noted from these diagrams that the thermal efficiency, 12.8 per cent in 1888, has through improvement in design and by increased compression pressures in the later engines been increased to 30.5 per cent in the hot-surface type of oil engine. In the Diesel type it is now 32.5 per cent. All calculations are on the basis of brake or actual horsepower and not indicated horsepower.

TYPES OF ENGINES

11 Previous to about 1892 all oil engines were of either the hot surface or electric ignition type, with the fuel injected during either the first outward stroke of the piston or air inlet period of the cycle. In Fig. 9 are shown the periods of injection in the different types referred to.

12 A description of the Diesel system was first published about 1892. In the Diesel cycle the fuel injection period did not take place until compression was completed, or nearly so, and the compression pressure was carried to a point where a temperature sufficient to cause ignition was obtained. High pressure (about 1000 lb.) air was in-

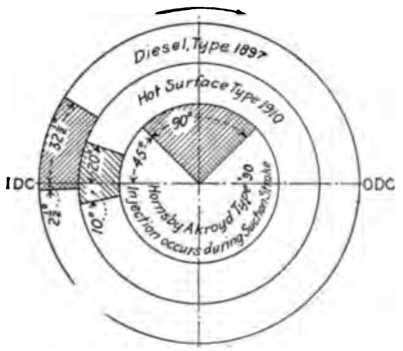


Fig. 9 Periods of Injection of Fuel

jected with the fuel, thoroughly atomizing or pulverizing it, and thus sufficient pressure to overcome that existing in the combustion space at the time of injection was obtained. This system gave in 1892 the highest thermal efficiency, as it does today.

13 Realizing that moderate sized oil engines of from about 75 h.p. to about 400 h.p. and a thermal efficiency somewhat less than that attained in the Diesel type and with lower range of pressures could be produced with some advantages and at less expense, various manufacturers in Europe and in this country have built so-called semi-Diesel or hot-surface type engines, embodying the Diesel method of fuel injection, but in some cases without the air blast.

14 Regardless of their cycles of operations, at the present time oil engines may be divided into two classes: *Diesel* and *hot-surface type or semi-Diesel.* With the exception of the engine shown in Fig. 16, all engines here illustrated are of the 4-cycle type. The majority of manufacturers in Europe and in this country are building 4-cycle

types both for stationary and marine purposes. With its comparative simplicity, the 2-cycle engine may have decided advantages in the smaller sizes. In the larger sizes the advantage for stationary service is reduced cost of manufacture, while for marine service, reduced weight and space occupied is also claimed by the advocates of the 2-cycle type. From cylinders of the same dimensions the power is about 70 per cent greater and the fuel consumption is regarded as approximately 10 per cent greater. Accumulated heat in cylinder head and other parts surrounding the combustion space and main crankshaft bearing troubles have, however, given some European builders considerable difficulty with the 2-cycle type in the larger sizes.

15 Figs. 19 and 20 show the combustion space, the fuel inlet and

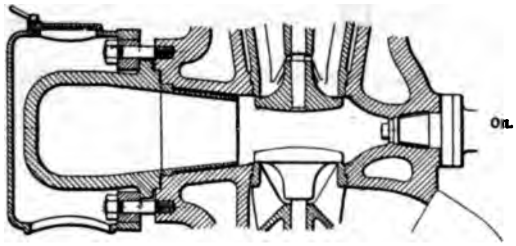


FIG. 10 CROSS-SECTION OF DE LA VERGNE OIL ENGINE

the arrangement of air inlet and exhaust valves of representative Diesel engines.

16 Fig. 10 is a section of the combustion space of the hot-surface type engine where the fuel and high-pressure air are not injected into the combustion space, as in Figs. 19 and 20, but are first forced into a vaporizing chamber which is heated before starting and in which the temperature is maintained by the combustion of the fuel in it.

17 The combustion space of another hot-surface type engine in which somewhat of the same system of operation as in the previous engine obtains is illustrated in Fig. 11,[1] but here the air is not injected with the fuel as in the engine in Fig. 10 but a mechanical sprayer or pulverizer is used and a slight amount of water is also allowed to enter the chamber in addition to the fuel.

[1]Shown by permission of Messrs. Iliffe and Co., London, Eng.

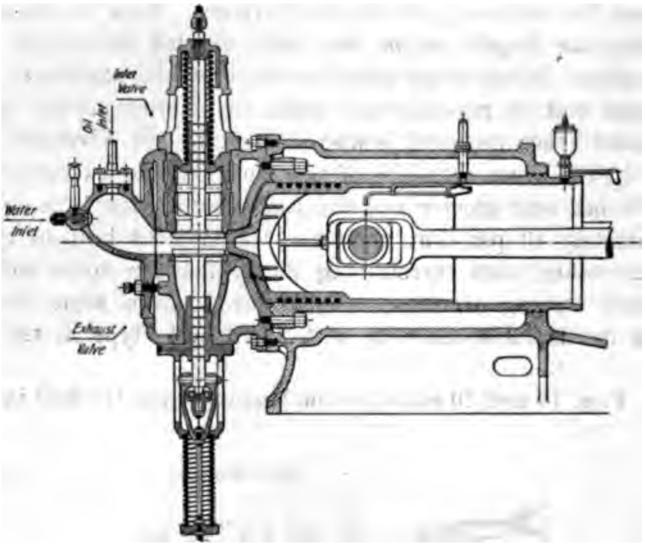


FIG. 11 CROSS-SECTION OF RUSTON PROCTOR OIL ENGINE

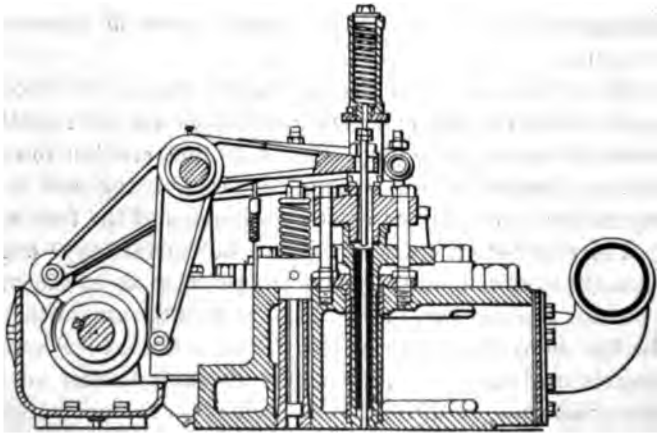


FIG. 12 SECTION OF WILLANS-ROBINSON CYLINDER HEAD

DESIGN

18 The general construction of both Diesel and hot-surface types of engines is shown in the different illustrations. Examination of the different details of construction of the very large number of Diesel and other oil engines built in different countries of the world shows many interesting designs.

19 It would not be feasible here to refer to all the different details of construction. Important elements of large modern oil engines are:

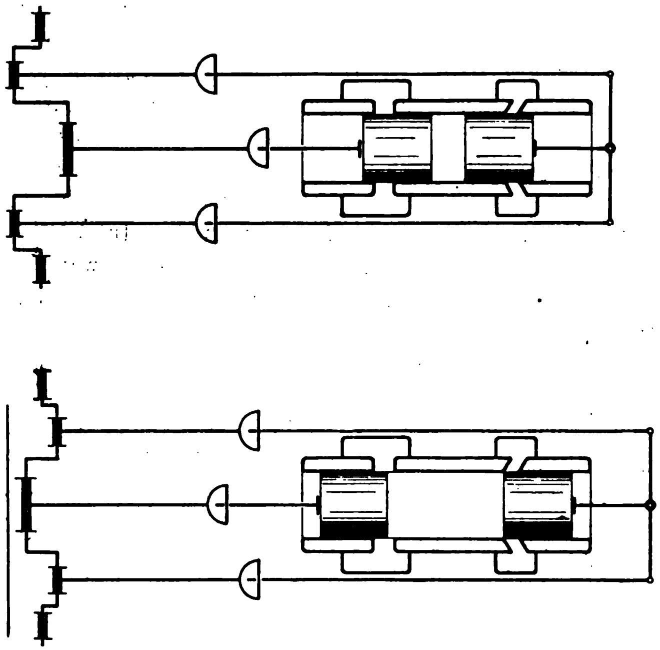


FIG. 13 ARRANGEMENT OF JUNKERS OIL ENGINE

a Cylinder heads,
b Trunk piston or crosshead and shorter piston,
c Sprayer or pulverizer.

20 *Cylinder Head.* Considerable difficulty was in the past experienced with the cylinder heads of Diesel as well as of hot-surface type engines due to fracture largely attributable to high temperatures and pressures obtaining in the combustion space. In recent years this difficulty has been largely or entirely overcome. In some cases the reason for this trouble was improper casting, the cores shifting and

causing unequal thicknesses of the walls of the cylinder head. Amended and improved design in modern engines has, in most types, eliminated this trouble. The heads are now made of such design and material that uniform thickness of the walls is obtained, and the material (soft charcoal iron or its equivalent) is such as to allow easy and uniform expansion and contraction.

21 Various designs of cylinder heads, among which the following are of interest, have been made to simplify the castings: Fig. 12[1] shows the cylinder head of the engine made by Willans-Robinson, Rugby, England, and by the Dow Steam Pump Co. in California. This is a notable design and with it the strains due to contraction in the foundry are reduced. In this cylinder head recesses are only made for the insertion of the air inlet and exhaust valves. The fuel inlet valve is inserted in a tube which has different diameters at its ends and which is pressed into place between the lower and upper facings of the walls of the cylinder head. The starting air valve is similarly inserted in a separate tube. No recess is cored in the cylinder head for these valves and in this way the shape of the castings is simplified and unequal masses of metal are eliminated.

22 A part section of the Junkers type of Diesel engine in which the cylinder head is eliminated entirely is shown in Fig. 13. This engine is equipped with two pistons in each cylinder, that farther from the crankshaft moving in the opposite direction to that nearer the crankshaft and both being connected by suitable crosshead and connecting rods to the outside cranks at 180 deg. from the main crank. In this type, the fuel is injected into the combustion space formed between the two pistons.

23 *Piston and Crosshead.* The trunk piston without crosshead is shown in Figs. 19 and 20. In 4-cycle, single-acting engines developing over 150 h.p. in one cylinder a crosshead is recommended. The advantages of the crosshead type may thus be stated:

(1) The guides in which the crosshead moves are maintained at an even temperature and are not subject to the expansion and contraction of the cylinder that they are with the trunk piston.

(2) It is simpler to lubricate the crosshead pin than the trunk

[1]Figs. 12, 15, 19 and 20 are reproduced by special permission of Messrs. Spon and Chamberlain, publishers of "Marine and Stationary Diesel Engines," by the author.

piston pin, as the former does not come in contact with the heated parts of the engine.

(3) The guides of the crosshead type can be more easily adjusted whereas the ordinary trunk piston does not allow of this adjustment.

24 Advocates of the trunk piston in preference to the crosshead type point out that the wear in the cylinder is due to the piston rings and not to the friction of the trunk piston, that in a well designed engine the pressure of the piston on the cylinder walls is so slight, its proper lubrication can easily be maintained and that the crosshead type requires more space and is more expensive to manufacture.

25 Fig. 14 shows the type of piston made by the M. A. N. in Germany. This is equipped with loose strips on its upper surface

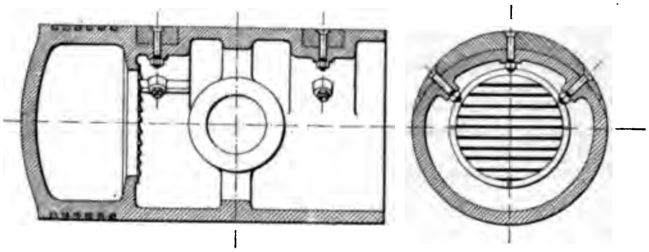


Fig. 14 Section of M. A. N. Piston

which permit adjustment to take up the wear on the piston. The makers of this type claim for it the advantages of a crosshead without the disadvantage of greater cost of manufacture and the occupation of more space.

26 *Sprayers or Pulverizers.* A most important feature of all oil engines is the sprayer or pulverizer through which the fuel and high-pressure air are injected into the combustion space. Its function is to thoroughly atomize the fuel and mix the particles with the air before the latter enters or as it enters the combustion space.

27 Where a heavy crude oil or tar is the fuel, a slight amount of lighter fuel is used in addition. The sprayer is then equipped with two openings and passages properly arranged so that the lighter fuel enters the combustion space and ignites before the heavier fuel enters. The temperature is raised and the ignition of the heavier fuel is facilitated. Separate fuel pumps are used for each fuel.

28 An interesting fuel inlet valve is that used by the Société des Moteurs Sabathé with their engine for submarine use; this valve is

shown in section in Fig. 15. It is equipped with two valves, one of which is the ordinary type of fuel inlet valve and the second is larger in diameter and loose on the former, being held in place by the spring as shown. This fuel inlet arrangement is designed to effect "mixed combustion" by allowing two periods of fuel injection. Its principle is as follows: The fuel injection from the first valve is arranged to enter the combustion space when the compression is about 450 lb. and the volume is constant, with the result that the pressure instantly rises. Immediately afterwards the injection of fuel from the upper passage hitherto held in check by the larger valve is allowed to slowly

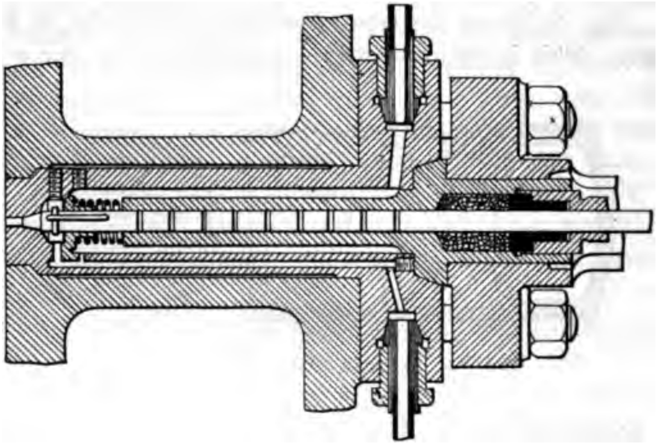


FIG. 15 SECTION OF SABATHE SPRAYER

enter the combustion space while the volume is increasing. The movement of the valves is, of course, mechanically controlled and the timing of injection can be altered to suit the requirements of varying speeds. Economy of injection air as well as greater efficiency is claimed for this type.

RELIABILITY AND ECONOMY

29 One of the most important questions of the operation of oil engines and one frequently discussed and referred to by those who are not fully informed on the subject or who are seeking information, is in regard to reliability of operation. The most forcible reply is that many oil engines have operated in different installations, some for three months, some for six and others for eight months continuously, day and night without stoppage, and with different crude oils.

30 A mining plant in Arizona which is operating continuously 24 hr. per day, is equipped with four hot-surface type engines, two of 180 b.h.p., one 250 b.h.p. and one 280 b.h.p., belted to shafting or direct connected to electric generators. The engines are installed at an altitude of about 7000 ft.; they are 90 miles from a railroad station and oil fuel and supplies have to be hauled this distance over mountain roads. The cost of hauling is approximately one cent a pound, which accounts for the high cost of fuel, viz.: 16.5 cents per gallon. The engines operate on fuel oil of 22-deg. Baumé. Table II shows power generation costs of the same installation for the year 1914.

31 The advantage of the oil engine for operating almost any class of machinery in comparison with other prime movers is conceded, but for its economy to be fully realized the load factor should be as high as possible. A combined installation, where the oil engine operates both a refrigerating or ice plant and also an electric generator furnishing current for light and power, forms an ideal installation where this condition is realized.

TABLE 1 OPERATION OF FOUR OIL ENGINES DURING 1914

Engine No.	Desired period of operation, hr.	Actual period of operation, hr.	Per cent of desired period
1	6803.80	6684.54	98.25
2 (spare unit)	....	700.47	.....
3	8556.75	8411.55	98.30
4	8626	8457.47	98.05

EXAMPLES OF ENGINES

32 Fig. 16 shows a 250-h.p. twin-cylinder 2-cycle Snow Diesel horizontal oil engine, direct connected by flexible coupling to a 60-cycle alternator and operating with California and Mexican crude oils; this engine has been in operation about six months. Fig. 17 shows a 60-h.p. single cylinder 4-cycle horizontal Diesel engine by the same maker, operating with 18-deg. Baumé California crude oil and driving by belt a deep well pump. This plant has been running about four months.

33 In Fig. 18 is shown a 280-h.p. FH De La Vergne oil engine operating an ice machine and electric generator by belt and using California crude oil, 14-deg. Baumé. This installation is at Oakland, Cal., and has been in operation for about one year. The manufacturers of

TABLE 2 POWER GENERATION COSTS FOR 1914

Estimated output		Oil consumed				Cost of operation, dollars							
				Lb. per		Labor		Materials and supplies					Total cost dollars
At switchboard kw-hr.	At engines h.p.-hr.	Gal.	Lb.	Kw-hr. switch-board	H.p.-hr. engines	Operat-ing	Main-tenance	Fuel oil 16½c. gal.	Lubricating oil 71c. per 1000 h.p.-hr.	Repair parts	Belts	Misc. supplies	
2,469,293	3,996,196	278,595	2,061,602	0.84	0.52	7907.31	2211.76	46603.62	2838 82	1659.50	792.92	1044.64	63058.57
Per kw-hr.	. . . .	0.113	0.835	. . . .	. . . .	0.0032	0.0009	0.0188	0.0011	0.0007	0.0008	.	0.0255
Per h.p.-hr.	. . . .	0.083	0.516	. . . .	. . . .	0.0020	0.0006	0.0116	0.0007	0.0004	0.0005	. . .	0.0158
								Fuel oil 2¼c. gal.	Lubricating oil 35c. per 1000 h.p.-hr.				
	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	5969.29	1400.00	. . . .	. . .	. .	20985.42
Per kw-hr.	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	0.0024	0.00057	. . . .	. . . .	. . .	0.0086
Per hp-hr.	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	. . . .	0.0015	0.00035	. . . .	. . . .	.	0.0053

Based on 8640 hr. per year, that is with fuel at 16.5c. per gallon and lubricating oil 71c. per 1000 h.p.-hr equivalent to:

$136.51 per h.p. year at the engines
164.41 per kw. year at the switchboard

Or with fuel oil at 2¼c. per gallon and lubricant at 35c. per 1000 h.p. hours equivalent to:

$45.70 per h.p. year at the engines
74.50 per kw. year at the switchboard

The last named figures are given showing costs under ordinary conditions. Allowance must be made for increased cost of materials and supplies due to excessive charge for transportation.

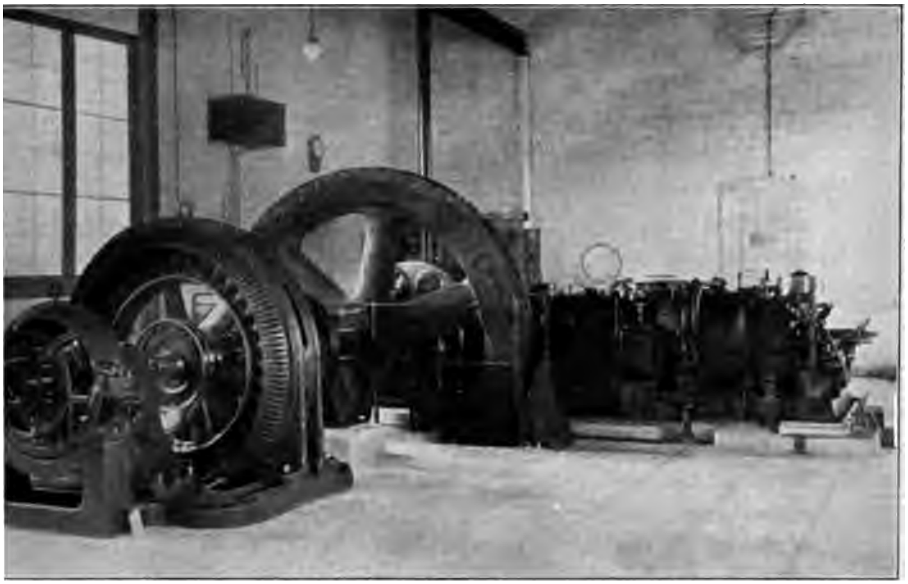


FIG. 16 250-H.P. 2-CYCLE SNOW OIL ENGINE

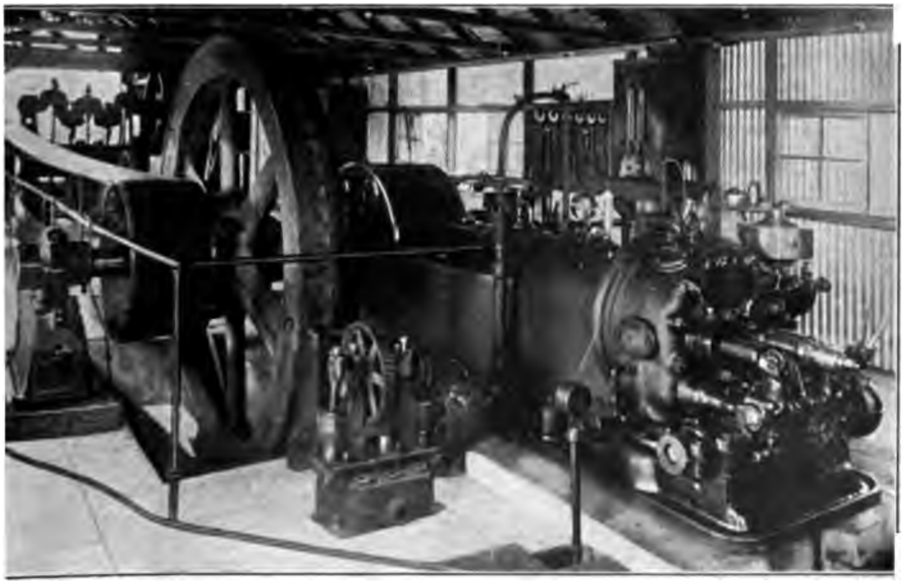


FIG. 17 60-H.P. 4-CYCLE SNOW OIL ENGINE

this engine guarantee it to operate on any fuel or crude oil produced in the United States and Mexico having 18,000 B.t.u. and not more than 1 per cent water. This engine has operated for 800 hr. continuously without stopping. Between Nov. 23, 1914 and July 23, 1915, the plant was in operation 5400 hr. during which period 12,330 tons of ice were manufactured. The fuel consumption per ton of ice varies in different localities and climates and in different installations according to the arrangement of auxiliaries and their method of operation. Between 4 and 5 gal. of oil is in most cases sufficient to produce one ton of ice even during the summer months.

34 Fig. 19 shows a partial sectional view of the McIntosh-Seymour enclosed type vertical 4-cycle 4-cylinder single acting stationary Diesel engine of 500 h.p., with 2-stage air compressor for furnishing high pressure injection air placed in line with the motor cylinders and operated, as is now standard practice, from an overhung crank on the main crankshaft. This drawing shows one of the most recently developed Diesel vertical engines in this country. The details of design, such as cooling, pulverizing of the fuel, valve motion, etc., have had careful attention. While the illustration shows the enclosed crank case type of engine, the latter is also made with "A" frame construction.

35 The Burmeister and Wain Diesel marine 4-cycle single acting type engine built by this firm in Copenhagen, Denmark, and installed in the latest motor ships made by them is shown in Fig. 20. Each engine has six cylinders, 29 9/64-in diameter and 43 5/16-in. stroke, and develops 2000 i.h.p. at 100 r.p.m. The engine is reversible by longitudinal movement of the camshaft in the regular way.

36 There are many interesting features in this engine, notably the spray valve which has two coned surfaces, one forming the valve seat and the other spreading the spray. It opens outward towards the piston, instead of inwards as in most types; in this way the spray of fuel is very evenly distributed throughout the whole of the combustion space and the heat evolved during combustion is distributed over the whole area of the piston. The process of starting and maneuvering is simplified by the starting valve which is automatically operated by the pressure of the air and only requires the opening of one valve. In this type of engine the camshaft is operated from the crankshaft by a chain of spur gears, which system has replaced the vertical intermediate shaft and gearing previously used.

37 This design is considered one of the most successful marine *engines in large* sizes and several motor ships equipped with it have

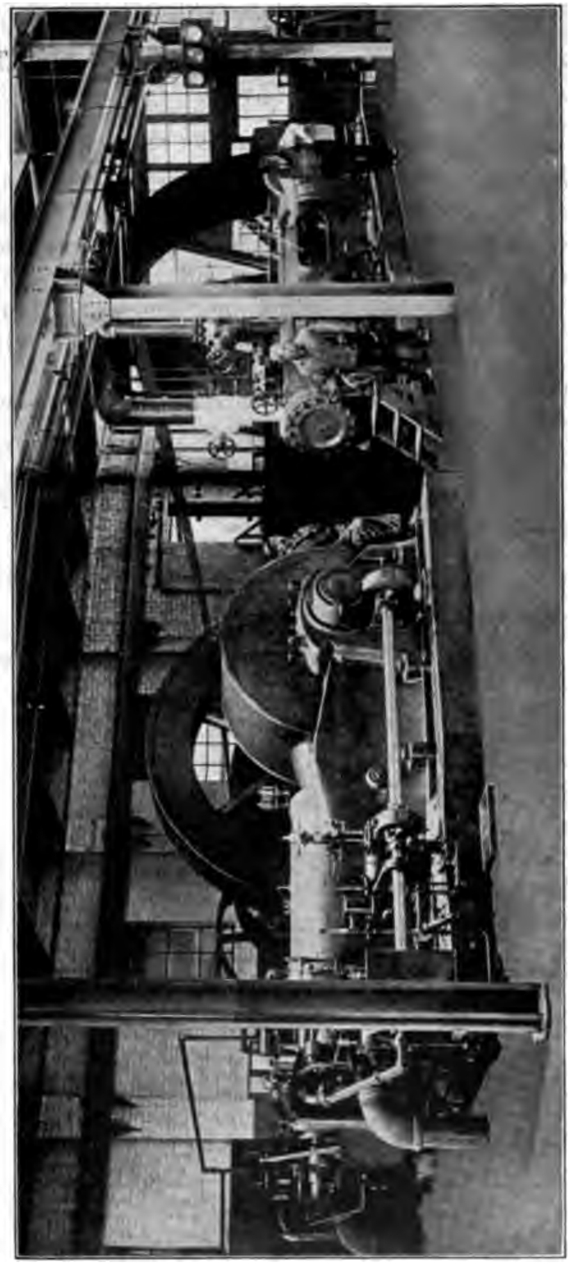


FIG. 18 DE LA VERGNE OIL ENGINE AT OAKLAND, CAL.

made long voyages. In a paper by I. Knudsen, read in July, 1914, at Malmo, Sweden, the figures in Table 3 showing particulars of a long voyage of one of these motor ships in comparison with a steamship of the same dimensions were given.

OIL ENGINES IN PIPE LINE SERVICE

38 Fig. 21 shows a sectional view of the latest design of De La Vergne 150-h.p. single and 300-h.p. twin-cylinder oil engine. In this construction the valve motion consists of the camshaft geared to the crankshaft in the ordinary way, but this intermediate shaft actuates a second shaft placed behind the cylinder head and operating parallel to the crankshaft. The air and the exhaust valves are so

TABLE 3 COMPARISON OF DIESEL MOTOR SHIP AND STEAMSHIP

	Steamship "Kina" (Single screw)	Motor ship "Siam" (Twin screw)
Length between perpendiculars, ft.	385	410
Breadth, ft.	53	55
Depth moulded	26 ft. 10¾ in.	30 ft. 6 in.
Cargo, tons	7673	8670
Distance covered, miles	27808	27818
Fuel, tons	4858.6, coal	1120.2, oil
Cost of fuel, dollars	5.25 (22 sh)	7.25 (35 sh) oil
Fuel cost, dollars, 1000 tons cargo 1 mile at 11 knots.	13.6 (6.08 d)	4.0 (2 d)
Fuel cost, dollars, 8500 tons 27.818 miles at 11 knots.	28.702	9.400
Saving in favor of motor ship, dollars	19.302	

Circuit of Europe, East Asia and return.
Extract from paper read by Mr. I. Knudsen, Malmo, Sweden, July, 1914.

arranged that they can be easily removed and the piston is made of greater length so as to reduce the pressure and minimize the amount of wear between it and the cylinder walls. Details of lubrication of all bearings and other moving parts have been improved. The piston pin is lubricated through the hollow connecting rod.

39 In 1902, the writer arranged and introduced the oil engine for oil pipe line pumping service. A number of engines for this service were first installed for the Gulf Pipe Line Company in Texas and since then the installation of several hundred engines with many of the leading pipe line companies in the south or southwest territories has followed. These pumping stations consist of several units, some stations having as many as six engines. The capacity of each unit varies from 90 to 150 h.p. and is composed of the engine direct connected to a vertical or horizontal power pump, as shown in Fig. 22.

The former installations were equipped with friction clutch couplings between the engines and the pumps. In the later installations the connection used between engines and pumps is a flexible coupling or,

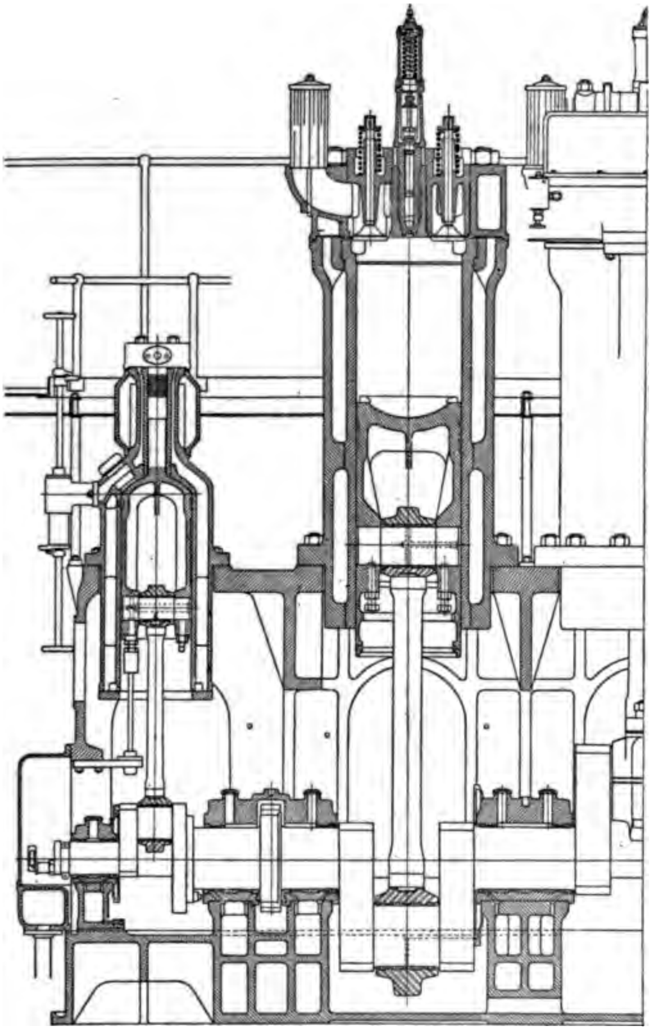


FIG. 19 SECTION OF McINTOSH AND SEYMOUR OIL ENGINE

in some cases, a rigid coupling. The pump is furnished with a by-pass between the suction and discharge so that, in starting the engine, the load is very much reduced. Fig. 22 shows the latest arrangement of

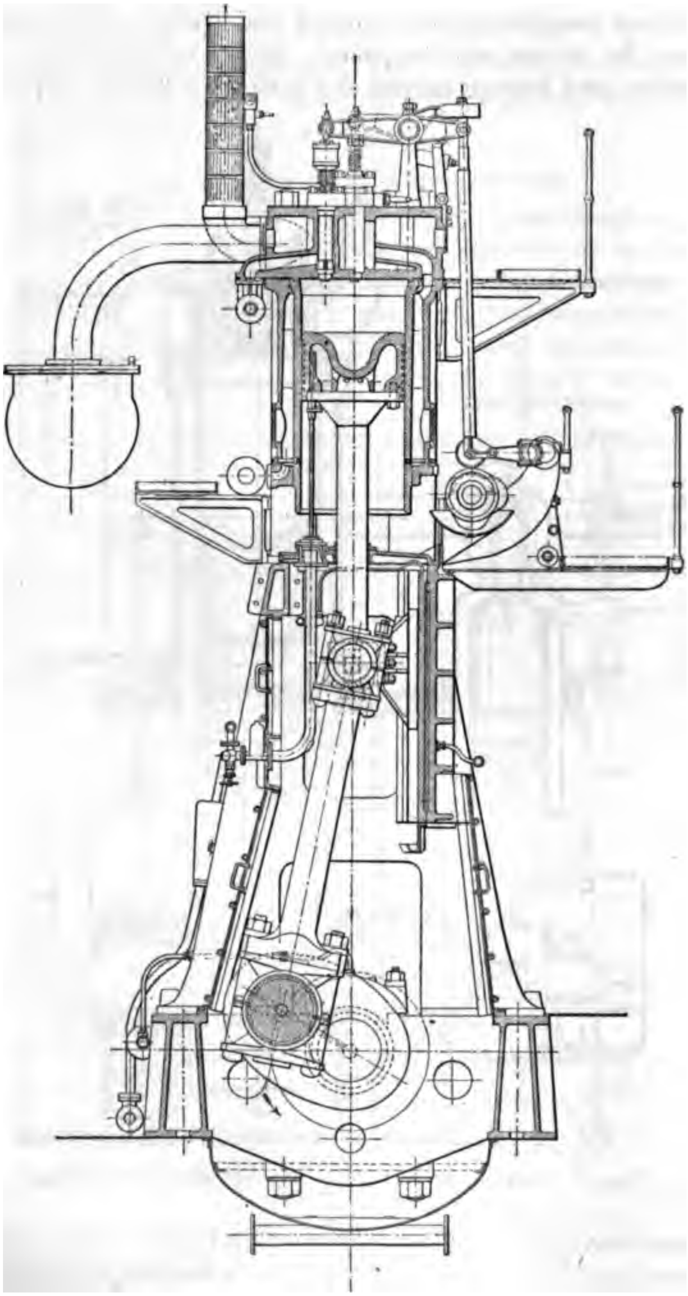


FIG. 20 SECTION OF BURMEISTER AND WAIN OIL ENGINE

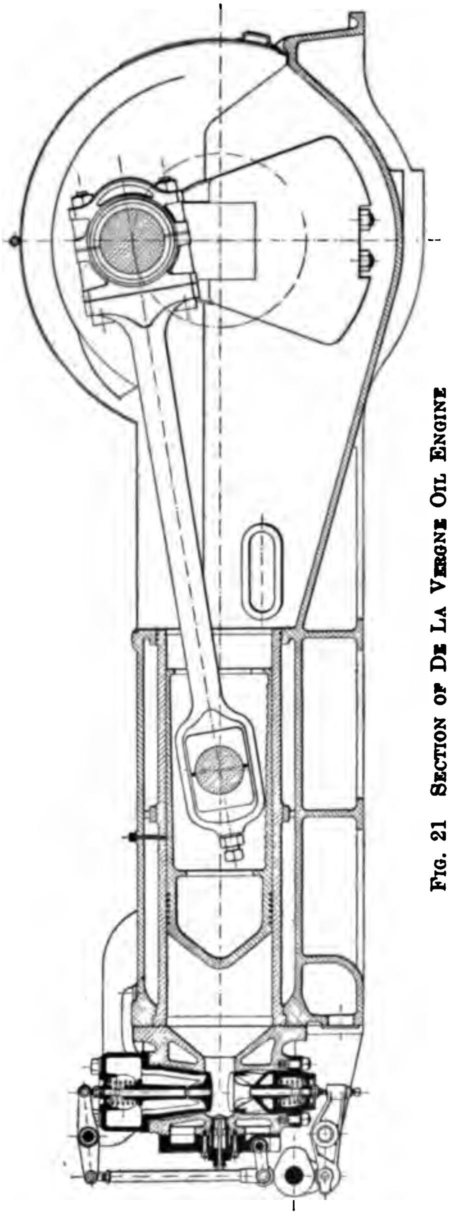


FIG. 21 SECTION OF DE LA VERGNE OIL ENGINE

TABLE 4 COMPARISON OF ENGINE FUELS

Maker	Union Oil Co.	Union Oil Co.	Std. Oil Co.	Std. Oil Co.	Union Oil Co	General Petroleum Co. National Petroleum Co.		
Trade name	California crude	Diesel oil	Calol fuel oil	Star fuel oil	Stove oil	Kellogs stove dist. hvy. slush tops	Kellogs No. 2 tops dist.	No. 1 water white dist.
Gravity at 60 deg. in deg. Baumé	14– 18	23.5– 24	24	27.2	28	36–40	43–45	48–51
Flashpoint (Abel - Pentsky) burning pt. closed cup deg. fahr.	150–175	175–185	150	190	175	100	50	25-30
	175–225	......	......	225	210	125	65–75	40–50
Flashpoint open cup burning	......	195–205	......	210	200	120	63	30–35
pt., deg. fahr.	......	235–245	......	243	225	140	72	50–60
B.t.u. per lb.	18,500	18,950–19,250	19,000	19,200	19,300	18,000–19,000	18,000–19,000	18,000–19,000
Sulphur, per cent	usually under 1	0.75	0.75	0.02	0.02	0.02	0.02	0.02
Asphalt, per cent	50	20	25	20	10–20	0.10	0.05	0.001
Water, per cent	usually under 2	0.5	......	0.08	Trace	Trace	None	None
Residue	......	......	......	at 300 deg. fahr. 3 per cent	......	....	at 500 deg. fahr. less than 1 per cent	......

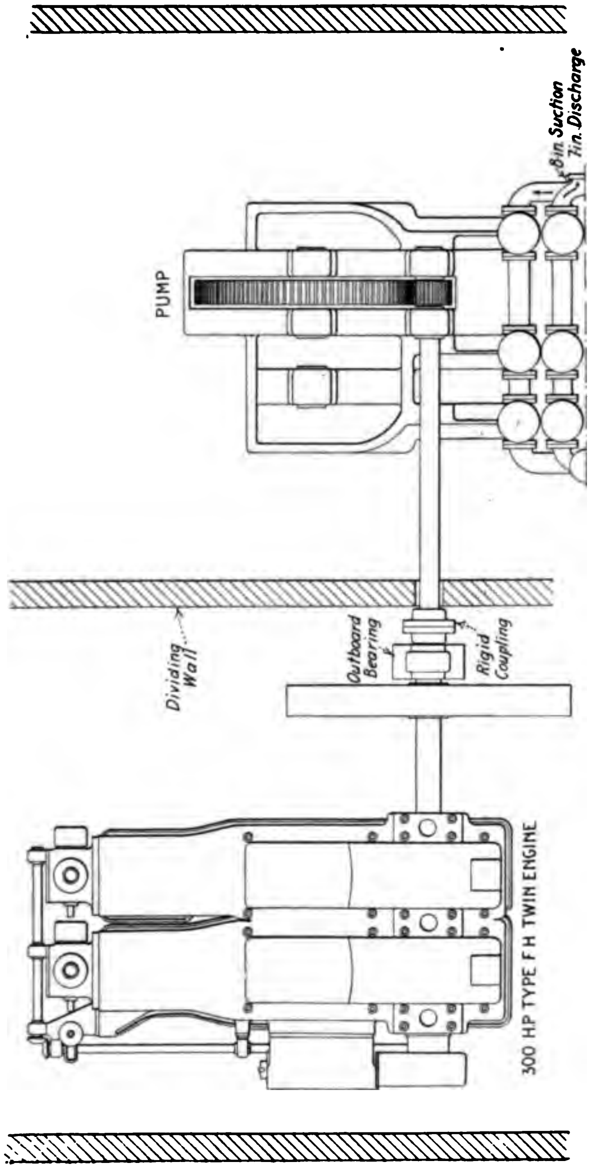


FIG. 22 DE LA VERGNE OIL ENGINE AND PUMP

this pumping unit where the oil engine flywheel is placed close to the out-board bearing. In this way the main bearings are relieved of this weight and unequal wear on the bearings is avoided.

40 This type of pumping outfit was fully described in a paper[1] before the Society by Forrest M. Towl, where it was stated the total efficiency of an 85-h.p. engine in three tests was 26.8, 27.75 and 27.52 per cent, respectively. The efficiency of pump and transmission was stated as 92.1 per cent. The fuel consumption per pump horsepower by displacement was 0.5171 lb. Tests were made with 33-deg. Baumé fuel having a heat value of 19,059 B.t.u. per pound.

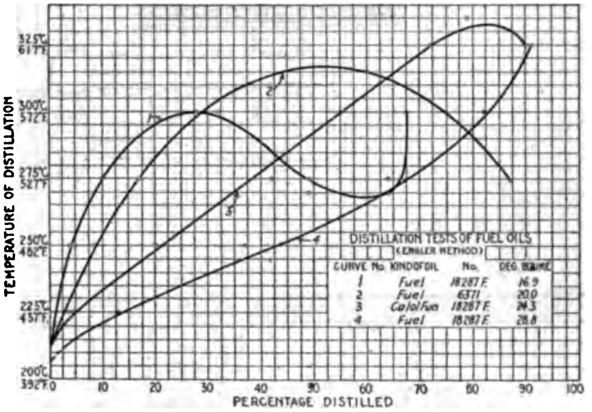


FIG. 23 CURVES SHOWING DISTILLATION TESTS OF FUELS

CALIFORNIA FUELS

41 The oil engine for pipe line service has not yet been used to any extent in the California oil fields. Table 4[2] shows the characteristics of California crude oils, while Fig. 23 gives results of distillation tests. It is necessary to heat such to a temperature that they will flow readily in the pipe lines and can be satisfactorily handled by the pump. The amount of heat necessary to raise the temperature of crude oil of 15-deg. Baumé, assuming its specific heat to be 0.333 (which value seems to be difficult to exactly determine), is shown by the following remarks.

[1] Trans. Am. Soc. M. E., vol. 33, page 905.

[2] Table 4 was partly compiled by Smith-Booth-Usher Co. and its insertion is due to their courtesy. Fig. 23 is shown by permission of the Standard Oil Co. of California.

TABLE 5 COST OF ENGINE OPERATION

Size of unit	Type of engine	b.h.p. per gal. fuel	Fuel cost per gal. Cents	First cost per b.h.p. Dollars	Total first cost Dollars	Yearly fixed 20 per cent charge, Dollars	Fuel cost 24 hr. per day for 100 days, Dollars	150	200	250 days	Total yearly charge, fuel plus fixed charges 100 days, Dollars	150	200	250 days
50 h.p.	Distillates	10	5	25	1250	250	600	900	1200	1500	850	1150	1450	1750
	Tops distillates	10	$2\frac{3}{4}$	25	1250	250	330	495	660	825	580	745	910	1075
	Semi-Diesel	10	$2\frac{1}{7}$	60	3000	600	257	385	514	624	857	985	1114	1242
	Diesel	16	$2\frac{1}{7}$	75	3750	750	160	241	321	401	910	991	1071	1151
100 h.p.	Distillate	10	5	30	3000	600	1200	1800	2400	3000	1800	2400	3000	3600
	Tops	10	$2\frac{3}{4}$	30	3000	600	660	990	1320	1650	1260	1590	1920	2250
	Semi-Diesel	10	$2\frac{1}{7}$	55	5500	1100	514	770	1028	1284	1614	1870	2128	2384
	Hot surface-high economy	16	$2\frac{1}{7}$	65	6500	1300	320	482	642	802	1620	1782	1942	2142
	Diesel	16	$2\frac{1}{7}$	75	7500	1500	320	482	642	802	1820	1982	2142	2302
150 h.p.	Distillate	10	5	30	4500	900	1800	2700	3600	4500	2700	3600	4500	5400
	Tops	10	$2\frac{3}{4}$	30	4500	900	990	1485	1980	2475	1890	2385	2880	3375
	Semi-Diesel	10	$2\frac{1}{7}$	50	7500	1500	771	1155	1542	1926	2271	2655	3042	3426
	Hot surface-high economy	16	$2\frac{1}{7}$	65	9750	1950	480	723	962	1203	2430	2673	2912	3153
	Diesel	16	$2\frac{1}{7}$	70	10500	2100	480	723	962	1203	2580	2823	3062	3303
250 h.p.	Distillate	10	5	30	7500	1500	3000	4500	6000	7500	4500	6000	7500	9000
	Tops	10	$2\frac{3}{4}$	30	7500	1500	1650	2475	3300	4125	3150	3975	4800	5625
	Semi-Diesel	10	$2\frac{1}{7}$	50	12500	2500	1285	1925	2570	3210	3785	4425	5070	5710
	Hot surface-high economy	16	$2\frac{1}{7}$	60	15000	3000	800	1202	1605	2005	3800	4202	4605	5005
	Diesel	16	$2\frac{1}{7}$	65	16250	3250	800	1202	1605	2005	4050	4452	4854	5255

Yearly fixed charge is arrived at as follows: interest, 6 per cent; taxes and insurance, 1 per cent; repairs, 3 per cent; depreciation, 10 per cent.

Engine using distillate 48–51 deg. B. oil has a thermal efficiency of 20 per cent under full load.
Engine using tops distillate 38–42 deg. B. oil has a thermal efficiency of 20 per cent under full load.
Semi-Diesel engine using 24–28 deg. B. oil has a thermal efficiency of 18 per cent under full load.
Hot surface high economy engine using 16 deg. B. oil has a thermal efficiency of 27 per cent under full load.
Diesel engine using 18 deg. B. oil has a thermal efficiency of 28.4 per cent under full load.

42 The amount of oil pumped per b.h.p.-hr. at a pressure of 570 lb. is 135 gal., and the heat required to raise this quantity of oil through a range of 41 deg., or from a temperature of 69 deg. to 110 deg. fahr., would be 18,400 B.t.u. The amount of waste heat from an oil engine consuming half a pound of fuel per actual h.p.-hr., both from the waste heat of the water jacket and that recoverable from the exhaust, with the most advantageous arrangement, is approximately 4200 B.t.u. Thus, taking this waste heat, it is evident that 14,200 B.t.u. per b.h.p. of the pumping outfit per hour would have to be furnished from an outside source to provide sufficient heat to raise the oil to the required temperature.

43 In a steam pumping plant this can be advantageously taken from the exhaust steam of the steam plant. The fuel consumption of the average steam pumping plant may be taken as 1.5 lb. of oil fuel per b.h.p.-hr., but sufficient heat is in this case also available for heating the crude oil to the temperature above referred to.

44 With the oil engine plant, 0.5 lb. of fuel per b.h.p.-hr. is required for the actual pumping process and from the figures above it will be seen that an amount of heating equivalent to that developed from a pound of oil would be necessary to heat the oil passing through the pipe line. Thus the total fuel consumption of the oil engine and that of the steam pump, allowing for heating, is approximately the same.

45 The oil engine outfit can operate with a poor quality of cooling water which would, however, be unsatisfactory for boiler use. This is a great advantage in favor of the former in many localities where only a poor quality of water can be procured.

46 Table 5[1] shows the costs of installation and operation of the different types of engines specified prevailing in California in 1914. The price quoted for each fuel may not now be correct and the costs of fuel may require slight modification to conform with prevailing prices. The writer is informed that the price for 48 to 50-deg. Baumé distillate is 6 cents per gal. in 110-gal. drums, while Diesel fuel oil is quoted at 85 cents per 42-gal. barrel in tank cars 25 miles north of San Francisco. Calol fuel oil 24-deg. Baumé is 75 cents per 42 gal. and fuel oil not less than 14-deg. Baumé is 60 cents per barrel in tank cars f.o.b. Richmond, Cal. The yearly fixed charge

[1]Table 5 was compiled by Messrs. Smith-Booth-Usher Co. in 1914. It is due to their courtesy that it is shown. Data regarding hot surface engine have been added.

of 20 per cent is a higher rate than is usual to allow in the Eastern States where 5 per cent interest and 5 per cent depreciation are considered sufficient.

DISCUSSION

H. R. SETZ noticed that in Table 5 the depreciation has been taken as ten per cent. He would recommend a depreciation of about five and certainly not more than six per cent in trying to arrive at the cost of the power produced in a Diesel engine, including the capital charges. It is doing the Diesel engine injustice if ten per cent is figured, because, at this rate, in less than eight years the whole engine will have been depreciated from its original purchase value to nothing; whereas, after an operating period of eight years a high grade engine of this type can for less than one-third of its original purchase price be put in new condition again.

Another reason for recommending a lower rate than ten per cent for depreciation is the fact that there is no new form of prime mover in sight which might possibly render the Diesel engine obsolete for many years to come.

No. 1494

THE STRENGTH OF GEAR TEETH

By Guido H. Marx and Lawrence E. Cutter, Palo Alto, Cal.
Members of the Society

(Second Paper)

The investigations reported upon in this paper were undertaken for the purpose of supplementing those presented to the Society at its Annual Meeting of 1912, by the senior author.[1] The methods of conducting the tests, and of mathematical analysis employed in working up the results, were substantially the same as described in the earlier paper and are not repeated here. Paragraph references to the first paper will be used to direct attention to explanatory matter which is here omitted.

2 The limitations of the apparatus available for the earlier experiments made it impossible to secure positive data at pitch speeds exceeding 500 ft. per min., although significant data of a negative character were obtained at speeds up to 1000 ft. per min.[2]

3 In order to get positive data at high pitch speeds, an improved form of the apparatus used in the earlier experiments was devised[3] and connected by chain drive to a 50-h.p. motor capable of taking care of momentary overloads of 100 per cent. The main difference between the new apparatus and the old lay in the employment of Hess-Bright ball-bearings throughout in place of ordinary journal bearings. Each shaft was carried on two of these radial bearings, No. 307.

4 A second improvement was in the prony brake (Fig. 1). This was devised to be self-contained, the friction load due to scooping up the circulating cooling water being weighed on the scales with the rest of the friction load. It worked with great steadiness and proved eminently satisfactory. Owing to the high rim speeds reached, the brake wheel was carefully finished all over and had a web in place of arms. A few holes drilled through the web at its outer circumference permitted the free circulation of the cooling water.

[1] Trans. Am. Soc. M. E., vol. 34; paper 1382, pp. 1323-1398.
[2] Trans. Am. Soc. M. E., vol. 34, p. 1390.
[3] Par. 8 and 9, paper 1382.

Presented at the Panama-Pacific International Exposition Meeting, San Francisco, September 1915, of The American Society of Mechanical Engineers.

OUTLINE OF TESTS

5 The gears which were tested were made by The Brown & Sharpe Mfg. Co. and the Fellows Gear Shaper Co. The investigation divided itself into the following main divisions:

a Tests on 30T meshing with 40T, Brown & Sharpe 14½-deg. involute, at pitch speeds from 500 to 2000 ft. per min. to determine velocity coefficients, *v*, to supplement those derived in the earlier experiments for speeds below 500 ft. per min.[1]

b Tests on 30T meshing with solid 60, 80, 100, and 150T, Brown & Sharpe 14½-deg. involute, to supplement earlier tests on effect of arc of action, to determine more fully arc of action influence.

c Tests on 30T meshing with 40T, Fellows 20-deg. involute

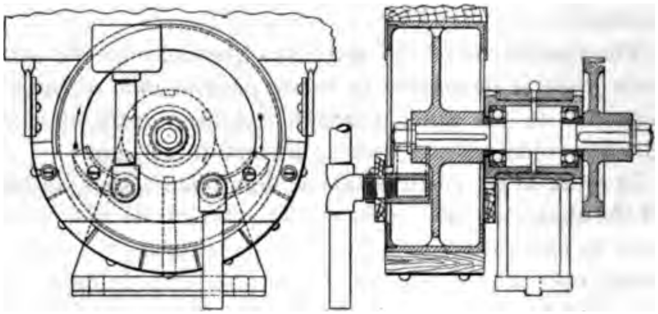


FIG. 1 PRONY BRAKE

stub-tooth, for speeds from zero to 2000 ft. per min. to determine velocity coefficients, *v*, for this form of tooth.

d Tests on 30T meshing with 20, 40, 60, 80, and 100T, Fellows 20-deg. involute solid gears, on effect of arc of action, to determine arc of action coefficients, *a*, for this type of gear.

e Static tests on Fellows gears to determine actual breaking strength of individual teeth, for determination of experimental values of factors for form of tooth as determined by the number of teeth in gear (Lewis's factor *y*). See Par. 28, Paper 1382.

The complete log of the tests will be found in Tables 14, 15 and 16.

6 The Brown & Sharpe gears were of their standard 10 diametral

[1]Par. 2, 7 and 21, paper 1382.

pitch, 14½-deg. involute form and those used in the tests involving the effect of pitch speed (Tests 1–12 inclusive) were sent from stock. These stock gears having a width of face of 1 3/16 in. were reduced by us to the width of 1 1/16 in. for which the apparatus had been constructed. The 30- and 60-tooth gears were solid discs and the others were webbed, in order to eliminate the effect of weakness of arms and rims; the 80-, 100- and 150-tooth gears were made from patterns furnished by the authors. Solid grease lubricant liberally applied was used on all gears.

7 The Fellows gears were all of their 10/12-pitch, 20-deg. involute form, and were solid discs in every case.

8 All gears when tested had the width of face of 1 1/16 in. The 20- and 30-tooth gears (except the 30-tooth gears of Tests 24A and 25A) had a bore of 1 1/16 in., and all others had a bore of 1 5/16 in.

9 The material (cast-iron) was tested in all cases by means of specimens cut from the gears themselves[1] and the results are given in detail in Tables 17 and 18. The gears of each manufacture were furnished in two separate installments. As shown in Tables 17 and 18, the material of the gears furnished for the tests on influence of arc of action (Tests 13–20 and 18A–23A inclusive) was stronger in both cases than that of the gears previously furnished for the tests on influence of pitch speed (Tests 1–12 and 1A–17A inclusive). Where the variation of the strength of material was such as to call for a correction in order to get consistent and comparable results, this was made; but in all cases the actual test results are given first.

SERIES A: TESTS TO DETERMINE VELOCITY COEFFICIENTS, BROWN & SHARPE GEARS

10 Table 3 of velocity coefficients, v, (Par. 21, Paper 1382), for Brown & Sharpe 14½-deg., cast-iron gears may be considered as quite definitely established for the entire range of speeds from zero to 2000 ft. per min. An explanation follows of the method by which these values were derived, from which it would appear to be safe to extrapolate beyond the speed of 2000 ft. per min., if desired,—safer than to make the tests. In fact, the authors are entirely willing to leave the conduct of all tests at higher pitch speeds to those who would like to see them made.

11 The data from which the values in Table 3 were derived are given in Tables 1 and 2 and in Fig. 2. Table 1 gives the results of the supplementary tests on the effect of speed on breaking strength of Brown & Sharpe 14½-deg. involute, 1 1/16-in. face, 30T meshing

[1] Par. 22, paper 1382.

TABLE 1 EFFECT OF SPEED ON BREAKING STRENGTH

Brown & Sharpe 14½-deg. Involute, 10-pitch, 1 $\frac{1}{16}$-in. Face, 30T Meshing with 40T; 1st Series, 1914

Test number	Cast Iron Gears		8-Pitch B. & S. Steel Change Gears			
	Pitch speed, ft. per min.	Equivalent load at teeth, pounds	Number of teeth: Driver	Number of teeth: Driven	Pitch speed	Equivalent maximum pitch load
1	501	1743	100	70	1462	598
2	507	1622	100	70	1479	556
3	872	1441	100	40	1454	864
4	872	1592	100	40	1454	955
5	1221	1289	70	20	1018	1547[1]
6	1163	1380	100	30	1454	1104
7	1176	1259	100	30	1470	1007
8	1163	1350	100	30	1454	1080
9	1724	1078	100	20	1437	1293
10	1734	1047	100	20	1444	1257
11	2021	1199	120	20	1684	1438
12	2057	1017	120	20	1714	1221

[1]Steel gears abraded.

TABLE 2 REDUCTION OF BREAKING LOAD TO BASIS OF UNIFORM MODULUS OF RUPTURE OF 39,000 LB. PER SQ. IN.

Brown & Sharpe Gears

Test number	Actual Modulus of rupture from table Appendix 2	Conditions of Running Test				Equivalent breaking load for modulus of rupture of 39,000 lb. per sq. in.
		Teeth in C. I. driver	Teeth in C. I. driven	Actual breaking load at pitch line by test	Velocity at pitch line, ft. per min.	
1	48,770	30	40	1743	501	1394
2	42,976	30	40	1622	507	1471
3	46,323	30	40	1441	872	1213
4	54,632	30	40	1592	872	1137
5	43,143	30	40	1289	1221	1165
6	49,040	30	40	1380	1163	1097
7	39,288	30	40	1259	1176	1249
8	42,382	30	40	1350	1163	1243
9	43,868	30	40	1078	1724	958
10	38,779	30	40	1047	1734	1053
11	47,400	30	40	1199	2021	987
12	41,300	30	40	1017	2057	961
13	69,600	30	60	2372	501	1329
14	59,460	30	60	2271	504	1489
15	58,840	30	80	1930	507	1279
16	59,300	30	80	2044	504	1344
17	55,850	30	100	1817	510	1269
18	59,840	30	100	2059	507	1342
19	66,620	30	150	1998	507	1170
20	61,100	30	150	1917	507	1224

with 40T. The experiments previously reported[1] for gears of these sizes and this type gave results up to a pitch speed of 415 ft. per min. These results, together with those of Table 1, are plotted in Fig. 2, Curve *A*.

12 In Par. 14 to 17 of Paper 1382, attention was called to an apparent rise of strength at speeds above 300 ft. per min. and the tentative suggestion advanced that this might be due to passing a maximum percussive effect. The tests at the speeds above 500 ft. per min. failed to maintain this view. Moreover, the tests on the Fellows gears reported on in a later section of this paper showed no such phenomenon. In the latter case ball bearings were used throughout for the series of tests involving 30 tooth and 40 tooth gears. The explanation of the irregularity of the Brown & Sharpe curve (including the

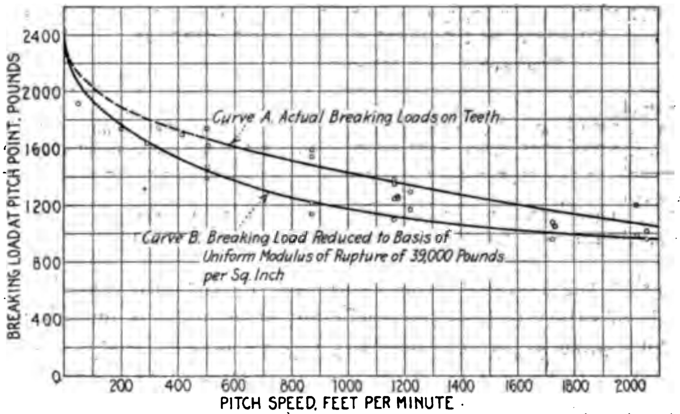


FIG. 2 RELATION BETWEEN PITCH SPEED AND BREAKING STRENGTH OF BROWN & SHARPE 30- AND 40-TOOTH 14½-DEG. INVOLUTE GEARS

phenomenon of its rise) lies in the use of ordinary journal bearings in the first set of experiments. The friction of the brake shaft bearings was neglected. In the original paper it was assumed that the only effect of this is to make the computed breaking load of the teeth a very little less than its real value in each case.

13 A study of journal friction and variations of coefficients of friction with speed, made by the senior author in another connection, clears up this matter. Without entering into detail, the coefficient of friction of the journals used in the earlier apparatus is a variable, being greatest at zero velocity and decreasing to a minimum at about the rotative speed corresponding to the pitch speed of 415 ft. per min.

[1]Par. 15, paper 1382.

At this speed the coefficient of friction (perfect film lubrication) is not widely different from the practically uniform coefficient of friction for ball bearings. Had this variable effect of friction not been neglected, the actual curves would probably not have shown this fall and subsequent rise.

14 If due allowance be made for this frictional resistance and it be properly added to the observed breaking load in each case, the actual curve for the entire range would be approximately that shown by the dotted line up to the speed of 500 ft. per min., and the full line from there on to 2000 ft. per min. (Curve *A*, Fig. 2). A correction was made for the added load on the gear due to journal friction for the test at zero velocity, assuming a value of 0.20 for the static coefficient of friction, giving as a result an added load of 175 lb. on the tooth,

TABLE 3 VELOCITY COEFFICIENTS (v). BREAKING LOAD ON TEETH REDUCED TO BASIS OF UNIFORM MODULUS OF RUPTURE

Brown & Sharpe 14½-deg. Involute Gears

Pitch velocity, ft. per min...	0000	100	200	300	400	500	600	700	800	900	1000	1100
Velocity coef., v.	1.000	0.795	0.730	0.675	0.635	0.595	0.565	0.540	0.520	0.500	0.484	0.470
Pitct velocity, ft. per min...	1200	1300	1400	1500	1600	1700	1800	1900	2000			
Velocity coef., v.	0.455	0.445	0.435	0.430	0.420	0.415	0.410	0.405	0.400			

thus bringing the average value up to 2435 lb. for zero velocity. It is on the side of safety to base the coefficient of velocity, v, on this value.

15 The stock gears used in Tests 1 to 12 showed a wide variation in material as indicated by flexure tests. (See Tables 17 and 18.) Since the material of the similar gears reported upon in Paper 1382 had a very uniform modulus of rupture of about 39,000 lb. per sq. in., it would seem better to reduce the actual results of the present experiments to a basis of a uniform modulus of rupture of 39,000 if the two sets of observations are to be combined. This has been done in the last column of Table 2. Fig. 2, Curve *B*, shows the results graphically. An explanation of this variation of strength probably lies in the reduction of the width of face of these gears after their receipt by us. The removal of one-eighth of an inch at one surface or the other would have a marked effect, since the surface material is recognized as being the stronger. If this material were removed on the side which originally had had the greater depth of cut taken from it in finishing the gear blank, the result would be different from that if the reduction in width had been made on the other surface.

16 Objection may be made to using Curve *B* rather than Curve *A* for the determination of the velocity coefficients. It is to be noted, however, that this reduction, made for the purpose of having the present and earlier experiments more justly comparable, also makes the present test results more consistent and it gives velocity coefficients which, being lower than those which would be derived from Curve *A*, are on the side of additional safety.

SERIES B: TESTS TO DETERMINE INFLUENCE OF ARC OF ACTION, BROWN & SHARPE GEARS

17 The experiments under this head were originally planned (when the gears for the series were ordered) to supplement those on the influence of arc of action, described in Par. 32, Paper 1382, and shown there in Fig. 9, Curve *B*; but subsequently it was decided to run the tests independently of the previous ones, at a pitch speed of approximately 500 ft. per min., in order to make them comparable with the Fellows tests, Nos. 5A, 6A, 18A, to 23A inclusive, Par. 29 to 34 of this paper, which had already been made at this speed. Table 4 gives the results.

18 The unusually high modulus of rupture[1] shown by the material of most of the gears used in this set of tests, made it seem desirable to adopt the expedient of reducing the actual test results to a basis of an imaginary uniform material having a modulus of rupture of 39,000 lb. per sq. in.

19 It is impossible, however, to get any satisfactory light out of this series of tests on the actual influence of the arc of action. While the unmodified results, with the exception of Tests 13 and 14, are not incompatible with those obtained in the earlier paper, when reduced to a basis of a uniform modulus of rupture they show an apparent falling off of breaking strength with increase of arc of action for ratios of arc of action to pitch arc greater than 2. This may actually be the case, but it is contrary to the effect of ratio of arc of action to pitch arc for values ranging from 1 to 2 and does not seem rational. It is also contrary to the results given in Tables 10–14, Paper 1382.

20 It must be borne in mind that reducing the tests to a uniform modulus of rupture on the basis of a test bar cut from the gear is only a crude expedient adopted in default of a better. Cast iron is too variable a material to permit certainty that the strength at the tooth which first yielded was just that of the test bar. In addition, we have the fact that the teeth do not fail by pure flexure and that, therefore, using the modulus of rupture for flexure as the unifying

[1]Tables 17 and 18.

basis is open to some question. But it is equally evident that material which showed as much variation as did this calls for the reduction of test results to some kind of comparable basis.

21 Thinking that the discrepancy of the results might be due to the possible failure of the teeth of the larger gear rather than the smaller, owing to the chance of weaker material in the larger gears, test bars were cut from the 60-tooth and 80-tooth gears of Tests 13 and 15. While the material proved to be a little weaker than that of the corresponding 30-tooth pinions[1] computations showed that this

TABLE 4 TESTS ON INFLUENCE OF ARC OF ACTION

BROWN & SHARPE 14½-DEG. INVOLUTE, 10-PITCH GEARS

Test number	Number of Teeth C. I. Test Gears		Pitch speed, ft. per min.	Equivalent breaking load at pitch line, pounds	Arc of action, inches[1]	Equivalent breaking load. Reduced to uniform modulus of rupture of 39,000 lb. per sq. in.
	Driver	Driven				
1	30	40	501	1743	0.628	1394
2	30	40	507	1622	0.628	1471
				Av. 1683		Av. 1433
13	30	60	501	2372	0.649	1329
14	30	60	504	2271	0.649	1489
				Av. 2322		Av. 1409
15	30	80	507	1930	0.662	1279
16	30	80	504	2044	0.662	1344
				Av. 1987		Av. 1312
17	30	100	510	1817	0.671	1269
18	30	100	507	2059	0.671	1342
				Av. 1938		Av. 1306
19	30	150	507	1998	0.685	1170
20	30	150	507	1917	0.685	1224
				Av. 1958		Av. 1197

[1]See Appendix No. 3, Paper 1382, for discussion of determination of arc of action.

was more than compensated for by the stronger form of the teeth of the larger gears. This possible explanation of the discrepant results also had to be abandoned.

22 Another explanation of the reduced results running in the wrong direction may lie in the possibility that the shafts were not exactly parallel. This would cause severer stress conditions the larger the radius of the gear.

23 Weighing all the evidence, we consider that the arc of action coefficients, a, derived in the earlier paper from experiments on more uniform and normal material, are essentially correct and they are

[1]Tables 17 and 18.

repeated here in Table 5, being extended to a ratio of

$$\frac{\text{arc of action}}{\text{pitch arc}} = 2.2.$$

From the way an additional tooth comes into action, as shown by the imperceptible increase of a for the range from a ratio of 1 to a ratio of 1.4, it is probable that the value of a remains at about 1.6 from a ratio of 2 to a ratio of 2.4.

24 The gears employed in Tests 13 to 20 inclusive were made

TABLE 5 COEFFICIENTS BASED ON ARC OF ACTION

BROWN & SHARPE, 14½-DEG. INVOLUTE GEARS

Ratio: $\frac{\text{Arc of action}}{\text{Pitch arc}}$	1	1.4	1.6	1.7	1.8	1.9	1.95	2.00	2.2
Corresponding a..............	1	1.05	1.1	1.15	1.24	1.38	1.47	1.60	1.60

TABLE 6 FELLOWS 20-DEG. INVOLUTE STUB-TOOTH GEARS, $^{10}/_{12}$ PITCH, 1 $\frac{1}{16}$-IN. FACE 30-TOOTH MESHING WITH 40-TOOTH

1st Series, 1914

Test number	Cast Iron Gears		8-Pitch, B. & S. Steel Change Gears				
	Pitch speed, ft. per min.	Equivalent load at teeth, lb.	No. of teeth		Pitch speed, ft. per min.	Equivalent maximum pitch load, lb.
			Driver	Driven		
13A	0	2953	....	....	....	....
14A	0	3256	....	....	....	....
1A	72	2575	20	100	301	618
2A	71	2711	20	100	297	651
3A	266	2273	30	40	444	1364
4A	266	2257	30	40	444	1354
5A	501	1955	100	70	1462	670
6A	501	1985	100	70	1462	681
7A	872	1804	100	40	1454	1082
8A	872	1773	100	40	1454	1064
9A	1149	1683	100	30	1437	1346
10A	1149	1592	100	30	1437	1274
11A	1654	1577	100	20	1378	1892[1]
12A	1654	1562	100	20	1378	1874
16A	1960	1403	120	20	1633	1683
17A	1984	1380	120	20	1654	1656

[1]Log of test says, "The steel 20T showed about the limit of its endurance without abrasion."

from special patterns sent to the manufacturers. All of them were solid or webbed and the hubs were lengthened to permit a 2½-in. key being used; this precaution being necessary because of the longer lever arm of the tooth load.

25 It was intended to carry out tests on the static strength of the individual teeth of these gears, as shown in position A, Table 8,

Paper 1382, but because of the exceptional strength of the material, which caused the shaft to spring, it was not certain that the teeth were being held in the proper relation at the instant of rupture and this purpose was abandoned.[1] The experimental value of the factor for form of tooth (Lewis's factor y), as determined in Par. 29 and Appendix 4, Paper 1382, is therefore retained, being—for the Brown & Sharpe 14½-deg. involute teeth—equal to

$$\left(0.154 - \frac{1.26}{n}\right)$$

where n is the number of teeth in the gear.

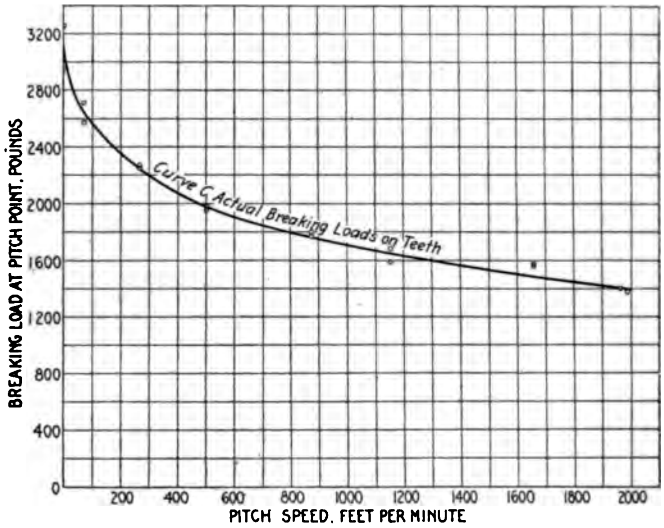


FIG. 3 RELATION BETWEEN PITCH SPEED AND BREAKING STRENGTH OF FELLOWS 30- AND 40-TOOTH 20-DEG. INVOLUTE STUB-TOOTH GEARS

SERIES C: TESTS TO DETERMINE VELOCITY COEFFICIENTS, FELLOWS GEARS

26 The tests for effect of pitch velocity on the breaking strength of Fellows 20-deg. involute, stub-tooth gears, 10/12-pitch, 1 1/16-in. face, 30-tooth meshing with 40-tooth, gave remarkably uniform and consistent results. These are tabulated in Table 6 and shown graphically in Fig. 3. This uniformity is in a large measure due to the uniformity of the material of this set of gears (see Par. 28 following) but may also have been influenced by the method of cutting. We had

[1]Tables 14, 15 and 16. Test No. 1C.

no apparatus to measure accuracy of tooth spacing and made no attempt to do so.

27 From the curve of Fig. 3, Table 7 of velocity coefficients, v, for Fellows 20-deg. involute gears has been computed. It will be noted that these values of v correspond quite closely to those obtained in the experiments with the Brown & Sharpe gears. (Compare Table 3.)

28 In the case of the Fellows tests the expedient of plotting the breaking loads reduced to a uniform modulus of rupture, and basing the velocity coefficients upon this curve, was not resorted to, as it was in the case of the Brown & Sharpe tests. The material of the gears used in this set of tests, 1A to 17A inclusive, being from a single melt, showed only moderate variation in the flexure tests and had an average

TABLE 7 VELOCITY COEFFICIENTS (v), BASED ON ACTUAL BREAKING LOAD ON TEETH

FELLOWS 20-DEG. INVOLUTE, STUB TOOTH GEARS

Pitch velocity, ft. per min	0000	100	200	300	400	500	600	700	800	900	1000
Velocity coefficient, v.	1.000	0.825	0.755	0.705	0.665	0.635	0.615	0.595	0.580	0.565	0.550
Pitch velocity, ft. per min	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	
Velocity coefficient, v.	0.540	0.525	0.515	0.505	0.495	0.485	0.475	0.470	0.460	0.450	

modulus of rupture of 39,600 lb. per sq. in. (See Tables 17 and 18.) For this reason, and for the further reason that the velocity coefficients derived from the reduced curve would be slightly higher than those derived from the actual test results and therefore tend away from safety rather than toward it, no curve is drawn in Fig. 3 reducing the actual test results to a uniform basis of a modulus of rupture of 39,000 lb. per sq. in. Table 8, however, gives the results of reduction computations in the same manner as shown in Table 2 for the Brown & Sharpe gears and is included here to complete the record. If the modified results be plotted it will be noted that they are not as regular and consistent as the actual test results.

SERIES D: TESTS TO DETERMINE ARC OF ACTION COEFFICIENTS, FELLOWS GEARS

29 The shortened addendum used in the Fellows system makes for shorter and less varied arcs of action. By the method described fully in Appendix 3 of Paper 1382, the values of the arcs of action for various combinations of Fellows gears were computed. The results are given in Table 9.

30 Tables 10 and 11 give the results of those experiments which were made to determine the effect of arc of action on breaking strength in the case of the Fellows gears. From Table 10 it is seen that in the static tests there is an increase in strength in the ratio of 3105/2506 = 1.24 for an increase in arcs of action in the ratio of

$$\frac{0.45008}{0.31416} = 1.4327.$$

31 Taking account of the single tooth, weakest position, static strength of the 30-tooth gear 2506 lb. as shown by the average of

TABLE 8 REDUCTION OF BREAKING LOAD TO BASIS OF UNIFORM MODULUS OF RUPTURE OF 39,000 LB. PER SQ. IN.

FELLOWS 20-DEG. INVOLUTE GEARS

Test number	Actual modulus of rupture from Table 18	Conditions of Running Test				Equivalent breaking load for modulus of rupture of 39,000 lb. per sq. in.
		Teeth in C. I. driver gear	Teeth in C. I. driven gear	Actual breaking load at pitch line by test	Velocity at pitch line, ft. per min.	
1A	40,420	30	40	2575	72	2485
2A	38,977	30	40	2711	71	2713
3A	42,415	30	40	2273	266	2089
4A	36,403	30	40	2257	266	2418
5A	35,920	30	40	1955	501	2123
6A	39,250	30	40	1985	501	1972
7A	39,450	30	40	1804	872	1783
8A	39,771	30	40	1773	872	1739
9A	36,890	30	40	1683	1149	1779
10A	Not tested	30	40	1592	1149	Blow hole
11A	39,262	30	40	1577	1654	1567
12A	38,240	30	40	1562	1654	1593
13A[1]	44,480	30	40	2953	0000	2589
14A	44,480	30	40	3256	0000	2855
15A	44,480	30	40	3861[2]	0000	Void
16A	43,897	30	40	1403	1960	1247
17A	38,850	30	40	1380	1984	1386
18A	46,030	30	80	1643	507	1392[3]
19A	46,900	30	80	2029	507	1688
20A	45,310	30	100	1907	507	1642
21A	46,460	30	100	2059	507	1729
22A	44,190	30	60	1969	507	1738
23A	44,860	30	60	2070	507	1800
24A	....	20	30	1477	476	....
25A	....	20	30	1437	476	...

[1]Same gears used in tests 13A, 14A and 15A.
[2]Void. Not weakest position.
[3]Void. See log of test.

Tests 1B and 2B, and multiplying by the velocity coefficient 0.635 for 500 ft. per min., we get 1591 lb. as the breaking strength at this speed for a 30-tooth Fellows gear with an arc of action of 0.31416.[1] This point is plotted in Fig. 4 with the averages of Table 11, which therefore shows the relations between equivalent breaking load and arc of action for the Fellows gears at this speed.

32 An interesting check is to compare with the results of Table 10, the increase for strength at this speed of 500 ft. per min., for an increase of arc of action from 0.31416 to 0.45008. The ratio of increase of strength is 1970/1591 = 1.24, as before.

33 Objection may be made to there being too few points to locate the curve of Fig. 4 with reasonable accuracy. As seen in the next paragraph the matter is not vital. The variation in value of the arc of action coefficient, a, in all cases except those involving very small pinions, is so slight that no appreciable error is introduced if it be taken as uniformly equal to 1.25 in the case of the Fellows gears.

34 Table 12 gives values for the arc of action coefficient,[2] a, as deduced from Fig. 4 for the gear combinations of Table 9 as covering the ordinary range. The values for others can be computed readily or interpolated with sufficient accuracy. It can be seen that $a = 1.33$ is about the maximum value in any practical case as compared with a corresponding maximum of about 1.60 for the Brown & Sharpe system.

SERIES E: DETERMINATION OF VALUE OF FACTOR Y, LEWIS FORMULA

35 The next step in the investigation of the Fellows gears was the determination of the expression for the factor depending upon the change of tooth-form as dictated by the number of teeth in the gear. This is Mr. Lewis's well-known factor y. For a 20-deg. involute tooth with addendum equal to 0.8 ÷ diametral pitch, Mr. Flanders[3] gives values from which we obtain

$$y = \left(0.173 - \frac{0.720}{n}\right)$$

By laying out the form of the 12, 30, 40 and rack teeth of the actual Fellows 10/12-pitch, ten times full size, and employing the method described in Appendix 2, Paper 1382, we obtained

[1]Par. 32, paper 1382.
[2]Par. 33, paper 1382.
[3]Trans. Am. Soc. M. E., vol. 30, p. 930.

TABLE 9 ARCS OF ACTION

COMBINATIONS OF $^{10}/_{12}$ PITCH GEARS FELLOWS 20-DEG. INVOLUTE

Gear Teeth: Driver	Gear Teeth: Driven	Arc of action, inches	Ratio of $\frac{\text{arc of action}}{\text{pitch arc}}$
Single tooth engagement	Weakest position[1]	0.31416	1.0000
12	12	0.38760	1.2334
20	30	0.43109	1.3722
30	40	0.45008	1.4327
30	60	0.45844	1.4591
30	80	0.46319	1.4744
30	100	0.46643	1.4852
30	Rack	0.48075	1.5271
100	100	0.48996	1.5596
100	Rack	0.50427	1.6051

[1]Position A, Table 8, Paper 1382.

TABLE 10 INFLUENCE OF ARC OF ACTION

FELLOWS 20-DEG. INVOLUTE GEARS

STATIC TESTS

Test number	Conditions of test	Breaking load at pitch line, pounds	Arc of action, inches	Ratio of $\frac{\text{arc of action}}{\text{pitch arc}}$
1B	30 T, position "A"........	2536	0.31416	1.0000
2B	30 T, position "A"........	2475	0.31416	1.0000
		Av. 2506	0.31416	1.0000
13A	30 T, driving 40 T.........	2953	0.45008	1.4327
14A	30 T, driving 40 T.........	3256	0.45008	1.4327
		Av. 3105	0.45008	1.4327

TABLE 11 INFLUENCE OF ARC OF ACTION AT APPROXIMATELY 500 FT. PER MIN., PITCH SPEED

FELLOWS 20-DEG INVOLUTE GEARS

Test number	Gear Teeth: Driver	Gear Teeth: Driven	Pitch speed ft. per min.	Breaking load at pitch line, pounds	Arc of action, inches	Ratio of $\frac{\text{arc of action}}{\text{pitch arc}}$
5A	30	40	501	1955	0.45008	1.4327
6A	30	40	501	1985	0.45008	1.4327
				Av. 1970	0.45008	1.4327
22A	30	60	507	1969	0.45844	1.4590
23A	30	60	507	2070	0.45844	1.4590
				Av. 2020	0.45844	1.4590
19A	30	80	507	2029	0.46319	1.4744
				Av. 2029	0.46319	1.4744
20A	30	100	507	1907	0.46643	1.4852
21A	30	100	507	2059	0.46643	1.4852
				Av 1983	0.46643	1.4852

$$y = \left(0.169 - \frac{0.972}{n}\right)$$

36 As was found in regard to the similar factor for the Brown & Sharpe system[1] this method, based upon an unmodified flexure theory, does not correspond to actual conditions. The teeth in every case show much greater breaking strength than either of these values of y would give when substituted in the single tooth static strength formula, $W = s\,p\,f\,y$.

37 A series of tests was made on the static strength of single

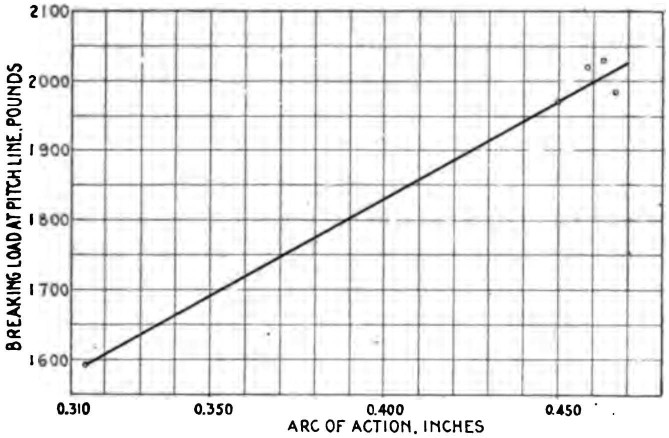


FIG. 4 RELATION OF BREAKING LOAD TO ARC OF ACTION: FELLOWS 20-DEG. INVOLUTE AT 500 FT. PER MIN. PITCH SPEED

teeth under the conditions of load application corresponding to engagement at their weakest position. This is Position A of Table 8, Paper 1382, and corresponds to the maximum strength of such gears when the arc of action is just equal to or is less than the pitch arc (i.e. circular pitch). In this case both velocity coefficient and arc of action coefficient become equal to unity and the value of y can be computed directly. The loads were applied by means of a steel pinion. Par. 27, Paper 1382. The results of these tests are given in Table 13. In computing the values of y, s is taken equal to the average shown by all the Fellows test specimens, 40,910 lb., (See Tables 17 and 18); $p = 0.31416$ in.; and $f = 1.0625$ in.

38 By the method of Appendix 4, Paper 1382, these results

[1]Par. 29, paper 1382.

TABLE 12 COEFFICIENTS (a), BASED ON ARC OF ACTION

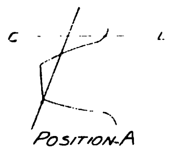
FELLOWS 20-DEG. INVOLUTE, STUB TOOTH GEARS

Teeth in Engaging Gears		Ratio of $\frac{\text{arc of action}}{\text{pitch arc}}$	Corresponding a
Single tooth engagement		1.0000	1.00
12	12	1.2334	1.13
20	30	1.3722	1.20
30	40	1.4327	1.24
30	60	1.4591	1.25
30	80	1.4744	1.26
30	100	1.4852	1.27
30	Rack	1.5271	1.29
100	100	1.5596	1.31
100	Rack	1.6051	1.33

TABLE 13 EQUIVALENT STATIC BREAKING LOADS (W), AT PITCH LINE

FELLOWS 20-DEG. INVOLUTE, STUB TOOTH, $^{10}/_{12}$ PITCH GEARS

POSITION OF STRESSED TOOTH



Teeth in gear	Test number	Equivalent breaking load, W, pounds	Average W, pounds	$y=\frac{W}{s p f}$ ($s p f = 13{,}656$)
20	3B	1702		
20	4B	1762	1732	0.127
30	1B	2526		
30	2B	2486	2506	0.184
40	5B	3029	3029	0.222
60	7B	3582		
60	8B	3481	3532	0.259
80	9B	3443		
80	10B	3858		
80	11B	3631	3643	0.267
100	12B	3803	3803	0.279

give approximately a value of

$$y = \left(0.317 - \frac{3.81}{n}\right)$$

But an examination of this equation shows that it would lead to a zero value of y, and hence of W, for $n = 12$, which is obviously incorrect. It would also lead to a value of $W = 4329$ for a rack tooth ($n = \infty$); while the experiments showed that the teeth would fail by shear, as indicated in Fig. 5, rather than by flexure, long before such a load could be reached. A note made in the log at the time of these tests says: "It seems a fair conclusion that these stub-teeth when loaded at the end are apt to fail by shear before reaching a load that would break them out, in the case of 60-tooth gears and larger." Our judgment based upon observed experiments was that this limiting

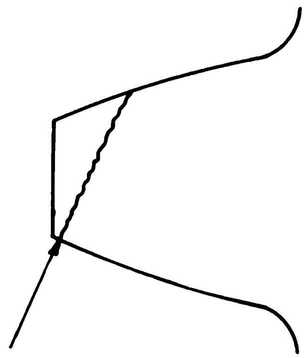


FIG. 5 MANNER OF TOOTH FAILURE

value for a rack tooth end load would be 3800 lb. Upon this basis and by combination with W for $n = 20$, 30 and 40,

$$y = \left(0.278 - \frac{2.69}{n}\right)$$

was deduced. It will be seen in Tables 19 and 20 that the results derived from its use check very closely with the actual running test results. Where they depart from the test results in the static cases they do so on the safe side.

39 It is to be borne in mind that we questioned the test results in these static tests above 40-tooth gears, as we noted in the log that because of the torsional deflection of the shafts we could not be certain that the teeth were in position A at the instant of rupture. Any deflection would put them in a position to carry a heavier load.

SUMMARY OF INVESTIGATION OF BREAKING STRENGTH OF BROWN & SHARPE 14½-DEG. INVOLUTE, AND FELLOWS 20-DEG. INVOLUTE, STUB-TOOTH, CAST-IRON, CUT GEARS

SYMBOLS, BOTH SYSTEMS

W = safe equivalent load at pitch line, pounds
s = modulus of rupture = 36,000 lb. per sq. in. for cast iron
p = circular pitch, inches = pitch arc
f = width of face of gear, inches
n = number of teeth in gear
k = factor of safety
Suggested values: $k = 4$, for steady load, no reversal of stress
$k = 6$, suddenly applied load, no reversal of stress
$k = 8$, suddenly applied load, with reversal of stress
v = velocity coefficient. See tables
a = arc of action coefficient. See tables

FORMULÆ

Brown & Sharpe 14½-deg. involute:

$$W = \frac{s\,p\,f}{k}\left(0.154 - \frac{1.26}{n}\right) v\,a$$

Fellows 20-deg. involute, stub tooth:

$$W = \frac{s\,p\,f}{k}\left(0.278 - \frac{2.69}{n}\right) v\,a$$

Neither formula holds for values of n less than 12.

VALUES OF (v)

Pitch velocity, ft./min.	v: Brown & Sharpe 14½-deg. involute	v: Fellows 20-deg. involute stub tooth	Pitch velocity, ft./min.	v: Brown & Sharpe 14½-deg. involute	v: Fellows 20-deg. involute stub tooth
0000	1.000	1.000	1100	0.470	0.540
100	0.795	0.825	1200	0.455	0.525
200	0.730	0.755	1300	0.445	0.515
300	0.675	0.705	1400	0.435	0.505
400	0.635	0.665	1500	0.430	0.495
500	0.595	0.635	1600	0.420	0.485
600	0.565	0.615	1700	0.415	0.475
700	0.540	0.595	1800	0.410	0.470
800	0.520	0.580	1900	0.405	0.460
900	0.500	0.565	2000	0.400	0.450
1000	0.485	0.550	....	....	....

VALUES OF (a)

Teeth in engaging gears		Corresponding a: Brown & Sharpe 14½-deg. involute	Corresponding a: Fellows 20 deg involute stub tooth
Single tooth engages		1.00	1.00
12	12	1.10	1.13
20	30	1.15	1.20
30	30	1.47	1.22
30	40	1.60	1.24
30	60	1.60	1.25
30	80	1.60	1.26
30	100	1.60	1.27
30	Rack	1.60	1.29
100	100	1.60	1.31
100	Rack	1.60	1.33

40 In the Fellows system of tooth forms the ratio of addendum to pitch is not a constant one for different pitches, hence the results obtained on 10/12-pitch gears do not hold with exactness for other pitches. However, the differences in tooth proportions are not sufficiently great to forbid the use of these results with a reasonably close degree of accuracy for other pitches.

CONCLUSIONS FROM ENTIRE SERIES OF TESTS

41 For convenient reference the conclusions arrived at from the entire experimentation, reported upon in both this paper and No.

CHECK OF FELLOWS FORMULA

Test number	W, by test	W, by formula $s = 39{,}588$
1A	2575	2622
2A	2711	2629
3A	2273	2225
4A	2257	2225
5A	1955	1957
6A	1985	1957
7A	1804	1753
8A	1773	1753
9A	1683	1642
10A	1592	1642
11A	1577	1480
12A	1562	1480
13A	2953	3081
14A	3256	3081
15A	Void	....
16A	1403	1399
17A	1380	1393

1382, are here summarized on page 520. The formulae and factors represent the best judgment of the present writers, based upon painstaking and unprejudiced study of the complete data. To enable others to reach their own conclusions, should they question those here presented, the logs of all the earlier tests will be found in Paper 1382 and those of these tests in Tables 14, 15 and 16 of this paper. The original goal of the investigation, namely the definite determination of the effect of pitch-speed upon breaking strength, has been attained, we feel, for the ordinary working range of velocities.

42 Too much stress must not be laid upon the comparison, in individual cases, of the formula results and the test results reduced to the modulus of 36,000 lb. per sq. in. as shown in Tables 19 and 20. In a non-homogeneous material, like cast iron, the chances are altogether against the strength of the single test-specimen being

TABLE 14 LOGS OF GEAR BREAKING TESTS OF 10 DIAMETRAL PITCH, BROWN & SHARPE, 14½-DEG. INVOLUTE CAST IRON GEARS

MOTOR SPROCKET, 19-TEETH; CHANGE GEAR SPROCKET, 37 TEETH; BRAKE ARM = 30.25 IN.; ZERO BRAKE LOAD = 4.75 LB.

Series	Test number	Date, 1914	Teeth in Gears: Steel change gears, S P., B. & S. Driver	Steel change gears Driven	C. I. tested gears Driver	C. I. tested gears Driven	Net brake load, pounds	Motor r.p.m. observed by tachometer	Notes made at conclusion of each test
1	1	Aug. 18	100	70	30	40	115.25	870	11 teeth out of 30; 10 teeth out of 40
1	2	Aug. 19	100	70	30	40	107.25	880	5 teeth out of 30; 7 teeth out of 40
1	3	Aug. 19	100	40	30	40	95.25	865	25 teeth out of 30; 13 teeth out of 40
1	4	Aug. 19	100	40	30	40	105.25	865	8 teeth out of 30; 9 teeth out of 40
1	5	Aug. 19	70	20	30	40	85.25	865	30 teeth out of 30; 8 teeth out of 40; both steel gears abraded
1	6	Aug. 20	100	30	30	40	91.25	865	29 teeth out of 30; 19 teeth out of 40
1	7	Aug. 20	100	30	30	40	83.25	875	30 teeth out of 30; 7 teeth out of 40; 2 of which were not broken completely across the face
1	8	Aug. 20	100	30	30	40	89.25	865	30 teeth out of 30; 24 teeth out of 40
1	9	Aug. 20	100	20	30	40	71.25	855	30 teeth out of 30; 16 teeth out of 40
1	10	Aug. 20	100	20	30	40	69.25	860	29½ teeth out of 30; 13 teeth out of 40
1	11	Aug. 24	120	20	30	40	79.25	[illegible]	30 teeth out of 30; 21 teeth out of 40
1	12	Aug. 24	120	20	30	40	67.25	850	30 teeth out of 30; 18 teeth out of 40
2	13	Dec. 18	100	70	30	60	235.25	870	8 teeth out of 30; 4 teeth out of 60
2	14	Dec. 18	100	70	30	60	225.25	875	30 teeth out of 30; 5 teeth out of 60
2	15	Dec. 18	100	70	30	80	255.25	880	9½ teeth out of 30; 4 teeth out of 80
2	16	Dec. 18	100	70	30	80	270.25	875	30 teeth out of 30; 3 teeth out of 80
2	17	Dec. 18	100	70	30	100	300.25	885	10 teeth out of 30; 6 teeth out of 100
2	18	Dec. 18	100	70	30	100	340.25	880	2 teeth out of 30; 5 teeth out of 100
2	19	Dec. 19	100	70	30	150	495.25	880	6 teeth out of 30; 5 teeth out of 150; all clean breaks
2	20	Dec. 19	100	70	30	150	475.25	880	17 teeth out of 30; 5 teeth out of 150

CAST IRON GEARS

BRAKE ARM = 30.25 IN.; ZERO BRAKE LOAD = 4.75 LB. MOTOR SPROCKET, 19 TEETH; CHANGE GEAR SPROCKET, 37 TEETH

Series	Test number	Date, 1914	Teeth in Gears: Steel change gears, 8 P., B. & S.: Driver	Teeth in Gears: Steel change gears, 8 P., B. & S.: Driven	Teeth in Gears: C. I. tested gears: Driver	Teeth in Gears: C. I. tested gears: Driven	Net brake load, pounds	Motor r.p.m. observed by tachometer	Notes made at conclusion of each test
1	1A	Aug. 21	20	100	30	40	170.25	895	7 teeth out of 30; 4 tooth out of 40; break very deep arc
1	2A	Aug. 21	20	100	30	40	179.25	885	1 tooth out of 30; 3 teeth out of 40
1	3A	Aug. 21	30	40	30	40	150.25	880	4 teeth out of 30; 4 teeth out of 40; gears seem noisier than the B. & S.
1	4A	Aug. 21	30	40	30	40	149.25	880	16 teeth out of 30; 7 teeth out of 40; including 3 part across and 1 battered in 30; 1 part across and 1 battered in 40
1	5A	Aug. 21	100	70	30	40	129.25	870	30 teeth out of 30; 11 teeth out of 40
1	6A	Aug. 22	100	70	30	40	131.25	870	15 teeth out of 30; 16 teeth out of 40; some teeth partially broken out of 30
1	7A	Aug. 22	100	40	30	40	119.25	865	30 teeth out of 30; 19 teeth out of 40; these gears show more partial teeth broken off
1	8A	Aug. 22	100	40	30	40	117.25	865	29½ teeth out of 30; 11½ teeth out of 40
1	9A	Aug. 22	100	30	30	40	111.25	855	30 teeth out of 30; 11 teeth out of 40
1	10A	Aug. 22	100	30	30	40	105.25	855	30 teeth out of 30; 5 teeth out of 40
1	11A	Aug. 22	100	20	30	40	104.25	820	29 teeth out of 30; 19 teeth out of 40; steel 20 showed about limit of its endurance without abrasion
1	12A	Aug. 22	100	20	30	40	103.25	820	30 teeth out of 30; 18 teeth out of 40 including 4 or 5 partial
1	13A	Aug. 22	Static	..	30	40	195.25	Static	1 tooth out of 30; used hand lever brake set tight
1	14A	Aug. 22	Static	..	30	40	215.25	Static	1 tooth out of 30; 2 teeth out of 40; same gear ¼ way round
1	15A	Aug. 22	Static	..	30	40	255.25	Static	Test 13A to 15A. Struck weakest place first time
1	16A	Aug. 24	120	20	30	40	92.75	810	30 teeth out of 30; 40 teeth out of 40
1	17A	Aug. 24	120	20	30	40	91.25	820	30 teeth out of 30; 39 teeth out of 40
2	18A	Nov. 28	100	70	30	80	217.25	880	30 teeth out of 30; 7 teeth out of 40; broke platform board of scales. Brake wheel web and hub cracked
2	19A	Dec. 12	100	70	30	80	268.25	880	New brake wheel and scales. This run shows previous run was unreliable and should be cast out. Key way had expanded in sprocket wheel shaft. New shaft made for next run
2	20A	Dec. 16	100	70	30	100	315.25	880	All but 7 teeth (which were badly broken) stripped from 30T. 7 teeth stripped or cracked on 100T
2	21A	Dec. 16	100	70	30	100	340.25	880	Heard crack and stopped run. 1 tooth cracked on 30T gear. 100 tooth gear not broken
2	22A	Dec. 16	100	70	30	60	195.25	880	All of 30 and 5 teeth on 60 broken
2	23A	Dec. 16	100	70	30	60	205.25	880	13 teeth from 30
2	24A	Dec. 21	40	20	20	30	73.25	885	Broke 20T into 4 pieces, no teeth out. No teeth out of 30. Teeth stronger than 20T gear
2	25A	Dec. 21	40	20	20	30	71.25	885	20T gear in 4 pieces. 30 not broken. No teeth out of either

exactly that of the tooth which first broke in the gear test. This is borne out by the fact that the actual test value of W sometimes comes out larger in the case of duplicate experiments (all conditions the same) for the gear whose material subsequently showed the lower modulus of rupture in the flexure tests. (Examples: Tests 1A and 2A, 3A and 4A.) Again, in many duplicate tests where the differences in breaking strength lay in the same direction as the differences in test-specimen moduli, the tooth strength varia-

TABLE 16 LOGS OF TESTS OF FELLOWS GEARS, STATIC LOADS. POSITION *A*

Test number	Date, 1914	C. I. test gear	Net brake load pounds	Equivalent load at pitch line, C. I. gears brake lever arm = 30.25	Average	Remarks
1B	Dec. 21	30	125.25	2526	....	
2B	Dec. 21	30	123.25	2486	2506	
3B	Dec. 21	20	56.25	1702	....	
4B	Dec. 21	20	58.25	1762	1732	
5B	Dec. 21	40	200.25	3029	3029	
6B	Dec. 21	40	238.25	3604	Void	Load not at end. Steel gear gouged out; case-hardened.
7B	Dec. 22	60	355.25	3582	....	Good break.
8B	Dec. 22	60	345.25	3481	3532	Tooth sheared; did not break out at bottom.
9B	Dec. 22	80	455.25	3443	....	Good break.
10B	Dec. 22	80	510.25	3858	....	Sheared and also broke at bottom
11B	Dec. 22	80	480.25	3631	3643	
12B	Dec. 22	100	630.25	3803	3803	
13B	Dec. 22	100	395.25	2391	Void	Failed by shear.

NOTE TO TEST 11. Very difficult to get load on tooth in position *A*. Probably it was applied near the pitch line rather than at end. This was due to spring of shafts, etc.

NOTE TO TEST 13. It seems a fair conclusion that these stub teeth when loaded at the end are apt to fail by shear before reaching a load that would break them out, in the case at 60T gears and larger. No use to go on with Fellows, pos. *A*.

On Dec. 22 also tested (No. 1 C.) a 60T, Brown & Sharpe, static load, position *A*. The net brake load was 285.25 lb. The equivalent load at pitch line, 2876 lb. NOTE: On account of spring and play in apparatus it does not seem as if these results in static position *A* for gears above 40T are reliable. We therefore conclude to stop the tests at this point.

tion was not as great as the variation in modulus. Examples:

Tests 1 and 2; Ratio of Moduli, 1.135; Ratio of W, 1.075.
Tests 4 and 3; Ratio of Moduli, 1.179; Ratio of W, 1.105.
Tests 6 and 8; Ratio of Moduli, 1.157; Ratio of W, 1.022.
Tests 9 and 10; Ratio of Moduli, 1.131; Ratio of W, 1.029.

43 To get a better check of the formula, therefore, we may select Tests 1A–17A, inclusive, of the Fellows gears which were made from a single melt whose average modulus of rupture was 39,588 lb. (Tables 17 and 18), substitute this value of s in the formula, and

compare with the actual breaking test results. This is done in the table given on page 521.

COMMENTS ON TESTS

44 The method of testing these gears was employed after due consideration, despite the criticisms made upon it in the discussion of the previous paper. It has the great merit of being both simple and

TABLE 17 LOG OF TESTS OF MATERIALS, SPECIMENS CUT FROM GEARS

DATA AND RESULTS; BROWN & SHARPE GEARS

Mark—test number	Date of flexure test	Thickness of test piece, inches	Width of test piece, inches	Breaking load at middle[1]	Corresponding modulus of rupture lb. per sq. in.
30T, 1	Aug. 28, '14	0.254	1.068	1282	48,770
30T, 2	Sep. 16, '14	0.246	1.065	1055	42,976
30T, 3	Sep. 16, '14	0.242	1.062	1098	46,323
30T, 4	Sep. 16, '14	0.235	1.066	1225	54,632
30T, 5	Sep. 16, '14	0.255	1.064	1137	43,143
30T, 6	Aug. 28, '14	0.256	1.065	1300	49,040
30T, 7	Sep. 16, '14	0.233	1.062	863	39,288
30T, 8	Sep. 16, '14	0.262	1.065	1180	42,382
30T, 9	Sep. 16, '14	0.250	1.065	1112	43,868
30T, 10	Sep. 16, '14	0.250	1.065	983	38,779
30T, 11	Aug. 28, '14	0.260	1.065	1300	47,400
30T, 12	Sep. 16, '14	0.244	1.063	995	41,300
30T, 13	June 2, '15	0.251	1.058	1767	69,600
30T, 14	June 2, '15	0.249	1.058	1485	59,460
30T, 15	June 2, '15	0.250	1.057	1475	58,840
30T, 16	June 2, '15	0.249	1.057	1480	59,300
30T, 17	June 2, '15	0.252	1.058	1430	55,850
30T, 18	June 2, '15	0.250	1.057	1500	59,840
30T, 19	June 2, '15	0.249	1.058	1665	66,620
30T, 20	June 2, '15	0.263	1.057	1695	61,100
60T, 13	June 4, '15	0.247	1.065	1475	59,610
80T, 15	June 4, '15	0.250	1.063	1400	55,350

[1]Distance between supports, 1.75 inches.

positive. The apparatus is relatively inexpensive and requires no calibration. Since the chief criticisms were directed against the power consumption, meter readings were kept for the entire series of runs. The total power consumption as shown by the recording meter was only 150 kw-hr. The power cost is therefore inappreciable. Were these experiments on wear or endurance, rather than breaking strength, the cost of the power might enter into the problem as a determining factor and indicate the necessity for some such apparatus as that described by Wilfred Lewis at the June, 1914, meeting of the Society.[1] However, with rupture tests, judging from our experience with the way the teeth are thrown at high speeds, it would seem inevitable that fractured cast-iron teeth would fall between the teeth of the steel gear and pinion and wreck the Lewis machine. It is also a question whether the steel gears (unless made of special material and heat-treated), necessarily having the same pitch and pitch-

TABLE 18 LOG OF TESTS OF MATERIALS, SPECIMENS CUT FROM 30T GEARS

DATA AND RESULTS; FELLOWS 20-DEG. INVOLUTE GEARS

Mark=test number	Date of flexure test	Thickness of test piece, inches	Width of test piece, inches	Breaking load at middle[1]	Corresponding modulus of rupture lb. per sq. in.
1A	Aug. 28, '14	0.241	1.068	955	40,420
2A	Sep. 16, '14	0.279	1.067	1233	38,977
3A	Sep. 16, '14	0.254	1.067	1112	42,415
4A	Sep. 16, '14	0.271	1.065	1085	36,403
5A	Sep. 16, '14	0.262	1.065	1000	35,920
6A	Aug. 28, '14	0.266	1.066	1128	39,250
7A	Sep. 16, '14	0.250	1.065	1000	39,450
8A	Sep. 16, '14	0.267	1.065	1150	39,771
9A	Sep. 16, '14	0.258	1.064	995	36,890
10A	Aug. 28, '14	0.260	1.065	Blow hole	Not tested
11A	Sep. 16, '14	0.262	1.066	1095	39,262
12A	Aug. 28, '14	0.254	1.064	1000	38,240
13A, 14A, 15A	June 4, '15	0.248	1.063	1108	44,480[2]
16A	Sep. 16, '14	0.259	1.065	1195	43,897
17A	Aug. 28, '14	0.241	1.067	917	38,850
18A	June 4, '15	0.249	1.058	1150	46,030
19A	June 4, '15	0.250	1.057	1180	46,900
20A	June 4, '15	0.250	1.057	1140	45,310
21A	June 4, '15	0.250	1.058	1170	46,460
22A	June 4, '15	0.250	1.058	1130	44,190
23A	June 4, '15	0.249	1.057	1120	44,860

[1]Distance between supports, 1.75 in.
[2]Same gears used in tests 13A, 14A and 15A.
Average first lot of material (tests 1A-17A, incl.), $s = 39{,}588$.
Average second lot of material (balance of tests), $s = 45{,}625$.
Average of all material, $s = 40{,}909$.

speeds as the cast-iron test gear, would be able to stand up without destructive abrasion under the load which would be required to break the cast-iron teeth. Our own experience (Tests 5 and 11A) leads us to doubt that ordinary, unhardened, steel gears would stand up under these conditions. To carry out our static tooth strength tests, where

[1]Trans. Am. Soc. M. E., vol. 36, p. 231.

the load was directly applied by 10-pitch steel pinions, we found it necessary to case-harden these steel pinions. (See Table 16, 6B.)

45 These experiments, incidentally, give data on the carrying power shown by soft steel gears of the 14½-deg. involute form, 8-pitch, 1 1/16-in. face, at pitch speeds ranging up to 1700 ft. per min. (See Tables 1 and 6.) Under the conditions of lubrication here employed the limiting load seems, roughly, to be about 1500 lb. per in. width of face for these gears.

46 In respect to the ball bearings in the improved testing apparatus, some light is thrown indirectly upon the carrying power of such bearings under rather severe conditions; for it must be borne in

TABLE 19 COMPARISON OF FORMULAE WITH BREAKING LOAD ON GEARS BY TEST

BROWN & SHARPE 14½-DEG. INVOLUTE GEARS

Test number	Test Gears: Driver	Test Gears: Driven	Pitch line velocity, ft./min.	v	a	s by test	W, actual test value	W, test value reduced to modulus of rupture = 36,000	W, computed by formula, with s = 36,000
1	30	40	501	0.595	1.60	48,770	1743	1287	1281
2	30	40	507	0.594	1.60	42,980	1622	1359	1279
3	30	40	872	0.505	1.60	46,320	1441	1120	1087
4	30	40	872	0.505	1.60	54,680	1592	1049	1087
5	30	40	1221	0.453	1.60	43,140	1289	1076	975
6	30	40	1163	0.461	1.60	49,040	1380	1013	993
7	30	40	1176	0.459	1.60	39,290	1259	1151	988
8	30	40	1163	0.461	1.60	42,380	1350	1147	993
9	30	40	1724	0.414	1.60	43,870	1078	885	891
10	30	40	1734	0.413	1.60	38,780	1047	972	889
11	30	40	2021	0.399	1.60	47,400	1199	911	859
12	30	40	2057	0.397	1.60	41,300	1017	887	855
13	30	60	501	0.595	1.60	69,600	2372	1227	1281
14	30	60	504	0.594	1.60	59,460	2271	1375	1279
15	30	80	507	0.594	1.60	58,840	1930	1181	1279
16	30	80	504	0.594	1.60	59,300	2044	1241	1279
17	30	100	510	0.592	1.60	55,850	1817	1171	1275
18	30	100	507	0.594	1.60	59,840	2059	1239	1279
19	30	150	507	0.594	1.60	66,620	1998	1080	1279
20	30	150	507	0.594	1.60	61,100	1917	1130	1279

Formula: $W = \frac{spf}{k}\left(0.154 - \frac{1.26}{n}\right) v a$

s = 36,000 for cast iron, recommended; p = 0.31416; f = 1.0625; k = 1

mind that the bearings were running, at the instant of rupture, in some cases at as high a rotative speed as 2500 r.p.m. and that the actual suddenly applied stress upon them, after the teeth began to break and wedge, must have been much greater even than the recorded breaking strength of the teeth, high as this was.

47 The tests were made in the laboratories of the Leland Stanford Junior University and the writers wish to express their appreciation of the cordial coöperation of the university authorities. Particular thanks are due Prof. W. F. Durand, executive head of the department of mechanical engineering, and to Prof. W. R. Eckart, in charge of the experimental engineering laboratories. Both the Brown & Sharpe Manufacturing Company and the Fellows Gear Shaper Company generously donated the gears necessary for the experiments.

TABLE 20 COMPARISON OF FORMULAE WITH BREAKING LOAD ON GEARS BY TEST

FELLOWS 20-DEG. INVOLUTE STUB TOOTH GEARS

Test number	Test Gears Driver	Test Gears Driven	Pitch line velocity, ft./min.	v	a	s by test	W, actual test value	W, test value reduced to modulus of rupture = 36,000	W, computed by formula, with s = 36,000
1A	30	40	72	0.851	1.24	40,420	2575	2293	2384
2A	30	40	71	0.853	1.24	38,980	2711	2504	2390
3A	30	40	266	0.722	1.24	42,420	2273	1929	2023
4A	30	40	266	0.722	1.24	36,400	2257	2232	2023
5A	30	40	501	0.635	1.24	35,920	1955	1959	1779
6A	30	40	501	0.635	1.24	39,250	1985	1821	1779
7A	30	40	872	0.569	1.24	39,450	1804	1646	1594
8A	30	40	872	0.569	1.24	39,770	1773	1605	1594
9A	30	40	1149	0.533	1.24	36,890	1683	1642	1493
10A	30	40	1149	0.533	1.24	Blowhole	1592	Not tested	1493
11A	30	40	1654	0.480	1.24	39,260	1577	1446	1345
12A	30	40	1654	0.480	1.24	38,240	1562	1470	1345
13A	30	40	000	1.000	1.24	44,480	2953	2390	2801
14A	30	40	000	1.000	1.24	44,480	3256	2635	2801
15A	30	40	000	1.000	1.24	44,480	Void	....	....
16A	30	40	1960	0.454	1.24	43,900	1403	1151	1272
17A	30	40	1984	0.452	1.24	38,850	1380	1279	1266
18A	30	80	507	0.634	1.26	46,030	Void	....	....
19A	30	80	507	0.634	1.26	46,900	2029	1557	1803
20A	30	100	507	0.634	1.27	45,310	1907	1515	1819
21A	30	100	507	0.634	1.27	46,460	2059	1595	1819
22A	30	60	507	0.634	1.25	44,190	1969	1604	1790
23A	30	60	507	0.634	1.25	44,860	2070	1586	1790
24A	20	30	476	0.643	1.20	No test	1477	....	1296
25A	20	30	476	0.643	1.20	No test	1437	....	1296
1B	..	30	000	1.000	1.00	No test	2526	....	2259
2B	..	30	000	1.000	1.00	No test	2486	....	2259
3B	..	20	000	1.000	1.00	No test	1702	....	1718
4B	..	20	000	1.000	1.00	No test	1762	....	1718
5B	..	40	000	1.000	1.00	No test	3029	....	2536

Formula: $W = \frac{s\,p\,f}{k}\left(0.278 - \frac{2.69}{n}\right) v\,a$

s = 36,000 for cast iron, recommended; p = 0.31416; f = 1.0625; k = 1.

DISCUSSION

L. D. Burlingame remarked that this series of investigations to cover one of the features which go to make up a successful gear whets our appetites for further investigation to cover other important features, such as the matter of wear and of quiet running, since we can only get at the most satisfactory or successful system of gearing by taking all these into consideration. Nevertheless, the investigation described is of great value as giving data on which we can base further investigations.

The Committee on Standard System of Involute Gearing, which investigated various systems, found from their tests that the 14½ deg. involute gear gave a decidedly greater percentage of efficiency, and a quieter running gear. It would be a very desirable use of funds available for research work to carry out further experiments along the lines of the paper and thus establish the basis of a system of gearing for universal application, taking all the above factors into consideration.

The question of strength of gearing, Mr. Burlingame stated, is affected very possibly by the consideration that with the small pressure angle (14½ deg. for the Brown & Sharpe system), the pinion is the weaker element (that is, where a small pinion runs into a large gear). The engineering solution is to make the pinion of steel, or if further strength is required, to temper it and thus get the benefit of strength without sacrificing wear or quiet running.

It is interesting to note in Fig. 2 that the curve of actual break of the Brown & Sharpe gear is materially higher than the theoretical curve on which the figures are based. This shows that the figures given by the authors are on the conservative side and are such as can be used with safety, that is, they give a larger margin of safety than perhaps the actual facts would warrant.

It is also interesting to note the suggestion as to contact with more than one tooth, with a question mark against three teeth being in contact at certain times. This is largely a matter of accuracy of cutting and of the system on which the teeth are based. He doubted if in many cases three teeth are actually in contact, from the fact that the point of the tooth of the ordinary Brown & Sharpe gear is eased off for quiet running, which would probably prevent actual contact except under extremely heavy pressure.

Kate Gleason said that one of the very interesting things to her that Professor Marx had found in comparing the Lewis formula with his own work, is that the strength did not fall off at high speeds as rapidly as Mr. Lewis found it did. The Gleason Works had found, just as Professor Marx did, that the strength does not fall off at high speeds, but the load is apt to be so much harder—due to the starting and stopping at the high speed—that much stronger gears have to be allowed in practice than tests indicate.

For instance, in rolling mill machinery, running at a high speed as it often does, if mills are being driven by motors, almost double the strength has to be given to the gears than would in even the Lewis formula, whereas, the formula when applied to machine tool gears gives strength more than ample. It seemed to her that when the loads come on the gears so quickly, the action is almost like a hammer and crystallizes the material, and though the latter may be amply strong for the first month or two, it will disintegrate if not provided for, perhaps by putting in nickel steel, or something of that kind.

Miss Gleason stated that they found in the same way in testing worm gears, that they will not show up on a test at all as they do in practice. The Society has had papers showing efficiencies up to 90 per cent on worm gear drives, but in practice we cannot get 20 per cent out of a worm gear sometimes.

G. H. Marx replied that he was particularly interested in Miss Gleason's discussion of the paper, because it was about twenty-five years ago, when he was working for the Gleason Works in Rochester, that his interest in gearing began.

He agreed that the experiments are limited merely to the question of strength, which is a very small part of the whole problem, and they are limited to cast iron gears. But the only way to get an investigation done is to take a small portion of the field and cover it as thoroughly as can be, and then take the neighboring area and perhaps go into that.

As to the question of the falling off of strength to speed, of course, it was as Miss Gleason said, and she must remember that these tests were made under running conditions. He said their plea is that where there is shock, where there are reversals of stress, or where the stress is suddenly applied, that factor should be taken care of by a proper factor of safety and not in a velocity coefficient. So they have separated those and have a factor of safety in their formula, in which the question of the matter of application of load, or the question of reversal of stresses is provided for.

No. 1495

SCIENCE IN ITS RELATION TO ENGINEERING

PRESIDENTIAL ADDRESS 1915

By Dr. John A. Brashear, Pittsburgh, Pa.
President of the Society

The early history of engineering in at least three of its phases, —namely, civil, mining and mechanical,—has much of interest to the technical student of today, and we of the twentieth century would be false to our convictions did we not give our mead of praise to the pioneers who have left in evidence some of the monuments of their labors which stand almost unimpaired after the ravages of many centuries.

Some of the members of our Society had the pleasure of visiting the Panama-Pacific International Exposition and participating in the celebration of the completion of the Panama Canal. It was a great event, but we should not forget that many centuries ago the possibility of building a canal to connect the Mediterranean and the Red Sea was seriously discussed by the engineers of that day. The project was considered impractical because it was suggested that there was a difference of level of over thirty-two feet between the two bodies of water, although the great mathematician La Place insisted that this could not be possible, since by the law of gravity there could be no difference in the mean level of large bodies of water on the earth.

Greek historians tell us that it was the fear of flooding Egypt with the waters of the Red Sea which prevented Darius from undertaking the canal project, yet this canal was successfully constructed centuries after the time of Darius.

The pyramid builders were certainly engineers of no mean type. The handling of those massive stones, many of them weighing as much as thirty tons, their transportation and the placing of them in position, were problems that would concern the engineers of today.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

As a matter of minor interest, it seems almost certain that hollow diamond drills were used at this early period.

I had the pleasure of a personal acquaintance with the astronomer Piazzi Smyth, who made a critical study of one of the great pyramids of Egypt; and while I cannot agree with his dictum that the base of the pyramid is the divine standard of measurement, I could not help being deeply impressed with his views of the engineering problems that were mastered in the erection of these wonderful structures.

I cannot refrain from mentioning a piece of engineering work of later date, namely the building of that splendid highway, the Via Appia. While the time element in its construction would be inexcusable in modern road building, and although it was probably never subjected to a tithe of the use and abuse of modern highways, the fact that it was in perfect repair nearly six hundred years after it was finished speaks volumes for the character of the engineering work done upon it. Three bridges and a small portion of the roadway still remain after the lapse of twenty-two centuries.

Roman engineers built more than forty thousand miles of such roadways through and out from the Roman provinces, and it is recorded that the water supply of the empire at the beginning of the Christian era would suffice for a population of seven million people. The aqueducts of Rome are considered among her grandest engineering works.

It would be impossible and, indeed, out of place to enumerate the achievements of the past in the many lines of civil, mining and mechanical engineering. My purpose in mentioning some of the work of our ancient brethren is to note the facts that what they constructed was largely upon an empirical basis; and that, nevertheless, their factors of safety were amply large. In many of the blunders of later date which, for want of a better name, may be styled scientific empiricism, "safety first" has unfortunately not always been the slogan.

In an admirable address given before the British Association of Science some forty years ago, Sir John Hawkshaw tells us that when knowledge in its higher branches was confined to the few, those who possessed it were called upon to perform various services for the community to which they belonged, and that mathematicians, astronomers, painters, sculptors and priests performed duties which now pertain to the professions of the architect and engineer.

Thus while it is true that the methods which were followed up to, let us say for safety three centuries ago, were largely empirical, perhaps we can see in the work of the painters and sculptors the why of the beautiful types of architecture from which the architects of today are unwilling to depart.

I dare not undertake to mention in extenso the various steps in the evolution of scientific research and its correlation with engineering problems. I bow my head in reverence to the great geometers and mathematicians of the past, men whose names are household words to the engineer of today, Pythagoras, Euclid, Pascal, Gregory, Ptolemy, Huygens, Descartes, Newton, Hamilton, La Place, Napier. Great have been their contributions to human knowledge, giving us the key that has enabled other master minds to solve some of the mysteries of God's illimitable universe.

But the science of mathematics has many handmaidens for probing into the hitherto unsolved problems of this old round world. Astronomy, physics, chemistry, and kindred sciences, have and ever will have their place in the world's great workshop, and ever will hold a place of honor in the great field of engineering.

There is not a single problem in modern engineering that can not be more readily solved by a knowledge of facts developed by scientific study and research, and no man knows this better than the successful engineer of today.

Dr. Maclaurin tells us a story of Matthew Arnold, the apostle of Sweetness and Light, in which he is made to say, that when a candle burns the oxygen and nitrogen of the air combine with the carbon in the candle to form carbonic acid gas, *but who cares?* The story is told because it suggests an attitude to science that is far from rare, even amongst people of intelligence today. He recalls the poetic query of Keats when he writes of the rainbow: "Do not all charms fly at the mere touch of philosophy? There was an awful rainbow once in heaven, we know her woof and texture. She is given in the dull catalogue of common things."

"The complaint seems to be that science with its analysis robs us of the pleasing sense of awe and mystery, but if you dig deep, you will find enough mystery left to satisfy the keenest yearner after half lights and the obscure. At best, science replaces one mystery only by another of grander order."

Now, leaning for a moment from the purely utilitarian side of science, to the esthetic, there comes a joy to the lover of the beautiful that cannot be expressed in words, for he sees in the color of an American Beauty rose the same light waves that tint yonder red star, whose light waves, coming to him at the rate of 180,000 miles per second, left it a thousand years ago; he listens to the varying bell tone of the swiftly moving locomotive and translates it into the motion of stellar worlds, whose distances have never before been measured by the most refined modern instruments.

Does the photographic picture of your favorite landscape lose any of its beauty when you are told that during an exposure in your camera of only one-tenth of a second, from forty to eighty millions of millions of light waves have hammered upon your negatives tending to shatter them to pieces or change their molecular arrangement?

Maclaurin gives a charming illustration to help us better understand what it means when a ray of violet light impinges on our photographic plate for one-tenth of a second. "Imagine you are watching a log floating near the seashore, and that it strikes against a pier as it rises and falls with the waves, say, once in six seconds. In order to correspond to the number of light waves in one-tenth of a second, the log would have to beat against the pier for more than two million years."

But enough for the present of the esthetic side of science, though it has beauties in its every phase, whether it be in the flower or in the rainbow—whether it be in the structure of glass from which the prism is made, or the story it tells of yonder far-off stellar universe. But can we connect this science beautiful with our engineering problems? Why not?

It is a long way from the sun dial of Ahaz to the Riefler clock; a long way from the Pyramid of Cheops to the stately steel structure of our great cities; a long way from the cubit span and handbreadth to the standard metre whose value is determined in light waves; a long way from the ox-cart with its wheels cut from the end of a log to the steam and electric locomotive; a long way from the log bridge across the narrow streams to the magnificent steel and concrete spans that now cross our widest rivers; a long way from the tiny Egyptian cedar boat, built without iron, steel or copper, to the majestic steel steamships that daily cross the ocean; a long way from the smoke balloon of Montgolfier to the dirigible, or the biplane that soars like a bird through our skies; yes, and all along the pathway of the evolution of these and other inventions of man are unwritten histories which, if told, would be filled with romance and ofttimes with tragedies,—aye, with the sacrifice of the lives of many of the world's noblest pioneers who have contributed to the treasures of our vast storehouse of knowledge.

We can look over time's vistas and learn that to Aristotle we owe the beginning of exact science and research, fostered by his love of experimentation. Indeed, Hale records that, although his theory of the fixity of the earth was based upon false premises, he left his impress so deeply upon the islands and borderlands of the Mediterranean that, after he had passed from his labors, there was a gradual separation of the investigations in natural science from the speculations of the philosophers, and that true scientific research, in the modern sense, developed with remarkable rapidity.

I have always had the conviction that we give too little thought to the pioneers in any line of research, and I have often bowed my head in reverence to those who, with the most limited means and equipment, by patient, persistent plodding, wrested the secrets of nature from their entanglement.

There has often arisen in my own mind, and no doubt in yours, the question: Where shall we draw the line between pure and applied science? For myself, I have been unable to find aught but a hazy line of demarkation, and in my long life I have known but few men who, in their mathematical researches and calculations, could see nothing in the way of an entity before them; indeed, but one great mathematician has told me it was his regret that he could see no picture of a beautiful comet while he was calculating its orbit, nor could he form a visual image of the planetary world whose perturbations he was computing by his wonderful mathematical formulae.

Nor can I see how pure science and its allies can be relegated to opposite poles. For a time apparently they may point in opposite directions, but at each end there lies some great truth. One may be a hitherto undiscovered law of nature and the other, let us say, something of an utilitarian character, that, material as we may call it, will cause these opposite poles to curve toward each other and add to the sum of human knowledge, human betterment and happiness.

Let me illustrate. When the velocity of the propagation of light waves was determined by scientific reasoning and experimentation of the most refined nature, and successively by Roemer, Bradley, Fizeau, Foucault, and our own Newcomb and Michelson, the process of solving the problem remained for a long time in the domain of the exact sciences as a masterpiece of the human mind.

But who of us dreamed to what an utilitarian purpose these light waves would be made subservient? The genius of a Michelson carried them into the workshop, thence to the International Bureau of Weights and Measures at Sèvres, and gave us a value for the International metre in terms of light waves that will remain absolutely unalterable as long as this old world moves in the lumeniferous ether of the universe. And so we now know that our standard metre, measured in terms of the red radiations or wave length of cadmium, equals 1,553,163.5; for the green radiation, equals 1,966,-249.7; and for the blue radiation, 2,083,372.1, in air at 15 deg. cent. and normal atmospheric pressure.

Michelson tells us that the absolute accuracy of these measures is one part in two millions, the relative accuracy, one part in twenty millions.

Getting nearer the utilitarian service of the scientific study of light waves, let me say that Dr. Anderson of the Johns Hopkins University has utilized them in making screws of hitherto unheard-of accuracy. Let me quote from a letter received from Dr. Anderson before writing this address:

> In reply to yours of the 4th, I will say that from measurements made by using a Fabry and Perot Interferometer, the screws we have made for ruling purposes have the following accuracy:
>
> 1. The maximum variation in pitch along the screw did not exceed *one-ten millionth of an inch.* (By pitch, I mean the average value of the pitch as given by a well-fitting nut three or four inches long.)
> 2. The axis of the screw had a radius of curvature longer than *250 miles.*
> 3. The axis of each pivot or bearing coincided with the axis of the screw to within one two-hundred thousandth of an inch.
>
> These are three important quantities as far as the screw itself is concerned. When the screw is mounted ready for use, it is also important that it be prevented from moving endwise or longitudinally when it is rotated. Our mounting, using the flat ruby against a spherical steel surface, makes it possible to reduce this motion to something less than one millionth of an inch. Errors of the magnitude given above can be easily detected with the interferometer.

Our fellow member, George M. Bond, has given us a most valuable compendium of the development of measures of precision in his work on "Standards of Length," from which I quote as most interesting, and bearing forcibly upon this part of our topic:

It is worthy of note that a remedy for the evil complained of by master car builders, that nuts made by some firms or at some shops would not screw on bolts made by others, at first baffled the ability of the most prominent manufacturers of tools of precision in the country, and that to provide an adequate remedy it was necessary to secure the assistance of the highest scientific ability in the country, which was supplied through the coöperation of the Professor of Astronomy of the oldest and most noted institution of learning in the land. The man of science turned his attention from the planets and the measurement of distances counted by millions of miles, to listen to the imprecation, perhaps, of the humble car-repairer lying on his back and swearing because a ⅝ nut—"a leetle small"—will not screw on a bolt—"a trifle large." It is a striking example of the assistance which science can give in conducting the "practical" affairs of life.

Here I wish to pay a tribute to our American engineers who have developed instruments for mechanical measurements to such a high state of precision, which in their turn have been such valuable factors in the development of interchangeable machinery. I need not dilate upon these interesting topics, but I cannot refrain from offering a word of praise to those earnest men whose ideals have been of the highest—by their work we know them.

I am sure all will be interested in what may be called a paradoxical statement—but is not—namely, that through the marvelous precision attained by Rowland, and later by Anderson, in the construction of accurate screws, we have made what must be considered an utilitarian use of pure scientific research. This utilitarian use of science has reacted, as it were, and enabled the scientific mechanician to produce an optical device that rivals, if it does not surpass, the telescope in unraveling some of the most profound secrets of the universe.

I hold in my hand (shows a diffraction grating) a little instrument called a diffraction grating. On the plane surface of this polished plate, made accurate to one-tenth of a light wave, or within one-tenth of one forty-five thousandths of an inch, are ruled more than 45,000 lines between which there is no greater error than one two-millionths of an inch. With this delicate piece of apparatus, made possible *first* by rigorous scientific research, *second* by the skill of the artisan, *third* by a knowledge of and vigorous care to avoid temperature changes and *fourth* by the accuracy of the mechanism which includes the accurate screw mentioned above, the astrophysicist has been able to tell us the composition, temperature and distance of the stars. It is also possible for the physicist, the chemist, to tell us the purity of the material he is called to investigate; indeed, it makes itself subservient to many phases of engineering in the domain of metallurgy. And the end is not yet. Where can we then draw a sharp line of demarkation between pure science, and its relation to any and every form of engineering?

Twenty-seven years ago I was the guest of my friend Sir James Dewar, the worthy successor of Tyndall in the Royal Institution of Great Britain. Taking a tiny piece of apparatus out of the "holy of holies," and placing it in my hands, he told me it was the father and mother of all the dynamos and electric lighting systems of that day. *It was the first little dynamo made by Michael Faraday.* What has come to us since that visit, in the domain of the electric development through scientific investigation and mechanical devices!

Thirty-seven years ago I listened to the first faint telephone message over a few miles of wire. On the 24th of January last, through the courtesy of our friend Mr. Carty, chief engineer of the American Bell Telephone Company, and Dr. Bell, I listened to the voices of Watson and Moore across the continent. When this Society was the guest of the officials of the Panama-Pacific Exposition, I was taken to the private office of the telephone company in the Exposition building, where, through the courtesy of our friend Engineer Carty, I not only listened to cheerful words spoken in this city, but heard the sound of breakers as they dashed upon the shore of the Atlantic.

When we made the first little spectroscope to determine the moment when the last ounce of carbon had disappeared from the Bessemer converter, little did I dream that through science, aided by a delicately accurate instrument, devised by my good friend, Dr. George Ellery Hale, and made at "the little workshop on the hill," we would be able to plunge it, as it were, into a storm on the sun and photograph the burning hydrogen or any other element in that maelstrom of fire, the temperature of which this earth knows no correlative.

Here let me add some altruistic words of our good friend Dr. Pritchett:

> The last fifty years have seen a greater betterment of the theoretical basis of physical science than ever before in the history of the world. This development has been marked by a notable stimulation of scientific research, a differentiation of scientific effort, and the creation thereby of a great number of special sciences or departments of science. The possession of a secured

theoretical basis and the intellectual quickening which has followed it have resulted in the application of science to the arts and to the industries in such measure as the world has never before known. These applications have to do with the comfort, health, pleasure and happiness of the human race, and affect vitally all the conditions of modern life.

As members of this association we may well be proud of our Bureau of Standards, organized and brought to its present high state of efficiency by our fellow member, Dr. Stratton, and his splendid corps of associates. This scientific department of our Government, concerning as it does almost every phase of scientific research, valuable to the engineer of every calling, has made a record even greater than the Institute at Charlottenberg and the famous Bureau International des Poids et Mesures at Sèvres, though they have done splendid work for a half century or more. I am certain that every member of this association would place a very high value upon the scientific studies of our Bureau of Standards did they know how closely they are related to their profession.

Other scientific departments of the Government, such as the United States Coast and Geodetic Survey and the Smithsonian Institution, have in the past and are continually contributing to the sum of knowledge that is of value to the engineering profession in some of its many phases. Nor can we pass over the splendid research work of the Carnegie Institution at Washington without a tribute of praise for its great accomplishments in the past and its present activities in the domain of scientific investigation of the highest value to engineering and its correlated interests.

It would be a serious oversight in this paper did I not call attention to what I may name humanitarian engineering investigation, based upon a phase of scientific research we would probably consider far removed from engineering problems *per se*. I refer to the magnificent, and now classic applied scientific medical research of my dear departed friend, General Sternberg, and General Gorgas, who made possible the carrying out of one of the greatest engineering projects the world has ever known. All honor to these good men!

But I must stop. The great, the illimitable field of truth opens up before us; aye, I love to liken it to the "Widow's Cruse;" take from it as you will, it will never be emptied of its priceless treasures.

And is there not a splendid field opened up to us in the endowment of engineering research, now made possible by the gift of one of our honored presidents? I have said little of specific lines of investigation in relation to engineering, having given only a few instances to show their close and intimate relationship, and it needs no further words of mine to verify the aphorism that "the field is ripe for the harvest."

In closing may I quote from the recent publication by Dr. Hale, entitled: "National Academies and the Progress of Research:"

> But the average man of business is much better able to appreciate the value of research directly applied to the improvement of manufacturers than to comprehend the more fundamental importance of pure science. We must show how the investigations of Faraday, pursued for the pure love of truth and apparently of no commercial value, nevertheless, laid the foundations of electrical engineering. If we can disseminate such knowledge, which is capable of the easiest demonstration and the most striking illustration, we can multiply the friends of pure science and secure new and larger endowments for physics, chemistry and other fundamental subjects.

Dr. Hale's valuable brochure can be found in our library and is worthy of our careful study. Here we have a striking note of the value of scientific research, and a touch of its utilitarian value, but after all is said, a knowledge of the true and beautiful, whatever its bearing may be, stands for human progress, human betterment and human happiness.

May I also quote from a letter received from the same gentleman by the donor of our research fund:

> I am delighted that you, with a thorough knowledge of the situation, agree with me in thinking that the time has come for much closer coöperation between engineers and men of science. Previously I have not been able to judge the matter from the standpoint of the engineer, but it was plain to me that the separation between pure and applied science, which seems to have been increasing, must be harmful to both. The man of science is liable to underestimate the importance of practical applications of his subject, while the engineer is in danger of forgetting that if research were conducted only in the hope of securing profitable ends, most, if not all, of the great discoveries which lie at the very foundation of both pure and applied science, would never be made.
>
> Thus every effort should be made to encourage and develop research in both pure and applied science, for neither can be neglected without loss to the other. The dual nature of this problem is thus perfectly evident. Sound strategy requires that the attack be made simultaneously from both sides, since in this way all difficulties can be overcome far more quickly than by two successive attempts.
>
> These difficulties are likely to be due mainly to the conservative tendencies of both engineers and men of science. By establishing your research fund

you have provided means to obviate the chief practical hindrances on the engineering side, where the way has always been paved by the successful research of the General Electric Company and various other large corporations. It now remains to accomplish an equally important advance on the side of pure science.

Dr. Hale is very anxious that there shall be a closer relationship with the National Academy of Sciences, and I think I can say without hesitation that it will be a wise move upon the part of all engineering societies to appoint a special committee to act in conjunction with the Academy toward this most desirable affiliation.

In my desire to be brief, I have refrained from going into the many, very many fields where pure science may take the hand of fellowship of the engineer, and now that there has come to us a grand opportunity, fostered by the splendid research fund placed in our keeping, may we look forward with hope to the day when science and engineering will join hands for the betterment of our loved profession.

"Today we are learning but single notes, tomorrow we will blend them into chords, the hour will chime when all humanity shall know the law of harmony—when every note in every chord shall find its part in the sublime oratorio of the Universal life."

No. 1496

HEATING BY FORCED CIRCULATION OF HOT WATER IN TEXTILE MILLS

By Albert Greene Duncan, Boston, Mass.
Member of the Society

Since the days of the old coal stove, three methods of heating textile mills have been in use: *First,* direct circulation of live or exhaust steam at low pressure; *second,* fan or indirect hot air system, using either live or exhaust steam in heating coils arranged in a central battery to heat the air; and *third,* use of hot water through direct heating surface in the various rooms, heating the water with either live or exhaust steam in closed heaters. It is of the last method that this paper treats. With the widely distributed areas which are encountered in large textile mills, the gravity system of hot water circulation, or it might be called the house heating system, is usually impracticable, so that the third method here treated involves forced circulation of the hot water.

2 The paper describes the method of handling the problem of properly heating the widely separated units of the textile mills[1] shown on the outline sketch (Fig. 1). The problem was given careful study, and the relative advantages and disadvantages of all the three methods of mill heating enumerated above were considered.

3 A system of forced circulation of hot water was decided upon for the following main reasons: *a* The mills are primarily driven by water power, and the auxiliary steam plant which, by the way, is in daily operation to carry about 20 per cent of the mill load, reaching a maximum of 50 per cent in times of low water, is located at a central point, the distribution of power being by electricity. *b* The central plant arrangement necessitated the carrying of heat over long distances. Fig. 1 shows that heat must be carried from the power plant,

[1]Harmony Mills, Cohoes, N. Y.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Textiles.

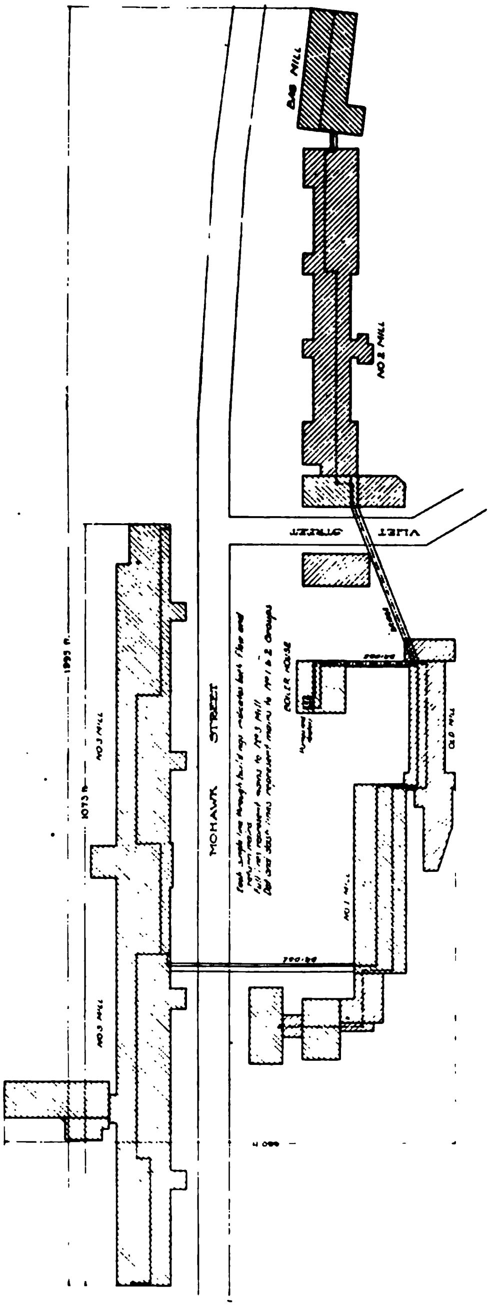


FIG. 1 PLAN OF HARMONY MILLS, COHOES, N. Y.

a distance of 1650 and 1370 ft., in opposite directions, respectively. These distances are not measured in an air line, but owing to the location of buildings, the crossing of streets, and other peculiarities such as power canals, the mains were forced to pursue a somewhat more roundabout course than if the buildings had been built to fit a proposed heating system. *c* As certain of the mills were located below

FIG. 2 HARMONY MILLS, COHOES, N. Y., LOOKING NORTHEAST

FIG. 3 HARMONY MILLS, COHOES, N. Y., LOOKING SOUTHWEST

the level of the power plant a gravity system of return, either of hot water or steam, was impracticable.

4 The mills consist of seven buildings, varying in size from 40 by 100 to 1073 by 76 ft., and an office building; they contain 825,434 sq. ft. of floor surface and have a cubic content of 10,351,000 cu. ft. The buildings present a somewhat typical condition in cotton mills,

where buildings have been built of various types over a period of many years. The mills under consideration still use a building of three stories put up in 1837, and they occupy subsequent buildings, including their largest mill consisting of five stories with basement, built in 1876. Figs. 2 and 3 are general views of the mill buildings. In both figures the location of the central power plant is clearly shown by the modern concrete stack in the center. The other stacks are those of abandoned boiler plants used under an old form of direct heating by live steam.

5 The heating system was divided into two sections for convenience of regulation. The general direction of the mills was north and south, and the east side of the longer mill was on the top of a bluff 80 ft. above the river and exposed to extremely high winds. (See Fig. 3.) The mills being located in a northern latitude, where atmospheric temperatures of 20 deg. fahr. below zero are not uncommon, the question of different exposures of various portions of the plant was a special problem, the method of solution of which will be taken up in due course. Further than this, the plant, though comprising many buildings, naturally divided itself by its configuration into two manufacturing groups approximately equal in area; and consideration was given to the possibility, in times of curtailment, of shutting down either of these groups and running the other at full capacity. In the pumping plant one spare unit was provided to act as a reserve or relay for either of the two systems which, by manufacturing or weather conditions, need temporarily to be reinforced.

6 Given a power station in a central location and for distributing electric power where needed to supply the deficiencies of water power, the possibilities of centralized regulation of heat were naturally very apparent. From experience gained from this installation, the author has no hesitancy in stating that excessive use of coal in any mill heating problem is largely a direct consequence of the overheating of mills. With a distributing steam coil always at a high temperature, the local regulation of heat is generally left to the individual overseer, whose method of lowering the temperature is usually that of opening windows rather than closing valves. It was early determined in the present installation that the regulation of the heat in the various departments of the mills should not be left to the individual overseers, but should be placed under the charge of the engineer in the power house, long distance reading thermometers being there installed so that the temperature in different portions of the plant could be read

at the power house. By this method, the two primary factors of heat control, namely, the temperature of the water and the speed of its circulation, could be accurately regulated in accordance with outside temperature.

7 While there was no engineering objection to solving the above problem by any one of the modern systems of direct steam heating, it was felt that the ability to control the temperature of water from a range of 100 to 240 deg. fahr. gave a flexibility to the system impossible of attainment in any other way. Practically stated, a steam system will operate economically only at full load, while a forced circulation of hot water can be readily regulated to meet all manufacturing and weather conditions and will maintain its efficiency at any degree of temperature.

8 Hitherto, only the conditions of operation have been considered, and it may now be desirable to discuss the questions of relative cost of installation, as well as cost of operation, of the system under consideration and a modern system using direct steam radiation.

9 A comparison of the amount of radiating surface required for a forced circulation hot water system with direct steam systems shows this interesting condition: If exhaust steam at 2 or 3 lb. per sq. in. pressure were used in a steam system as a heating medium, the amount of radiation required would be practically the same as with a forced circulation hot water system, since the temperature of the heating surface would be practically the same. In extremely cold weather, with hot water the temperature of the surface could, however, be raised to as high as 240 deg., which would correspond to the temperature obtained at 10 lb. steam pressure. This temperature could only be obtained in the steam system by raising the back pressure or injecting live steam. The amount of radiation required for live steam heat diminishes as the steam pressure used rises. However, a point in favor of a decreased amount of actual radiating coils in a hot water system is that the mains can be figured as part of the heating surface, the circulation being so rapid that the temperature is adequately maintained to the end of the run with none of the choking and heat losses due to condensation were steam mains are used in the same manner.

10 The method of determining the amount of radiating surface required in various departments for a forced circulation system of hot water does not differ materially from the well-known methods employed in determining the heating surface of the number of heat

units required for any other method of heating. Well-known factors were used in the installation described, and those in Table 1 are submitted as a basis for future determinations, bearing in mind that many modifying conditions apply which must be decided by the judgment of the engineer in each individual case.

11 The difference in temperature figured in the case under consideration was 70 deg. and the heat as that required to heat the mills to 70 deg. in zero weather with temperature of water 212 deg. For temperatures below zero, it was considered that the ability to heat water to 240 deg. and the heat given off by mill machinery in operation were ample margins of safety.

12 The engineer will find plenty of opportunity for the exercise

TABLE 1 HEAT LOSS IN B.T.U. PER HR. PER SQ. FT. OF EXPOSED OUTSIDE SURFACE OF BUILDING FOR 1 DEG. FAHR. DIFFERENCE IN TEMPERATURE BETWEEN INSIDE AND OUTSIDE AIR

Medium	Heat loss. B.t.u.
8 in. solid brick wall	0.40
12 in. solid brick wall	0.31
16 in. solid brick wall	0.26
20 in. solid brick wall	0.23
24 in. solid brick wall	0.21
28 in. solid brick wall	0.19
Single window	1.00
Double window	0.50
Slate on tight wood roof	0.30
Standard mill roof 2½ in. plank T. and G.	0.18

of his judgment in determining the percentage loss of heat to all for leakage, window exposure, roof exposure, and other factors pending in each case upon the location of the individual building.

13 Many other problems arise from the character of cotton m operation, and as a test of the judgment of the engineer may mentioned the loss of heat from warm buildings by the opening doors into cooler hallways and elevator shafts to allow of transpor tion of stock in process, loss by the removal of warm air from pic rooms by the picker fans and loss by circulation of air from the pic rooms to communicating rooms.

14 In comparing the cost of a forced circulation hot water syst with that of a good exhaust steam system, it may be said that w the exception of the additional cost of the pumping plant required the former, the cost of installation of either is about the same. T cost of maintenance of the hot water system is believed to be mu

the lower by virtue of gradual rather than sudden changes of temperature of radiating surface and absence of water hammer.

15 The amount of radiation in the case at point was also calculated on the basis that the system would be adequate to heat up the mill following a shutdown of several days at zero temperature. On this account, no allowance was made for the well known fact that cotton machinery gives off a great deal of heat during operation. Consequently, the theoretical water temperature necessary to heat the mills was found to be many degrees too high during ordinary running conditions.

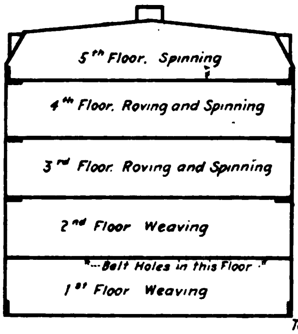
16 Fig. 4, which refers to one of the largest of the mill units, will make clear the proportioning of heating coils. The departments in this five story mill are shown in transverse section, together with the location of the heating coils on the side walls or ceilings. In this building there are about 1500 belt openings between the first and second floors. The table referring to each floor shows in the first column the heat required without taking into consideration leakage and wind exposure, in the second column the heat required with these factors allowed for, in the third the heat actually provided for in proportioning the heating surface and in the last column the percentage each floor contained of the total amount of heating surface in the building. It will be seen that the figures in the third column are not in the same ratio as those in the second column, although the totals as the same. The character of manufacturing operation in each department was a determining factor, and experience with the system later showed that more coils were placed upon the top floor and fewer upon the lower floor than were actually needed, this being due to belt openings mentioned above and to the natural effect of heat leaking up through elevator shafts and other passages to the top floor of the mill, which also contained the spinning machinery, the kind of cotton machinery producing the most heat.

17 As may be inferred, the heart of a forced circulation hot water system is the pumping plant. A drawing of the pumping layout in these mills is reproduced in Fig. 5. The essentials of each unit of the plant are a closed heater of the ordinary feedwater type, together with a pump. Two pumps are normally used, one for each section of the mills. Each is connected to its own heater and is so arranged that in case of breakdown the third pump or heater shown can be used interchangeably with either of the two sections of the system. As to the type of heater for these installations, one with a

short coil is preferable, to be run with the steam inside the coils surrounded by the water in the shell; this is an essential feature of the system, as the heater coil must be so proportioned as to clear itself readily of the water of condensation, which, at times of starting-up especially, is large in volume.

18 The heaters for this installation were of extra large capacity and adapted to use exhaust steam to heat the circulating water in any but the most extreme weather, when live steam is used. Their strength, however, was proportioned to receiving steam at full boiler pressure of 180 lb., so that in case of an accident to a valve when using live steam no danger could arise from explosion.

19 The three heaters were installed in connection with a boiler feedwater heater of the same size; and by a proper arrangement of



	1	2	3	4
	Btu Req'd without Leakage & Wind Exposure	Btu Req'd with Leakage & Wind Exposure	Btu actually Provided For	Percentage on each Floor to Total Radiation
	1,810,000	2,672,100	1,778,000	24.0
	1,003,000	1,308,900	928,700	12.5
	877,400	1,158,000	1,348,100	18.1
	877,400	1,158,000	1,151,400	15.5
	865,000	1,128,900	2,219,700	29.9
Total	5,432,800	7,425,900	7,425,900	100 Per Cent

FIG. 4 SECTION OF MILL UNIT, AND PROPORTIONING OF HEATING COILS

valves, a greater or less amount of the exhaust steam furnished by the pumping plant and auxiliaries of the station could be used in the circulating water heaters or the feedwater heater as was required. The temperature of the circulating water could thus be maintained at any desired value.

20 Whether the heaters were using live or exhaust steam, the drips were collected in an open heater to which the make-up water of the boiler plant was added and thus were pumped back through the closed feedwater heater to the boilers.

21 In the case under consideration, steam turbine driven centrifugal pumps were used. No rule can be laid down as to the proper form of pump required, but each case must be decided by the character and service of the power station of which the system is a part. In the present instance, the heating plant was installed in a station used

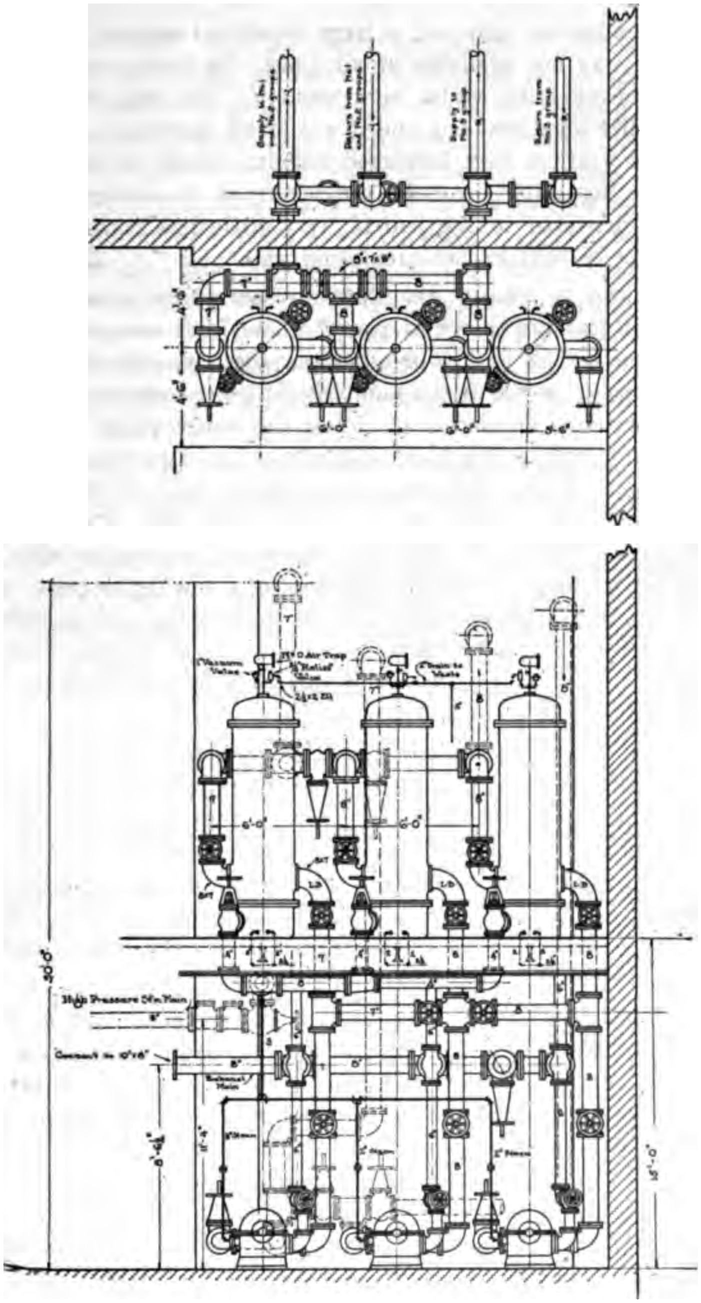


FIG. 5 HARMONY MILLS HEATING SYSTEM PUMPING PLANT

only as an auxiliary unit, and a large amount of exhaust steam from auxiliaries was not available at all times, the main steam turbine being run condensing under high vacuum. For this reason steam driven pumps were installed, and it was found in practice that, except in extreme weather, they furnished sufficient steam to heat properly the circulating water. As the returns from the heaters were immediately delivered to the boilers, very little loss occurred in the transference of heat to the circulating water.

22 It can be readily seen that, in cases where a power station is used as a primary source of power, where large amounts of steam are used for service pumps or other purposes, an electrically driven or other form of power driven pump might prove most desirable. But it is the author's opinion that, unless the power plant in question operates on a twenty-four hour basis, a method of pumping by steam for a portion of each day should be installed, as it is the universal experience of textile mills that the larger proportion of the heat required during the twenty-four hour day must be supplied when the mill is not running. With the exception of a few departments, such as weaving, in any but the most extreme weather the heat engendered by the machinery is sufficient to keep the rooms at working temperature after 8 or 9 A. M. Details of this gain in heat from the machinery will be considered in due course.

23 As the system centers round the pumping plant, this was necessarily installed at an additional cost over what would be required for a system of direct steam heating, but the advantages of centralized control and steady and uniform temperatures more than offset the additional cost entailed. Although the amount of this additional cost would vary in different plants, it may be stated that in the example under consideration the pumping plant cost 29 per cent of that of the total system.

24 After three years of operation, careful records can be given showing the result of the system in actual service. The steam pipe supplying the turbine pumps, from which also the live steam was taken for the heaters in case of need, was furnished with a steam meter and, as a check, all other lines in the power plant leading to the main turbine, for mill uses and other purposes, were also metered. The total result was checked by a venturi meter on the feedwater side, and over a period of years the sum of these metered results checked within approximately 5 per cent, an amount which could be easily accounted for by boiler blow-offs, leakage, emptying of boilers, and

other minor losses. Another meter was placed in the branch line feeding the circulating pumps alone, so that by subtracting the amount of steam required to drive the pumps, the steam going direct to the heaters could be accurately determined. A third meter in connection with the heating system was installed in the exhaust pipe as a check upon both the amount and the direction of flow of steam, as to whether or not the heaters were receiving exhaust steam from power plant auxiliaries, or whether exhaust from the circulating pumps was being put into the feedwater heater supplying the boilers. Furthermore, two other flow meters measured all the water circulating through each of the two parts of the system, and these two main pipe lines were equipped with recording thermometers, giving the temperature of flow and return in each.

25 During the first year, 1912-13, considerable attention was given to the compilation of data for the purpose of making a thorough study of the conditions, with a view to obtaining the best possible results. During a part of that year very extensive readings were taken throughout the mills of temperature, humidity, amount of radiation in service, number of windows open, if any, etc. At the same time recording thermometers giving the temperature of flow and returns in the power plant were checked up frequently. A summary of the records follows for the three heating seasons 1912-13, 1913-14, 1914-15, respectively, is given in Tables 2, 3 and 4.

26 During the first half of the 1912-13 heating season, neither the mill conditions nor the method of operating the heating system were on a regular working basis. Subsequent to Feb. 1, 1913, no changes of material consequence that would affect the heating requirements or the method of operating the system have been made in the mill. It will be noted from Table 2, covering the year 1912-13, that there is a reasonably close agreement between the total heat supplied to the heating system as shown by the steam passing through the meters and that calculated from the pumpage and temperature drop between flow and return. In arriving at the total heat units in the water, it is of course necessary to multiply the pumpage by the temperature drop on each individual day, for in compiling the average temperature drop for a week, each day has been given equal weight, although all days may not have represented equal quantities of water.

27 During the past two years considerable trouble has been experienced in maintaining accuracy of the recording thermometers, and the temperature drop is not dependable for calculating the total

TABLE 2 OPERATION OF HARMONY MILLS HEATING SYSTEM 1912-1913

					No. 1 Group of Mills							No. 3 Mill							
Week Ending	Average Outside Temp. Deg. Fahr.	Total Steam to Pumps and Heaters 1,000 lb.	Average Boiler H.p. Equivalent 24 Hr.	Total Heat in Steam Million B.t.u.	Per Cent Time System in Service	Total Pumpage Million lb. Water	Temperature Water Flow Deg. Fahr.	Temperature Drop in Ret'rn Deg. Fahr.	Heat Supplied to Mill Million B.t.u.	Mill Temperature Deg. Fahr.	Per Cent Total Radiation in Service During Working Hours	Per Cent Time System in Service	Total Pumpage Million lb. Water	Temperature Water Flow Deg. Fahr.	Temperature Drop in Ret'rn Deg. Fahr.	Heat Supplied to Mill Million B.t.u.	Mill Temperature Deg. Fahr.	Per cent Total Radiation in Service During Working Hours	Total Heat Supplied to Both Mills Million B.t.u.
Nov. 9..	49	627	114	621	43	37.9	164	8.9	338	76	..	25	24.5	152	13.8	341	76	..	679
16.	45	745	131	738	45	36.8	159	8.9	329	74	..	23	22.3	165	11.7	263	76	..	592
23..	41	846	149	836	51	43.3	160	9.6	431	74	..	32	31.7	161	13.4	424	75	..	855
30..	37	928	163	919	49	43.1	165	10.6	460	72	..	37	36.3	169	12.8	465	74	..	925
4 weeks..	43	3,146	138	3,114	47	161.1	162	..	1,558	74	..	29	114.8	162	..	1,493	75	..	3,051
Dec. 7..	41	988	174	980	56	48.3	165	10.4	506	76	..	40	38.1	161	14.1	538	75	..	1,044
14..	30	1,994	350	1,968	99	86.9	170	10.4	902	76	..	99	97.2	165	11.3	1,101	76	..	2,003
21..	36	1,698	298	1,677	99	84.7	153	7.8	658	77	..	99	98.4	152	9.2	887	77	..	1,545
28..	30	1,939	340	1,912	100	87.6	163	9.4	825	77	..	100	98.6	164	11.1	1,094	77	..	1,919
4 weeks..	34	6,619	290	6,537	89	307.5	163	..	2,891	76	..	84	332.3	161	..	3,620	76	..	6,511
Jan. 4..	39	1,269	223	1,255	100	85.3	150	7.4	614	75	68	92	90.5	149	8.0	720	77	54	1,334
11..	31	1,427	251	1,410	98	77.3	153	7.7	586	73	61	90	86.3	151	9.5	834	75	64	1,420
18..	36	1,014	177	994	99	68.9	137	5.6	420	75	67	69	62.4	143	8.8	557	77	60	977
25..	36	1,015	178	1,000	100	73.2	135	5.4	410	75	61	92	83.3	136	7.3	623	77	68	1,033
Feb. 1..	33	1,108	194	1,091	100	71.6	140	6.0	429	74	61	93	84.8	138	7.8	677	77	70	1,106
5 weeks..	35	5,833	205	5,750	99	376.3	143	..	2,459	75	64	88	407.3	143	..	3,411	77	63	5,870
Feb. 8..	21	1,756	306	1,719	100	85.8	156	8.4	726	75	72	99	96.1	158	10.1	969	78	62	1,695
15..	17	1,877	327	1,838	99	83.6	161	10.4	848	75	80	99	91.2	160	10.6	981	78	62	1,829
22..	27	1,405	246	1,381	96	74.9	152	8.2	606	75	75	86	77.7	147	9.1	724	77	59	1,330
Mar. 1..	25	1,437	253	1,423	100	78.4	156	8.0	647	76	74	94	87.9	144	8.8	788	78	56	1,435
4 weeks..	22	6,475	283	6,361	99	322.7	156	..	2,827	75	75	95	352.9	152	..	3,462	78	60	6,289
Mar. 8..	22	1,562	275	1,546	93	77.0	154	9.0	704	74	75	93	83.2	140	8.7	723	77	59	1,427
15..	45	535	94	530	65	43.6	134	6.7	256	..	..	59	45.5	125	9.0	366	..	..	622
22..	43	375	66	371	56	40.4	130	4.7	194	..	..	34	27.1	130	5.8	156	..	..	350
29..	44	425	75	421	65	42.6	128	5.1	212	..	..	36	30.5	124	5.1	171	..	..	383
4 weeks..	38	2,897	128	2,868	70	203.6	136	..	1,366	..	..	55	186.3	130	..	1,416	..	..	2,782
Apr. 5..	46	347	61	344	67	40.8	130	4.8	142	..	..	41	32.0	124	6.0	184	..	..	326
12..	41	454	80	449	80	49.2	129	4.6	206	..	..	40	34.2	127	7.4	257	..	..	463
19..	53	139	25	138	37	18.1	125	4.9	49	..	..	13	10.8	118	9.0	76	..	..	125
26..	57	128	23	127	37	20.6	124	5.3	71	..	..	13	10.9	117	5.7	50	..	..	121
4 weeks..	49	1,068	47	1,058	55	128.7	127	..	468	..	..	27	87.9	121	..	567	..	..	1,035
25 weeks.	37	26,038	184	25,778	78	1,499.9	148	..	11,569	..	..	64	1,481.5	145	..	13,969	..	..	25,538

TABLE 3 OPERATION OF HARMONY MILLS HEATING SYSTEM 1913-1914

Week Ending	Average Outside Temperature Deg. Fahr.	Total Steam to Pumps and Heaters Thousand Lb.	Average Boiler H.p. Equivalent 24 Hr.	Total Heat in Steam Million B.t.u.	No. 1 Group of Mills					No. 3 Mill				
					Per Cent Time System in Service	Total Pumpage Million Lb. Water	Temperature Water Flow Deg. Fahr.	Temperature Drop in Return Deg. Fahr.	Mill Temperature Deg. Fahr.	Per Cent Time System in Service	Total Pumpage Million Lb. Water	Temperature Water Flow Deg. F.	Temperature Drop in Return Deg. F.	Mill Temperature Deg. Fahr
Nov. 8...	45	212	37	210	24	15.5	125	5.2	73	20	15.6	119	6.8	69
15...	41	318	56	315	28	21.2	122	8.3	74	29	25.2	114	8.3	68
22...	47	340	60	337	29	18.9	124	12.7	76	28	20.9	119	11.3	73
29...	38	637	112	631	48	33.7	125	7.7	72	47	41.2	123	14.0	70
4 weeks...	43	1,507	66	1,493	32	89.3	124	....	74	31	102.9	119	....	70
Dec. 6...	41	645	113	638	47	32.7	136	8.6	75	46	39.3	129	12.7	72
13...	30	1,019	179	1,008	85	47.1	128	6.6	76	91	66.3	125	10.0	73
20...	35	1,047	184	1,036	81	46.7	131	8.0	73	82	58.1	130	9.9	74
27...	30	1,104	194	1,093	84	54.9	135	9.7	69	81	62.1	133	11.5	73
Jan. 3...	17	1,762	310	1,743	96	61.1	146	11.2	69	96	74.5	149	13.8	73
5 weeks...	30	5,577	196	5,518	79	242.5	135	.....	72	79	300.3	133	....	73
Jan. 10...	30	1,230	216	1,218	93	55.8	146	11.6	71	74	47.7	136	11.7	76
17...	10	2,219	372	2,199	95	62.5	171	18.5	70	90	62.7	163	18.7	72
24...	22	1,362	239	1,349	89	56.8	161	17.4	76	81	62.7	139	12.0	74
31...	30	1,006	177	996	83	65.0	153	13.7	74	60	46.5	123	8.5	76
4 weeks...	23	5,817	251	5,762	90	240.1	158	....	73	76	219.6	140	....	74
Feb. 7...	31	806	142	798	85	60.3	150	12.1	75	62	47.6	128	11.9	76
14...	9	1,789	318	1,771	91	62.4	169	19.4	70	85	62.9	146	13.3	77
21...	8	1,961	345	1,941	99	63.8	179	19.1	75	99	77.2	148	12.5	78
28...	14	2,134	266	2,113	97	59.8	183	21.7	77	90	65.0	163	15.5	77
4 weeks...	15	6,690	268	6,623	93	240.3	.170	....	74	84	252.7	146	....	77
Mar. 7...	31	1,488	261	1,473	87	50.9	168	17.7	72	80	56.1	150	12.3	78
14...	26	1,563	275	1,547	91	54.6	171	18.8	75	90	64.0	155	12.7	76
21...	32	1,377	242	1,363	87	53.4	161	18.0	....	79	56.6	144	12.3	....
28...	38	1,166	205	1,154	70	41.8	164	18.2	....	72	47.4	145	12.2	....
4 weeks...	32	5,594	246	5,537	85	200.7	166	....	73	80	224.1	148	....	77
Apr. 4...	38	1,120	197	1,109	92	50.4	155	16.7	74	76	50.5	146	12.5	75
11...	39	1,081	190	1,070	75	42.2	162	16.8	72	71	44.5	144	10.2	74
18...	44	658	116	651	63	32.2	141	14.0	69	61	35.8	126	9.2	75
25...	50	121	19	120	18	7.5	128	14.8	67	6	4.2	118	....	....
May 2...	49	190	33	188	15	9.1	133	15.9	....	13	9.9	111	9.7	....
5 weeks...	44	3,170	111	3,108	53	141.4	144	....	70	45	144.9	129	....	75
26 weeks...	31	28,355	192	28,041	72	1,160.3	149	....	73	66	1,244.5	136	...	74

TABLE 4 OPERATION OF HARMONY MILLS HEATING SYSTEM 1914-1915

Week Ending	Average Outside Temperature Deg. Fahr.	Total Steam to Pumps and Heaters Thousand Lb.	Average Boiler H.p. Equivalent 24 Hr.	Total Heat in Steam Million B.t.u.	No. 1 Group of Mills: Per Cent Time System in Service	No. 1 Group of Mills: Total Pumpage Million Lb. Water	No. 1 Group of Mills: Temperature Water Flow Deg. Fahr.	No. 1 Group of Mills: Temperature Drop in Deg. Fahr.	No. 1 Group of Mills: Mill Temperature Deg. Fahr.	No. 3 Mill: Per Cent Time System in Service	No. 3 Mill: Total Pumpage Million Lb. Water	No. 3 Mill: Temperature Water Flow Deg. F.	No. 3 Mill: Temperature Drop in Return Deg. F.	No. 3 Mill: Mill Temperature Deg. Fahr.
Nov. 7...	49	242	43	240	27	20.2	117	7.2	73	18	15.3	108	5.8	75
14...	39	569	100	563	58	43.3	125	10.6	72	43	39.7	111	7.8	74
21...	33	961	170	952	78	63.1	136	13.1	73	71	57.1	117	8.8	73
28...	32	1,333	236	1,320	97	72.0	150	26.4	74	89	70.5	128	11.8	77
4 weeks...	38	3,105	137	3,075	65	198.6	•132	....	73	55	182.6	116	....	75
Dec. 5...	41	638	113	632	76	49.4	137	23.0	74	66	43.1	110	5.3	80
12...	31	1,203	212	1,191	96	69.4	153	28.1	74	86	63.1	126	12.7	80
19...	24	1,667	294	1,650	100	70.7	173	27.8	76	95	73.9	127	15.1	77
26...	16	1,972	348	1,952	100	68.4	173	32.7	73	99	75.6	142	18.8	77
Jan. 2...	20	2,379	419	2,355	100	74.0	169	14.2	75	100	85.5	157	15.5	78
5 weeks...	28	7,859	277	7,780	94	331.9	161	....	74	89	341.2	132	....	78
Jan. 9...	27	1,623	286	1,607	88	64.8	154	11.4	75	87	69.2	143	10.7	77
16...	31	1,323	234	1,310	87	63.1	149	9.5	73	83	65.8	139	7.0	81
23...	32	1,103	194	1,094	73	51.5	153	11.2	75	67	53.1	136	6.7	78
30...	23	1,424	252	1,410	84	55.3	155	11.1	74	81	60.7	146	10.2	78
4 weeks...	28	5,473	241	5,421	83	234.7	153	....	74	80	248.8	141	....	79
Feb. 6...	20	1,876	331	1,857	87	57.3	160	11.4	76	82	60.7	152	11.3	83
13...	26	1,399	246	1,385	95	58.4	146	10.7	74	91	65.1	131	7.9	77
20...	32	1,202	212	1,190	83	47.2	148	11.2	76	74	53.3	134	10.0	82
27...	36	750	132	743	63	35.1	128	10.2	76	48	34.0	124	9.6	84
4 weeks...	29	5,227	230	5,175	81	198.0	146	....	75	74	213.1	135	....	82
Mar. 6...	25	1,529	270	1,514	92	52.9	152	13.0	75	79	63.4	133	8.4	80
13...	33	972	171	962	81	50.3	138	11.1	73	68	52.8	126	8.6	78
20...	35	1,066	188	1,055	93	57.4	147	10.4	75	64	43.1	125	9.6	80
27...	38	917	161	908	77	44.8	145	9.8	71	61	46.1	125	8.0	80
Apr. 3...	34	1,082	191	1,073	90	55.1	144	9.3	....	65	50.3	132	8.3	81
5 weeks...	35	5,566	196	5,512	87	260.5	145	....	74	67	255.7	128	....	80
Apr. 10...	48	422	74	418	41	25.1	141	9.4	....	24	17.3	121	5.6	....
17...	50	275	48	272	28	15.4	141	10.3	....	24	17.5	121	6.0	....
24...	56	95	17	94	10	6.5	122	11.0	....	5	2.7	102	8.6	....
3 weeks...	51	795	46	784	29	47.0	135	....	....	21	37.5	115	....	....
25 weeks..	35	28,025	198	27,747	73	1,270.7	145	....	74	64	1,278.9	128	....	79

B.t.u. It will be noted that the temperature drop seldom exceeds 10 deg., so that even 1 deg. error in these thermometers would amount to 10 per cent of the heat so calculated.

28 The figures giving the average boiler horsepower for twenty-four hours are significant, and the author believes that no such result has ever before been accomplished in a textile mill of this size. Previous experience in other mills of about half the size of the one under consideration, heated by a combination system of direct steam and indirect hot air, gave results expressed in boiler horsepower of

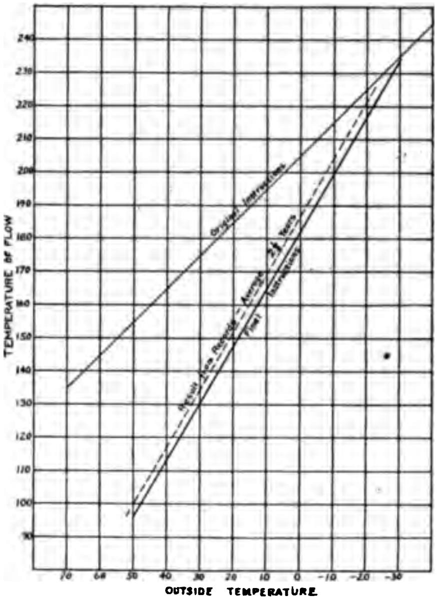


FIG. 6 TEMPERATURE OF WATER FOR VARIOUS OUTSIDE TEMPERATURES

approximately twice the horsepower required in proportion to the size of the respective mills.

29 Figs. 6, 7 and 8 indicate the relations between outside temperature, temperature of circulating water and total steam used. Fig. 6 shows the temperature of water required for different outside temperatures. The upper line represents the original instructions by the engineer of the system and the lower line represents the modified instructions followed by the power plant engineer since Feb. 1, 1913. The dotted line shows the average temperature of water that has been maintained during the past two and a half years. It will be noted that after the operation of the system had become thoroughly stand-

ardized, a marked saving in temperature of water and consequently in coal was effected. This result was accomplished after careful study of the amount of heat given off by manufacturing and machinery, attention to double windows over 80 per cent of the mill, elimination

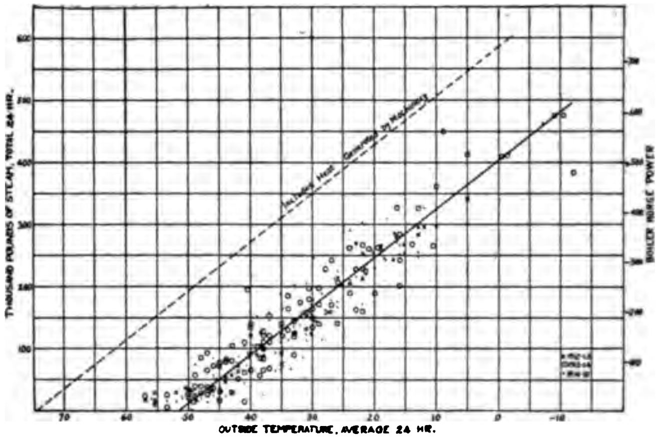


FIG. 7 TOTAL STEAM CONSUMPTION PER DAY

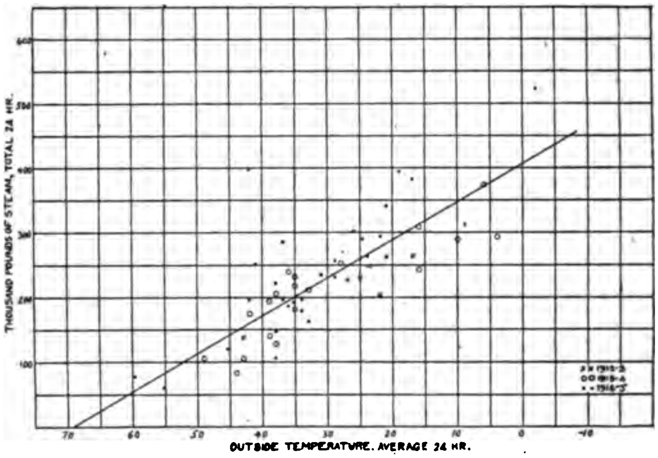


FIG. 8 HEAT SUPPLIED SUNDAYS AND HOLIDAYS

of draughts through stairways and elevator shafts, and any other conditions which prevented the system working at its best efficiency. This proves more than any other figures can the chief claim made at the beginning of this paper that centralized control of heating in textile mills is the method for the most efficient results.

30 Fig. 7 shows a plot of the total steam consumption per day with different outside temperatures during all full working days when the system was in operation during the past two and a half years. Mondays and the days following holidays are excluded, as some extra heat is required to bring the mill temperature back to normal on those days. It is noted from the average curve drawn through these points, that 500 boiler h.p. is required to heat the mills in zero weather. No heat whatever is required when the outside temperature is above 50 to 55 deg., yet a temperature of 75 deg. is maintained throughout the mills. It must be remembered that about 7000 h.p. is used to drive the machinery; all this power is dissipated in heat from the spinning frames and other machinery, and its total heat equivalent is about 180,000,000 B.t.u. per 10-hr. day, or substantially 180,000 lb. of steam. Therefore, if this heat equivalent is added to that supplied to the heating system, we find the upper dotted line extended to the base line would indicate no heat required with an outside temperature of 74 deg fahr., which is substantially the average mill temperature maintained. This, therefore, seems to check reasonably well with the results as given.

31 Fig. 8 gives a plot of heat supplied to the mills during Sundays and holidays during the past two and a half years when the system was in service. It was not the intention to maintain full working temperature of the mills when they were not running and a room temperature drop of 5 to 10 deg. usually took place, which was made up on the day following; therefore, the amount of heat on these days was less than that required to maintain a uniform temperature and that supplied on Mondays and the days following holidays was greater. An average line through the plotted points intersects the base line at an outside temperature of about 70 deg. and shows conclusively that more heat was supplied to the mills on Sundays and holidays than on full working days, even though the same temperature was not maintained.

32 Two other points are brought out by these diagrams, namely, the amount of heat required at zero weather is evidently the same on Sundays as it is on working days, and also there is a much closer agreement between the temperature of circulating water as actually carried and that contemplated by the contractor at zero weather than at higher temperatures. These facts point toward the conclusion that substantially the same heat is required when the outside temperature is zero or below, whether the mill is running or not. The reason for

this is that when the mill is running, an extra circulation of air through the buildings takes place due to the frequent opening of doors, the operation of elevators, etc., thereby causing a much greater leakage or circulation through the mill than takes place on Sundays and holidays when the mill is not in operation. There is a decided chimney action in heated buildings 60 to 75 ft. in height, sucking cold air into the lower floors and discharging it from the upper ones, and this increases very noticeably as the outside temperature approaches zero.

33 During the first year, the question of wind velocities was looked into, but there was no indication that it was a factor of sufficient importance to warrant including it in addition to the outside temperature in the instructions to the engineer.

34 The results from the operation of this system have demonstrated two facts: that the proper heating of textile mills is based on the one hand on a thorough understanding of the engineering problems involved, and on the other hand on a careful watching of losses of heat, which can be most readily accomplished by having the plant under centralized control.

35 Losses in heat arise from many causes. Our experience shows that the benefit of double windows offsets their cost many times, not only on the side of the mill exposed to prevailing winds but on all sides; in the case in question 80 per cent of the mill windows are double windows. Openings from the heated rooms to entry ways and elevator shafts are a constant cause of loss, especially at night, and during the first years of this system constant vigilance was necessary to impress upon night watchmen the importance of keeping all such openings closed. Non-enforcement of this regulation means that all the heat of the mill becomes centered in the upper story, where it is lost by the opening of windows, made necessary to keep the room at a working temperature. If study and proper regulation is made along the above indicated lines, except in extreme weather heating can be confined in the daytime to the lower floors of the mill, to the weaving and carding departments, and to portions of the mill where very little machinery is in operation.

36 Certain other general subjects naturally fall within the limits of the above discussion. The relative merits of heating circulating water by live steam and exhaust is a matter to be decided entirely by the individual plant, as already indicated, and is based upon the conditions of the power plant operation and the available supply o

exhaust steam. Where the amount of steam required for heating or other purposes is a small percentage of the amount required for power, it is not good judgment, in the author's opinion, to interfere with the highest possible efficiency of power production by the use of a small portion of the exhaust steam from the main power unit for heating purposes, and the question involved in that can usually simmer down to a matter of figures.

37 Offices or other rooms which require more heat or heat at different times of the day than is required in the mills and which do not share in receiving heat from manufacturing and machinery, must be separately provided for. These, however, represent usually but a small percentage of the total heat required that the general problem of mill heating is not affected by them.

38 In the mill in question, the office building is a detached structure, and during the first year of operation it is safe to say a large amount of heat was wasted by having this connected to the main mill system, as heat was required in offices on mild days when none should have been used in the mills. This difficulty was overcome by installing a siphon system of heating and circulating the water in this building by a special form of ejector or steam siphon which could be operated or closed off entirely independent of the mill system. The steam so added is condensed and added to the water of the system. This together with the air is taken care of through an expansion tank and relief valve, from which it is returned to the heaters in the power plant. The disagreeable noise encountered by this method was overcome by the simple expedient of two lengths of railroad rubber steam hose.

39 Experience is a valuable teacher, and in the installation described another method of running circulation coils would have been preferable to the one used. The method used was the standard one, running the supply mains to the top floor of the respective mills and then, after distribution through the coils on this floor, returning the water through return drops to the return mains placed in the bottom story; in other words, the distribution was in a vertical direction. Although not affecting the efficiency of the heating system, a method of circulating the water through each floor or department by itself, leading out of and returning to vertical risers and return mains is preferred in a cotton mill using different processes on each floor, as it enables the heat to be put on and taken off of each department by the operation of the fewest possible valves. This can be so sys-

tematized by a study of records that, except in sudden changes of temperature, the heating can be done on a regular schedule independent of the control of the working force in any department. This may seem to modify the point first made as to centralizing control and applies more in textile plants than in any other mills, because of the varying degree of import of heat due to manufacturing processes on the various floors, and especially the upper floors where it is the general practice to install spinning machinery.

ACKNOWLEDGMENT

40 The author is indebted to Ervin G. Bailey, Mem.Am.Soc.M.E., and to Frederick W. Parks, Mem.Am.Soc.M.E., who furnished the tables and calculations.

DISCUSSION

Charles H. Bigelow said he noted that live steam was used for heating the hot water part of the time, and the location of the heater required that all the condensation should be returned to the boilers through the feed pumps, causing an extra expense, which could have been avoided if a closed heater was located in the upper part of the boiler house sufficiently above the water line of the boilers, so that the condensed steam could return into the boiler by gravity, thus eliminating the expense of handling that steam, and making a further saving in the operation. Although the upper floors of the mill would be above this heater, yet the pumps for handling the hot water could be located on the boiler room floor and would not have any extra work to do in handling the circulating hot water on account of the heater being elevated. This should be the most economical method of heating the hot water as the steam would give up its latent heat of vaporization and return to the boiler without any work being done on it. In fact it would probably be more economical than trying to use exhaust steam part of the time, for as Mr. Duncan states, and as the speaker himself has noted considerable of the heating is required when no exhaust steam is available.

In a case of one large mill a hot water system was installed with the intention of heating the water by means of steam bled from a turbine. A reducing valve was also provided for use when the turbine was shut down. The condensation had to be returned to the boilers through pumps, which was a constant expense, and it worked out in practice that live steam was used a great deal more than the bled steam from the turbine, and that portion of the installation was not as economical as had been expected.

F. W. Parks said that a point the paper did not bring out was the comparison of coal cost.

There are very few industries that permit of a unit of heating cost comparison as well as does the textile industry. Mills running on the same grade of goods, and located in about the same climate, can be roughly compared in their heating costs on the basis of spindles, or looms, or production. In few other industries can such a comparison be made.

The cost of heating the mill which the paper refers to is just about half that of any other mill that he knew of, on anywhere near the same grade of goods, with the same necessary processes of manufacture. He said this statement was particularly true of coal costs which he had of mills in New England that are doing their heating by exhaust steam or live steam, and that he based it on figures he had for Mr. Duncan's mills of the cost per spindle, the cost per square foot, the cost per cubic foot, and the cost per loom.

A. F. Ernst asked whether the heating and control of the atmosphere in the mill is automatic, or whether it is left to the operating engineer. He asked also whether there is any attempt made to control the humidity of the mill, and if so, what is the means of humidifying.

The Author. In answer to the question whether the heaters could not be placed on top of the boilers, fed with live steam and the drips returned to the boilers by gravity, I would say that this plan was considered and in fact has been adopted by the author in a previous installation in a smaller plant. In the present case, however, it was desired to arrange the heaters so that they could be readily operated either by live or exhaust steam without the use of traps or other complicated devices.

The power plant in this mill is a steam turbine and it was considered at the outset whether exhaust steam should not be bled from the main turbine exhaust for heating. The knowledge, however, that most of the heating would be done when the turbine was not in operation, decided against this plan.

The power plant has steam driven auxiliaries and in addition to the usual boiler feed pumps and the circulating pumps for the heating system, a combination motor and steam driven unit, consisting of centrifugal injection and air pumps for the main turbine condenser. We had consequently a large and varied amount of exhaust steam from auxiliaries and never attempted to use steam from the main unit. By dividing the load of about 150 horsepower between the motor and steam turbine driving the condenser pumps, the amount of exhaust steam can be accurately limited to the amount that the feed water heaters and the heaters of the heating system can condense without excessive back pressure. We can also easily adjust the amount of steam going to the feedwater heater or the heaters of the heating system. These heaters were also placed high up in the plant and drained by gravity to an open heater to which make-up water was added and the feed pumps located lower than this open heater, took their suction from it under a head. Another reason we did not put the heater on top of the boilers and limit their use to live steam, was because we found during zero weather our water power was so irregular that calls on the steam plant were made to its full capacity; consequently, at such a time our auxiliaries are running full blast and have plenty of exhaust steam which can be utilized in the heating system and the water condensation easily pumped back to the boilers through the arrangement of an open heater as mentioned before.

In regard to the question by Mr. Ernst, the control of the heating in this mill is not automatic, but is done by the engineer in the power plant using a chart showing the temperature of water to be carried in relation to outside temperature. In times of very mild weather, additional regulation can be secured by running the circulating pumps at reduced speed. We have found this method amply able to take care of the varying requirements for heat at different times during the day in the different rooms of a textile mill without the expense and complication involved in automatic control of either water temperature or flow. The humidity control is left in the hands of the foreman of the different departments and is non-automatic.

No. 1497

RELATIVE VALUE OF PRIVATE AND PURCHASED ELECTRIC POWER FOR TEXTILE MILLS

By Frank W. Reynolds and Dan Adams, Boston, Mass.

Members of the Society

The rivalry between the isolated plant and the central station has called forth many valuable and interesting papers, usually prepared by those most vitally interested either in the isolated plant or the central station. The not infrequent result is that the two debaters approach the same facts from opposite points of view and draw opposite conclusions from them.

2 The central station advocate demonstrates that one kilowatt in the central station will do the work of several kilowatts in scattered small plants on account of the diversity in load; that the central station, besides requiring much less capacity, is also much cheaper per kilowatt than a number of small stations; and that its operating efficiency is higher, all of which is intended to prove that the small plant cannot compete with the central station in cost of power.

3 On the other hand, some mill engineers, manufacturers and other defenders of isolated plants show that the economic advantage of concentrating power in a large station is largely offset by the cost of distribution and high overhead expenses. Also, every mill must have a steam plant for heating and to supply steam for manufacturing needs. Many specific cases and figures are cited to prove that when these factors are properly considered, manufactured power is cheaper than purchased power.

4 Both sides are presented with equal sincerity and plausibility, and the inquiring mill manager is left without solid foundation of fact upon which to base a decision. It is perhaps safe to state that

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Textiles.

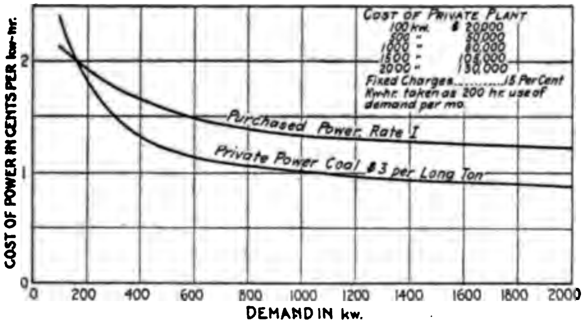
it is difficult to prove the general proposition entirely in favor of one side or the other. Each case is a problem to be decided on its own merits, and the correct solution can be assured only by a correct evaluation of all the various technical and financial factors involved. This paper purports to give an unbiased statement of the factors which affect the relative merits of private and purchased power for the textile industry.

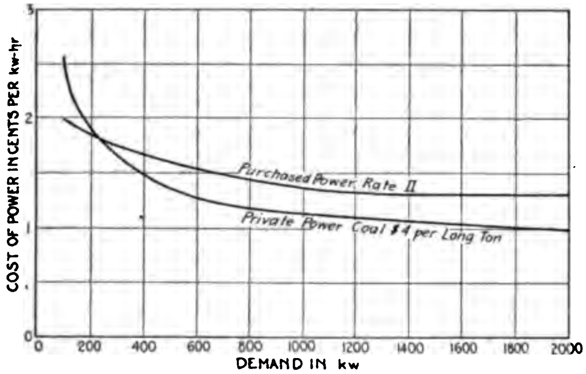
RELATIVE COST OF PRIVATE AND PURCHASED POWER

5 Rather than enter upon a theoretical discussion to determine what purchased power ought to cost, it will give a truer understanding of the matter to consider only what it actually does cost. Figs. 1 and 2 give the cost of purchased power for ordinary textile mill conditions for any load up to 2000 kw. These curves are plotted from the published schedules of public service corporations, one in New England and one in the Middle West. While rates as good as these cannot be obtained in all localities, still these are actual and representative rates. For comparison, the cost of private power has been plotted, using in each case the price of coal obtainable in the district served. The cost is for power delivered at the point of distribution at 550 volts in all cases. No provision is made for use of steam other than for power, and where such use is considerable, the relative cost of private power may be much lower than given. This factor is important and will be treated separately. While the cost of purchased power given is actual, the cost of private power is theoretical, and is not intended to give more than an approximate and general comparison, but is based on known actual conditions.

6 In an actual private plant the power cost very often exceeds that given. The investment as stated is too low to cover any reserve or relay capacity, and therefore the reliability must be considered somewhat inferior to that of those central stations which were used for this comparison. Also, the investment will not take care of any disadvantageous conditions, such as expensive foundations or difficulty in obtaining condensing water; and no cost items are included for land or for coal in storage. All of these items are too indefinite to be included in a general cost estimate, but are always encountered to some extent. The operating costs assumed, while no better than should be obtained, are really somewhat lower than the average. Nothing has been included for supervision other than the actual power plant labor, but this supervision is usually a legitimate item of expense.

7 To show the effect of some of these items on the cost of power, an actual case will be cited. An industrial plant (not a textile mill) in Canada had purchased power for four years, and before the termination of the contract considered carefully the feasibility of generating its own power. The maximum demand was 1500 kw. The proposed plant was to contain two 1000-kw. turbines. Reliability was of great importance, and the concentration of all generating capacity in one





FIGS. 1 AND 2 RELATIVE COST OF POWER FOR LOADS UP TO 2000 KW.

unit was not considered satisfactory. Purchased power, being supplied underground by duplicate feeders from a substation fed by several steam and hydroelectric stations, had proven very reliable. There were few interruptions in four years' use, the longest being of three hours' duration. A large amount of high pressure steam was needed in manufacture and there was enough capacity in the fairly new boiler house so that only one additional 350-h.p. boiler would have been required for power generation. On the other hand, foundations were rather expensive, and the condenser supply would have cost about $12,000, or $8 per kw. Most of the equipment would be subject to import duty and war tax of 35 per cent total. It was found that the plant would cost $80 per kw. of installed capacity, or about $107 per kw. of demand, and power cost would be $0.0123 per kw-hr. with coal at $3.50. As power was being purchased for $0.0104 per kw-hr., it was decided not to generate.

8 If we eliminate duty and tax from this, plant cost would be $93 per kw. of demand, as against $70 assumed in our curves, and power with coal at $4.00 would cost $0.0119. This has to be compared with purchased power at $0.0130 per kw-hr. The saving of private over purchased power would then amount to $4000 per year or about 3 per cent of the cost of the plant.

9 The average yearly saving of the private plant, 500 kw. in size and over, from the above curves, is about 9 per cent of the cost of the plant. The highest value is 12 per cent for the 2000-kw. demand. Therefore, the private plant under ideal conditions can compete favorably with the power rates given, but the margin is not large and must cover several indeterminate cost items not included.

10 The central station rates quoted apply to any industrial plant. A textile mill load is properly considered somewhat more favorable than the average industrial load on account of its very steady demand for power, not only throughout the working period of each day, but also throughout the year and over a period of years. For this reason, special power rates can be obtained by textile mills in some localities, notably in the South, and to some extent in New England. Under these special rates, power usually costs about 1⅛ cents per kw-hr. for demands from 1000 to 2000 kw. and about 1 cent above 2000 kw. Such rates leave little financial inducement for the mill to build its own power plant.

EFFECT OF SIZE OF PLANT ON RELATIVE COST

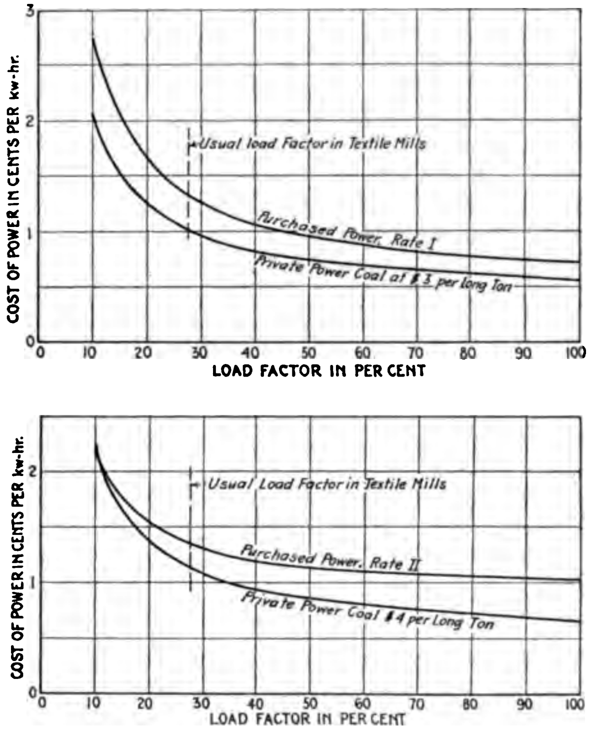
11 It will be seen that the curves for purchased and priva[te] power, Figs. 1 and 2, are very nearly parallel. This means that th[e] size of the demand has little effect on the relative cost. The larg[er] plants show a slightly larger proportional saving than the small[er] ones, and the very small plants—under 200 kw.—cannot compete wi[th] the purchased power rates given.

EFFECT OF LOAD FACTOR ON RELATIVE COST

12 Figs. 3 and 4 have been prepared from the same data as Fi[gs.] *1 and 2,* to show the effect of load factor on the cost of power in t[he]

case of a 1000-kw. demand. It will be seen that the curves for private and purchased power costs follow the same general direction. It appears, then, that load factor also has little effect on the relative cost of power.

13 There is a striking similarity between the curves shown for private and purchased power, both when the variable is the size of the demand and when it is the load factor. The inference is that



FIGS. 3 AND 4 RELATIVE COST OF POWER FOR LOAD FACTORS 10 TO 100 PER CENT

these power rates were devised to compete with isolated plants rather than from actual costs, or desired rates, of the central station power. This is entirely logical, being simply an application of the law of supply and demand. If it be true that the cost of central station power is determined by the cost of generating in isolated plants, then it will be unnecessary to discuss further the relative cost. It is improbable that the central station will sell cheaper than the average isolated plant cost, and if its rates are very much higher, it cannot sell much power.

14 The above, of course, applies only in the case of new plants, where power can be purchased without any sacrifice of existing investment, and where steam is needed only for power. Also, it must be remembered that the individual plant may depart widely from the average.

EFFECT OF FIXED CHARGES ON RELATIVE COST

15 All that has preceded is a comparison of purchased and private power where the private plant is not yet built. But, in the case of a going plant, the purchase of power would save operating expense only, and would not wipe out fixed charges on investment already made. Fig. 5 shows the cost of private and purchased power, as in Fig. 1, but without including fixed charges. This, then, is a comparison between purchased power and a going plant under good working conditions. The difference in cost is so great that it appears improbable that central stations can cause existing plants to shut down unless in the case of important changes or additions, or the necessary renewal of considerable apparatus, or because of extremely poor operating economy.

EFFECT OF USE OF STEAM ON COST OF POWER

16 Every textile mill uses some steam for heating and manufacturing. This factor is of great importance, and should always be weighed accurately in considering purchased power.

17 In general, textile mills may be divided into two classes in this respect. The first contains those mills using only a relatively small amount of steam in manufacturing, and includes most silk and knitting mills and cotton and woolen mills not engaged in scouring, bleaching or finishing. Dye houses, finishing mills, print works, and textile mills which finish their product are included in the second class, and have a large but variable demand for steam in the process.

18 Slashing may be taken as typical of a small demand for manufacturing steam. Very often exhaust steam is used in this process. The pressure desired varies from 5 to 12 lb., which is a suitable pressure to bleed from an engine receiver or bleeder turbine. Only in rare cases would it be economical to run non-condensing apparatus to supply slashers on account of the high back pressure required and the relatively small demand. In the case of a bleeder turbine, the saving in live steam amounts to about 30 to 40 per cent of the steam

bled. This saving is reduced by additional fixed charges on extra cost of turbine and exhaust piping, for a relatively small service, and, in a large plant with a long run of pipe, may be negative. For a rough figure, applicable to average conditions, the gross saving will be about $200 per year for each 1000 lb. per hour bled, with coal at $4.00 per ton. This figured out about $0.0001 per kw-hr. in the case of a large cotton mill, or about 1/50 of 1 per cent of the cost of production. While there is no doubt that exhaust steam can be and is successfully used in slashers and similar machines, still live steam at reduced pressure has a tendency to reduce operating difficulties and increase production. When exhaust is used, there are apt to be occasional periods when results are not satisfactory, due to a temporary drop in pressure or excessive condensation or to adverse atmospheric

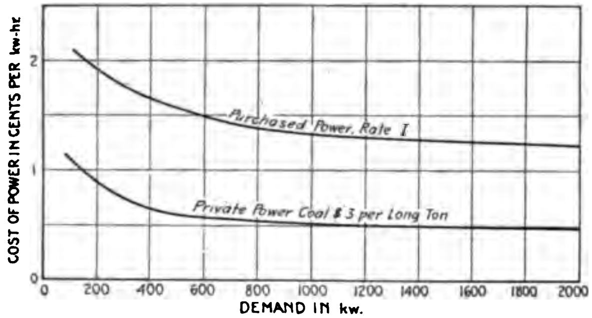


Fig. 5 Relative Cost of Power, without Including Fixed Charges

conditions. If live steam is used, it is fed to the slasher in a dry or even superheated condition and this, together with the perfect flexibility in control of pressure, obviates most of these difficulties. Where a saving is relatively so minute, the wisdom of obtaining it at the cost of even a small operating handicap is fairly questionable.

19 As regards steam for heating, there are two factors, often ignored, which militate against the use of exhaust in textile mills, especially in cotton and woolen mills. One of these factors is the large amount of heat liberated from the machinery during working hours and the second is the diversity in time between the use of power and the use of steam for heating.

20 Table 1 shows the amount of heat required to warm each room in a large cotton mill in zero weather and also in 35-deg. weather, which may be taken as an average during the heating season, and also

the heat liberated from machinery in each room. It is assumed that all the power used in cotton machinery is dissipated as heat.

21 The mill chosen for this example is in northern Massachusetts and has a very large window area and a saw-tooth weave shed. The heating requirements are certainly much more severe than the average. Yet, even in this case, the machinery provides more than half of all the heat required in zero weather, and for much of the heating season no extra heat whatever is needed during working hours, except in the storehouse and basements. In a mill of this kind, the heat is usually turned on at 4 or 5 A. M. to get the mill warmed up before starting

TABLE 1 COTTON MILL HEATING

Building	Contents	H.P.	Heat Radiated from Building, (Calculated) B.t.u. per Hour		Heat Supplied by Power, B.t.u. per Hour*	Heat to be Supplied by Steam in Mill Hours. Zero Weather
			0 Deg.	35 Deg.		
Weave shed......	Looms	1628	8,550,000	4,275,000	4,140,000	4,410,000
Weave shed basement..........	Storage	....	980,000	490,000	....	980,000
Mill, basement...	Storage	....	1,550,000	775,000	....	1,550,000
Mill, 1st floor....	Picking Carding Roving	660	1,900,000	950,000	1,680,000	220,000
Mill 2nd floor....	Picking Carding Roving	647	1,900,000	950,000	1,650,000	250,000
Mill 3rd floor.....	Filling Spinning	691	1,900,000	950,000	1,760,000	140,000
Mill 4th floor.....	Warp Spinning Spooling Warping	1378	2,900,000	1,450,000	3,500,000	....
Storehouse.......	Storage	....	1,800,000	900,000	....	1,800,000
Total...........			21,480,000	10,740,000	12,730,000*	9,350,000

*Assuming all power transformed to heat.

time, and is turned off sometimes as early as 9 A. M., and sometimes not at all, depending on the weather. In a weaving mill in northern Massachusetts, where the power used per cubic foot of space is much less than in a balanced mill, the heating system was actually in operation only 25 hours per week average during working hours during the heating season, or about 1/4 of the entire yearly operating time of the mill. Of course, all heat used outside of working hours—nights and week ends—must be live steam. If exhaust is to be used the rest of the time, as a general rule, it is proper to count on a demand only about one-half of the maximum, since the machinery provides the remainder, and this quantity probably will not be used more than 700 hours per year. The only way to meet a demand for exhaust of this character with economy is by bleeding from an engine or turbine. In a cotton mill cited above, bleeding steam for heating would effect a saving amounting to $0.00005 per kw-hr., a little less than in the case of slashers. In knitting and silk mills, the saving will be relatively greater but in warmer climates much less.

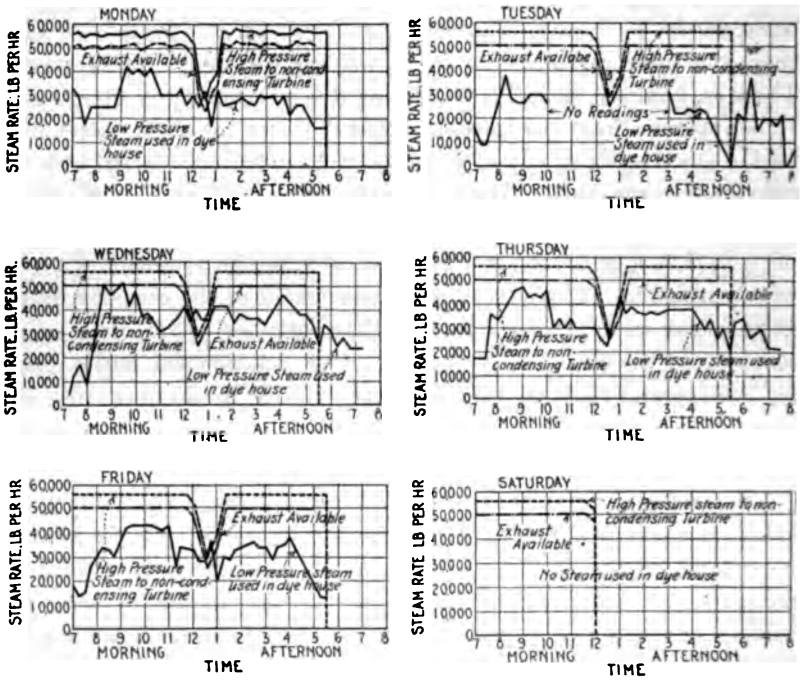
22 It appears, then, that the coal saving effected by using exhaust steam for heating and slashing is very small—smaller, in fact, than the unavoidable errors in estimating the total cost of power. There is, however, some further saving due to economy in investment. It is apparent from the foregoing that if steam for these demands is bled, the extra boiler capacity necessary to meet them is reduced approximately one-half. Also, it is cheaper to get this extra boiler capacity as part of a larger plant built for power generation than to provide it in a separate plant by itself as would be necessary if power were purchased. This matter was investigated in the case of a fairly large cotton mill. It was found that the complete power plant would cost $65.00 per kw., but if the cost of an independent steam plant for heating and slashing were deducted, the cost would have come out about $58 per kw., this latter being the figure used in comparison with purchased power.

23 While it is entirely unsafe to apply average figures to individual cases, it is desirable to form a clear conception of the relative importance of the various factors, and we may say that, in general, the entire fuel and investment saving due to the bleeding of steam for heating and slashing will range from $0.0005 to $0.0010 per kw-hr. A saving of this size cannot have a very important influence on a decision for or against purchased power. Nevertheless, the matter should always be considered.

24 A very different result is obtained in those mills having a large demand for steam for boiling water and drying cloth. In such cases, a large proportion of the prime mover exhaust may be utilized. In some cases, all power is produced in non-condensing apparatus, and all exhaust utilized, and manufactured power becomes then so cheap that no central station can compete. It is seldom, however, that a textile plant can utilize all of its exhaust all the time. This is because of diversity in time and amount between the use of steam and

use of power. The importance of this can best be shown by citing an actual case.

25 Figs. 6 to 11 give curves showing the demand for steam in a worsted mill dye house for one week's operations. These curves are plotted from actual flow meter readings in exhaust main, which were taken every fifteen minutes. In this case the exhaust was provided by a non-condensing turbine. For comparison, the steam fed to the turbine was measured on one day and plotted. For the other days



FIGS. 6 TO 11 ONE WEEK'S STEAM DEMAND IN WORSTED MILL DYE HOUSE

the steam to turbine was averaged from the kilowatt-hour readings. The curves show how the demand for steam varies from hour to hour and day to day, and also lags behind the power load. The demand for steam in a dye house is light early in the forenoon, but persists for two hours or more after the mill is shut down. The dye house does not run at all on Saturday mornings, which is a custom not uncommon. The result is a very large surplus of exhaust which must be wasted, and in spite of this, considerable live steam make-up is required, as shown by the curves. It is obvious that such conditions result in great inefficiency. Actually less coal would have been burned if a standard condensing turbine had been installed and live steam exclusively used in the dye house. The expected result of this installation was that power would be furnished as a by-product at a purely nominal cost. The actual result was that the power cost was considerably higher than in a plant having no use for steam in the process. This case is especially worthy of study because it is a typical balanced worsted mill of fairly large size doing its own dyeing; there were no special or unusual features involved; the installation was actually made, and the results as given were determined by careful tests.

26 The reasons and calculations leading to the installation of this non-condensing turbine are not known to the authors, but there is no reason to doubt that the installation promised to be very profitable. The total steam used in the dye house was considerably in excess of that to be supplied by the turbine, and it was apparently safe to assume that all exhaust would be utilized. With the turbine, there would be no trouble from oil, and the exhaust pressure was taken at 10 lb. to give good service.

27 The factors which made this attempt a failure were

1 The diversity in time and variable demand for steam
2 The deterioration of turbine which gradually increased its steam consumption
3 It was found by experience that a few of the processes in which the use of exhaust steam was tried suffered either in quality or speed of production. For this reason the actual demand for exhaust probably was not as great as that estimated, although this difference was not very large.

It is evident that all of these factors may be encountered in any similar installation and must be allowed for in the estimates.

28 It is interesting to inquire if it would have been possible to make this installation efficient and still use exhaust steam. The demand is so variable that it is hardly probable that a non-condensing turbine could be satisfactory. It would be impossible to avoid surplus exhaust at times and live steam make up at others. A bleeder turbine appears to meet the situation better, but is not without its difficulties. The required output of the turbine is about 1200 kw. The maximum amount that can be bled at that load from a 1200-kw. machine is about 12,000 lb. per hr., but the greatest demand for exhaust steam in the dye house is 50,000 lb. per hr. If the turbine has different

characteristics, involving practically a 2400-kw. steam end, all the exhaust needed could be bled, but the turbine would run at half load economy. It would be better to use an intermediate size and bleed what steam is possible, supplying the dye house peaks with live steam make-up.

29 Table 2 gives the total steam used per week during the operating time of the mill for turbine and dye house for the operation as actually observed; as it would have been if guaranteed steam rate of turbine had been maintained; if a condensing turbine had been installed; and with a 1600-kw. bleeder turbine.

TABLE 2 STEAM USED FOR POWER AND MANUFACTURING

	High Pressure Steam to Turbine Lb. per Wk.*	High Pressure Steam Make-up in Dye Ho. Lb. per Wk.	Total High Pressure Steam Lb. per Wk.
Present operation, 1200 kw. non-cond. turbine	3,110,000	....	3,110,000
Same with guaranteed steam rate of turbine maintained	2,466,000	47,000	2,513,000
With 1200 kw. cond. turbine	1,120,000	1,580,000	2,700,000
With 1600 kw. bleeder turbine	2,294,000	273,000	2,567,000

*Includes turbine auxiliaries. Condenser driven by motor.

30 These figures show that the condensing turbine is cheaper to operate than the scheme installed, and is not much more expensive than any of the schemes. It is safe to state that the condensing turbine would have been the lowest of all in first cost and the most satisfactory from the standpoint of dye house operation.

31 In this particular case, then, a large demand for steam in the process has little influence on the relative cost of private and purchased power. If power generation in a condensing turbine with live steam used in the process is as economical as the use of exhaust steam, the comparison previously made between purchased and private power where there is no demand for steam is approximately true in this case. There would be some investment saving in providing steam capacity for manufacturing as a part of a larger plant.

32 A careful study of this case is recommended to anyone desiring to supply a large and variable demand for exhaust from a prime mover having a constant power output.

33 Another interesting case is that of a large print works where the demand for hot water and steam was very large in proportion to the power. In this case, condensing turbines were installed and for

the greater part of the time all the condensing water is utilized in the process. This is a closed heat cycle and therefore as economical as to use exhaust, but has the advantage of economical turbine operation irrespective of the demand for waste heat. This example has little practical value in a general discussion because it is so seldom that the demand for hot water is sufficient to absorb all the condenser discharge.

34 It is hardly possible to give general figures of any value to show the effect of these large steam demands on the cost of power. In many cases, the factor is of less importance than has commonly been supposed. Nevertheless, it seems entirely improbable that purchased power can compete successfully in very many cases of this kind. The success of the private installation, however, depends very largely on the skill and thoroughness of the preliminary study of conditions, with special attention paid to the diversity factor and variations in demand.

WATER POWER

35 The proportion of textile mill power supplied by water is constantly becoming smaller. This does not necessarily mean that existing water powers are being abandoned, but very few new mills are now built at water power sites. Where a mill has water privileges of any considerable value, purchased power can hardly displace water power. Even in such plants, however, power is sometimes purchased to relay the water wheels, or for demands in excess of the water power capacity.

RELATIVE RELIABILITY OF PRIVATE AND PURCHASED POWER

36 In textile work, the reliability of power is usually of greater importance than its cost. This is because the cost of power is a very small item in the cost of manufacturing, whereas reliability affects the earning power of the whole mill at its foundation. It is rather difficult to collect satisfying statistics on power reliability by each method of supply. One cotton mill buying power reports interruptions totaling 45 min. in five years. In this case the transmission line was less than half a mile long. Reports from several central stations give total interruptions on the individual lines ranging from 12 min. to 4 hr. per year.

37 A hydroelectric company in the South, which sells to many mills, reports that interruptions of service to textile mills from all causes have averaged less than 2/5 of 1 per cent for the last nine

years. Such records are doubtless better than the average. A large number of managers of representative mills in the South purchasing hydroelectric power state that they consider this more reliable than private steam power.

38 In most cases, purchased power is transmitted electrically by pole lines. Where these lines are short, well constructed and in duplicate, the reliability is very good. On the other hand, a single line many miles in length and fed at a long distance from the consumer is subject to interruptions and regulation troubles. This has been repeatedly proved. It is fair to state, however, that great improvements have been made in the reliability of transmission.

39 The reliability of generation is usually better in a large central station than in an isolated plant, and this is especially true when a number of stations are tied to the same system.

40 An interruption of purchased power is more annoying than a shut down in a private plant, because the management does not immediately know the cause or how long the power will be off. For this reason, purchased power is sometimes condemned as unreliable when the total interrupted time per year is small. An isolated plant with considerable relay capacity must, as a rule, be considered more reliable than purchased power. If the plant is small, however, the investment charges on this relay capacity add materially to the power cost, and few isolated plants carry spare apparatus.

41 In considering the purchase of power, reliability should be treated as a very important factor, and carefully investigated. If there is any serious question on this point, the private plant would be the wise decision in most cases.

GENERAL DESIRABILITY OF PURCHASED POWER

42 Assuming that reliable power can be purchased at a cost approximating that of private power, it is not apparent that there are any attending disadvantages. There are, rather, many points in its favor.

43 The saving of investment for a power plant often is considered important, especially when that money can be invested to better advantage in the manufacturing plant. An example will make this point clear.

44 A cotton mill of 100,000 spindles represents an investment of about $2,000,000, and will earn from $2.00 to $4.00 per spindle per year. If it earns $3.00 per spindle, or $300,000 per year, this is

15 per cent on the investment. Power needed would be about 4000 kw. The power plant might cost about $300,000, and power $0.0085 per kw-hr. Purchased power would ordinarily cost about $0.01 per kw-hr. Under these conditions, private power would save in money $14,400 per year. Total earnings are then $314,400 on an investment of $2,300,000, or 13.7 per cent. This is a poorer showing than with purchased power, in spite of considerable saving in power cost.

45 Specialization in manufacture of product only is worth something. A considerable part of the effort of the engineering department of a mill is devoted to keeping the power plant running efficiently. Where cost of power is only 2 or 3 per cent of the cost of manufacturing, the services of the engineering department would probably be of greater value to the mill if devoted solely to keeping the producing machines running efficiently.

46 In the case of a new development subject to future growth, the central station offers a perfectly flexible source of power.

47 When a radical change in existing conditions is introduced it is inevitable that there will be some wrong applications. Whenever the purchase of power proves undesirable, or too expensive, it is easy to give it up. The fact that so very few plants have gone back to private power after once purchasing is pretty good evidence that purchased power really is desirable.

POWER CONTRACTS

48 Where schedules for purchased power are published and intended to cover all conditions of use in industrial plants, they are necessarily complicated, and some are unnecessarily complicated. Cases have been known where a manager has refused to buy power for the sole reason that to do so required him to sign a contract he could not understand. Conditions governing the use of power in textile mills are such that power companies often offer them special rates at a flat price per horsepower-year or per kilowatt-hour, with a guaranteed minimum. Such a contract would be received more favorably by the average mill man than one based on demand and monthly use of demand with various discounts, even though the final results were identical.

49 Contracts for purchased power should be carefully studied in all details before signing, as there are many technical features which are sometimes made to react to the disadvantage of the consumer. In general, power companies show a tendency to get the consumer's

point of view and to make their contracts more simple and more liberal.

AMOUNT OF POWER PURCHASED IN THE TEXTILE INDUSTRY

50 The figures in Table 3 from the 1910 United States Census show the total and relative amounts of power purchased in the various textile industries, and also the total figures for all industries, for the year 1909.

TABLE 3 POWER PURCHASED FOR TEXTILE MILLS

From U. S. Census Report for 1910

Industry	Total h.p. Used	Total h.p. Purchased	Per Cent Purchased
Cotton goods, including small cotton wares	1,296,517	108,512	8.4
Hosiery and knit goods	103,709	13,286	12.8
Silk and silk goods including throwsters	97,947	10,354	10.6
Woolen, worsted and felt goods and wool hats	362,209	13,783	3.8
Total	1,860,382	145,935	7.8 Average
All industries	18,675,376	1,749,031	9.4

51 It will be seen that the hosiery and silk industries purchase relatively more power than the average textile mill, and this might be attributed to the comparatively small demand for power in these mills. The small use in woolen mills is probably due to the generally large use of steam in this industry. Since these figures were compiled, the capacity of central stations has more than doubled. It will be interesting to observe from later census reports if the use of purchased power in the textile industry has increased proportionately.

DISCUSSION

FRED N. BUSHNELL (written). One notable feature of the paper stands out prominently, and that is the absence of any attempt to show the relative value between mechanical and electrical drives, which indicates that material progress has been made in the trend of thought and state of mind with which engineers, closely in touch with textile problems, now view the application of the electric motor to this class of work.

One of the greatest difficulties encountered in any attempt to show comparative costs, and one that nearly always places purchased power at a disadvantage, is the incomplete knowledge of all the facts entering into the cost of private plant manufacture, and the necessity of assuming economies which may or may not be realized in practice.

The authors frankly point out that in an actual private plant the power cost very often exceeds that given; that the investment as stated is too low to cover any reserve or relay capacity; that the investment will not take care of any disadvantageous conditions, such as expensive foundations or difficulty in obtaining condensing water, also that no cost items are included for land or for coal in storage, all of which must be taken into consideration in studying the curves.

On the other hand a price named for purchased power is a definite, unequivocal and positively known quantity about which there is and can be no question, and this fact should also be borne in mind in any attempt at comparison.

We have now reached a period of highly developed specialized knowledge in matters immediately affecting our principal activities, and a tendency has very naturally developed to eliminate all problems of minor importance, not directly connected with our principal article of manufacture.

The shoe manufacturer is seldom a tanner of the leather or a weaver of the cloth he uses, the toolmaker does not mine or produce the steel he needs, nor the miller the wheat he grinds, and the day is approaching when the textile mill will find no more reason for manufacturing its own power than it will for the study of the intricacies and uncertainties of agriculture in order to supply its raw cotton.

The authors are to be congratulated upon the evident fairness with which they discuss the subject.

John A. Stevens (written). Although this paper purports to be an unbiased comparison of isolated and central station electric power, it conveys the impression that the latter is preferable.

As the authors state, each case is a problem to be decided on its own merits, and the correct solution can be assured only by a correct evaluation of all the various technical and financial factors involved. Each is, in other words, strictly a technical and financial problem which requires careful study indicating economies brought about by a certain investment, of which the savings affected by the increased economies represent a certain earning. Sometimes the balance is in favor of isolated power and sometimes of central station power.

A proper comparison of the two systems of power must include the following items, which are very often slighted, misrepresented or omitted: To the price of purchased power must be added the fixed costs of all transforming equipment, including housing, transformation losses and cost of attendance; and if an existing plant is superseded, its fixed charges must also be included at full value.

The cost of isolated power should include every item associated with the power plant, that is light, heat and power and all thereto connected, due reduction being made for use of steam for heating and manufacturing purposes. The fixed charges on the investment, 12 per cent of its initial value being sufficient, should include heavy foundation waterways and coal handling apparatus along with the other plant equipment.

It should be brought out here that careful design of the private industrial plant is of prime importance if central station power is to be competed with. In numerous cases of industrial plants having been pronounced failures, more attention was paid toward making the plant attractive than economical.

The question of reliability has often been brought up. A prime mover is as reliable in an isolated plant as it is in a central station, and if the isolated plant possesses more than one unit, reliability is not a factor. The prime movers in a well run plant are the most reliable parts of the plant. The weakest part of a central station system is its distribution, and from this source the prospective purchaser of power will experience the most inconvenience.

The authors' examples of power costs clearly represent special cases in which, with study from different aspects, the favor might be swung back to isolated service. In the case of the dye house, for example, numerous factors are not considered. Low pressure steam requires piping much larger than does high pressure. A combination of a condensing and exhaust steam turbine will greatly help out the diversity of steam and power demand, and at the same time eliminate some of the losses in bleeder turbines.

One of the most important features not mentioned is an absolutely controllable supply of low pressure or stage exhaust steam at some predetermined pressure. That is to say, whether or not a low pressure system can be installed in a plant depends on the amount of low pressure steam to be used, the amount of power to be used and the cost of installation of the low pressure system, including the additional rates imposed on the steam plant by the low pressure system. A suggested method of approach is a complete analysis of light, heat and power in its every minute detail, including the land occupied by the equipment as against the purchased power, where practically no space would be absorbed by an isolated plant. Further, it is to be specifically recommended that in every case power be purchased in the form of energy, that is, on the basis of kilowatt-hours used.

R. J. S. Pigott. The authors conclude that for small power plants there is very little chance for the isolated plant to make good. In the small plant, the duties for the men operating it are small and cannot occupy their whole time. A low grade man is very frequently employed, and only works part of his time in the power plant, causing the reliability to suffer.

For a fair comparison between the isolated plant and the central station another point which must be known is whether or not proper reserve is carried in the former. The authors state that in few industrial plants is a proper reserve carried, and this has been my own experience. Enough engine or turbine power is put in to run the plant when all the machines are operating, but no spare units are carried. Such a plant is not in a position to be compared fairly with purchased power, because it is not as reliable as a central power station. If the central station carries a 20 or 25 per cent reserve, the isolated plant ought to carry that much, if the reliability is to be the same.

In the cotton industry there is stated to be very little chance for the use of exhaust steam, which seems extraordinary. It is possible that the temperature demand is such that live steam direct to the machines must be used, but it would appear well for mill owners to remodel their processes so as to use low pressure steam wherever possible in place of high pressure steam.

In the works where I am now engaged, nearly half the steam put through the turbines is employed either in the processes or for heating. The power plant has a capacity of 13,000 kw. In this particular plant, low pressure steam, 15 lb. gage, drawn from bleeder turbines, is supplied. Whatever steam is not required for processes or heating in the main plant is passed on to the condensers, with the consequent benefit of comparatively high economy. The steam consumption would be very nearly doubled if we were to use high pressure steam drawn from the boilers, and condense all the steam sent through the turbines.

We have remodelled a number of our processes in order to make use of low pressure steam; the effect upon kw-hr. cost is very pronounced, as the power from bleeder turbines is produced at 75 to 90 per cent thermal efficiency. The case is analogous to that of steam auxiliaries exhausting into a feed water heater.

From an inspection of most of the industrial plants which make use of steam in processes, it would appear that not sufficient attention is given to the ability of the bleeder turbine to furnish steam at a very low cost; and for those who are interested in the question of private or purchased power for industrial works, it would be very advisable to give serious consideration to the remodelling of their processes and heating conditions, to make as much use as possible of bleeder turbines. The growth of sales of the bleeder type of turbine in the last two or three years indicates that the advantages of low pressure steam for industrial purposes are now beginning to be realized.

F. J. BRYANT. This problem of exhaust steam came up when we enlarged the power plant of our cotton finishing works, where a large portion of the steam generated is used for drying. We considered the bleeder turbine very carefully and decided that it did not meet our needs.

We have at present a number of 5 to 10 h. p. "Twin-angle steam engines," which drive drying machines and serve a double purpose of speed regulator and reducing valve. The part of the machine which these engines control consists of a set of drying cylinders over which the cloth passes after it has been partly dried over steam coils. The steam in these cylinders, and the amount of steam in them, depends upon the weight of the goods which they are drying and the amount of moisture in them. As the engines discharge their exhaust into a header which supplies the cylinders of several machines, very good economy is secured. If the pressure falls too low more is admitted by a reducing valve, and if it gets too high a safety valve lets it off. A turn of the throttle valve speeds up the work and lets more steam into the system. All drips and condensations are trapped to a hot well and then returned back to the boiler.

As we now have a surplus of low pressure steam, we are considering the substitution of an alternate current motor and variable speed transmission for one of the engines, and thus cut down the exhaust.

Arthur L. Williston. In this paper the author has assumed that there are three objections to the isolated plant: first, the lack of opportunity to use the exhaust steam; second, lack of margin in the plant; and third, lack of reliability. These are serious shortcomings for any plant to have, but I think it is important that we should not associate them with any particular type of plant. I am sure that we all know central stations that have at least one of these defects; and in some instances all three may be present. It is also true that isolated plants may be so designed and operated as to have none of them. The point that seems to me to need special emphasis is the fact that the same high quality of skill and judgment that is usually bestowed upon the design of the central station is, as a rule, equally important for the isolated plant.

For example, there is not the slightest reason why the isolated plant should not have as large a margin as is needed. It surely is not necessary to go to a central station in order to get a wide margin. It may be obtained in any plant.

Likewise, it is not necessary to go to the central station to get reliability. In the isolated plants with which I have had experience for the past twenty-three years (which happen to have been plants in educational institutions) we have had as great a degree of reliability as we could reasonably expect to have from central stations. In the last plant, since it was started five years ago, we have had practically perfect reliability. A central station does not always give an absolute 100 per cent of reliability.

During the previous discussion it has been pointed out that in an isolated plant processes may often be remodeled so as to use exhaust steam in place of high pressure steam, and that when this is done there is distinct economy in favor of the isolated plant as compared with the price of power furnished from a central station. In educational plants there is little opportunity to use steam for special processes, but they do use large quantities of exhaust steam for heating.

Our shop and laboratory buildings at Wentworth Institute contain equipment not very different from that found in a great many manufacturing plants, and are, for illustration, in almost every way quite typical of the small plants in a great many industries. It is our experience that the exhaust steam used for heating alone consumes all, or very nearly all, the exhaust of our power plant, for six months in the year. There is during the other six months a certain amount of waste, but for one-half of the year the only cost of light and power is a small depreciation charge for the engine dynamo and the renewal of the lamps. A very distinct economy will usually be found in favor of the isolated plant wherever conditions approaching these can be found.

In a great many instances wrong conclusions have, I think, been drawn when making comparisons between isolated plants and central stations because persons have relied on data drawn from improperly designed isolated plants rather than from the results that would be obtained from well-designed isolated plants with thoroughly up-to-date equipment and with all opportunities for economy taken advantage of.

Charles H. Bigelow said that in connection with the problem of isolated plants, he had had some experience in using exhaust steam at a textile plant where there was a large demand for low pressure steam. A 500-kw. non-condensing turbine was first installed, and the exhaust steam used from that at about 8 lb. pressure. The demands for the exhaust steam were very variable, as well as the loads on the turbine; part of the time steam would be escaping through the exhaust pipe into the atmosphere, and at others the required amount would have to be made up through a reducing valve from the high pressure line.

They have since put in a 1000-kw. bleeding turbine, the load having increased, and installed a recording steam flow meter on the supply pipe to the factory. The load on the turbine was the factory load with a four or five-car traction load superimposed, the latter varying from nothing to 300 kw. almost instantaneously.

It was found from the charts that the demands for power as well as low pressure steam were very variable, but the bleeding turbine holds the pressure steady, at 10 lb., varying one pound each side of it as shown on the recording chart. The broad line is as straight as it can be drawn, showing that the regulating mechanism for holding the pressure steady in the first stage of the turbine is doing its part of the work, although it takes about two pounds variation in pressure to make it work.

Incidentally there is an indicating steam flow meter on the supply pipe to the turbine, and this swings from perhaps 15,000 to 30,000 lb. per hour, back and forth, depending on the load and the demands for steam. The bleeding turbine seems in this case to be solving the problem of supplying a variable amount of low pressure steam, without loss to the atmosphere as was formerly the case. The balance of the steam passes to the surface condenser through which the circulating water flows by gravity from a pond owned by the company and under winter conditions operates at over 29 in. vacuum. The condensate flows over a V-notch which incidentally gives a check on the two other measurements of input and the balance of output.

In regard to processes for using low pressure steam, there is a good deal of tradition in the pressure required for manufacturing processes. What is generally really required is temperature, and he wondered whether anything has ever been done to superheat low pressure steam when used in manufacturing processes.

WALTER N. POLAKOV. In this very interesting paper, the figures and curves are based on averages of actual performance, but the question is whether what has been done in the past is of any value to judge as to what should have been done.

It has been pointed out in the discussion that in mill plants the power plant employes are usually of not very high grade. The owners try to hire the cheapest kind of help they can get, and in my experience I have found that on an average between 30 and 40 per cent of the cost of power generation can usually be saved merely by a proper method of operation, as it is not very uncommon to see firemen in a mill wasting tons of coal each day.

Comparing cost curves of private and purchased power, and assuming that large public utility power plants are also not as efficient as they might be, the rate for purchased power will probably be reduced in the future by 20 per cent on the average; whereas, private power cost will also probably go down some 50 per cent. In other words, the difference between the cost of private and purchased power, as shown in Figs. 1 and 2, will be even more pronounced in the future than the authors pointed out from actual experience of the past.

Dan Adams. This paper was intended to be a discussion of the relative value of private and purchased power, as applied to the textile industry only, and the conclusions are not applicable to any other industry, such as machine shops. Mr. Pigott has shown the great savings which can be made from bleeder turbines in this latter industry, which is very different from the textile industry.

The chief difference in regard to the heating has been brought out very clearly in Mr. Duncan's paper. Simply there is no demand, or I will say only a small demand, for heating, coincident with the demand for power; and that is due to the large amount of heat liberated from textile mill machinery, which I think is peculiar to that industry.

Prof. Williston mentioned the saving by heating from exhaust in an institution, but the same thing holds true there. Of course there is a very large saving from the use of exhaust steam for heating in any plant where the demands for power or light and heating coincide, such as office buildings or educational institutions.

It was pointed out by Mr. Bryant and Mr. Bigelow that there are difficulties in controlling the use of exhaust steam in a textile plant, in order to obtain economical results. Usually these problems can be solved, and it is a fact that most textile mills of this character use exhaust steam very generally. It was not intended in this paper to discount the savings which can be made from using exhaust steam when conditions are suitable, but it was intended to point out some of the difficulties; and we feel that some expensive mistakes have been made when it has been attempted to apply exhaust steam to these processes, without an absolutely clear and detailed analysis of all the conditions.

Mr. Bigelow also cites the case of the bleeder turbine, which is running very successfully, and there are many such cases. The bleeding of steam through a turbine is undoubtedly very economical when conditions are suitable.

In the case cited in the paper we could not seem to produce very economical results by using a bleeder turbine, and this was owing to the relation between the demand for exhaust steam and the demand for power. It was found that at times the demand for exhaust steam was more than could be bled from a standard bleeder turbine of the right capacity. For instance, the power output was 1200 kw. and the total demand for steam at times reached 50,000 lb. per hour. All that can be bled from a 1200-kw. turbine is about 12,000 lb. per hour, which was inadequate.

Now, if you use a large bleeder turbine for the sake of being able to supply exhaust steam, you will necessarily reduce the economy of the turbine, when it is not bleeding, because you are then running it at part load economy.

In this particular case, there was no use for exhaust steam whatever for something over 10 per cent of the time. The dye house was shut down on Saturday morning, which is quite common in worsted mill dye houses. The combination of all these factors resulted in a very small saving in this case, but it does not necessarily follow that it would be so in all cases.

Mr. Williston mentioned the fact that isolated plants can produce power as reliably as central stations, and this point was also brought out in the paper, that isolated power is absolutely reliable if the isolated plant has sufficient relay capacity. It should be remembered, however, that if small plants are provided with relay capacity, the investment charges run up very fast and the cost of power then approximates or exceeds the cost of purchased power.

Mr. Polakov brings out the point that the private power cost given in the paper is subject to material reductions. While it is true, as he states, that industrial power plants can on the average be improved very materially, this is less true of the textile industry than of most industries. In other words, I believe that the textile mills have devoted more attention to the economical generation of power, and this is perhaps due to the fact that they are very large consumers of power.

No. 1498

THE ENGINEER AND THE BUSINESS OF FIRE INSURANCE

By Joseph P. Gray,[1] Boston, Mass.
Non-Member

The large Factory Mutual Companies are today, almost without exception, managed by engineers who were educated and trained in their professions. In years past, the province of an engineer was simply to give advice and lay out work, but not to execute it. Some of the older, technically educated members of this Society will recall how difficult it was in their early days to get any sort of a standing in the business world. This was due to the fact that the so-called practical business man of that day looked upon the technically trained man as being wholly "theoretical." He failed to understand that the technically trained young man who went out into the world soon acquired practical experience, which, in connection with his technical training, peculiarly fitted him for executive positions.

2 The engineer's connection with the fire insurance business was first brought about by changing conditions in the manufacturing field, which demonstrated the need of more scientific methods for the prevention of loss by fire. Previous to 1880 or 1885, the fire insurance business had been carried on wholly, even by the management of the Factory Mutual Companies, in an empirical manner. The management of the Stock Insurance Companies, previous to that time, devoted little or no attention to questions relating to construction and protection of property. Their efforts were devoted entirely to trying to estimate or guess what was the hazard connected with a particular piece of property and to obtain from the owners a sufficient amount of premium to cover not only the hazard connected with that particular piece of property, but also to protect the company against losses incurred on other property where their

[1]Pres. Boston Manufacturers Mutual Fire Insurance Co.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Textiles.

guess had not been correct. The management of the Factory Mutual Companies was of a somewhat broader character. Their efforts had been devoted largely to advising their members how to care for their property and to giving them what advice they could on how to protect themselves against the fire hazard. At that time, the fire hazard connected with manufacturing was much less than at present, due to slower running machinery, less haste for product, and lack of volatile oils. The fire protection at the mills was of a rather primitive character, consisting of a few standpipes, to which hose was attached, and to which were connected force pumps of limited capacity, usually located inside the mill. In some cases perforated pipe sprinklers had been provided over the more hazardous portions of the property.

3 I must make one exception to my statement that, previous to 1880, the fire insurance business had been carried on wholly in an empirical manner. That exception was in Lowell, and an engineer, James B. Francis, the father of hydraulic engineering in this country, was responsible for the exception. In 1850, the ten corporations located in that city formed a mutual agreement for the payment of all losses incurred by fire. This mutual agreement, which was really an insurance company in itself, was in effect for thirty-seven years. Mr. Francis, who at that time was engineer and agent of the power company in Lowell, which in turn was owned by the ten corporations in question, was given full power to lay out what protection, in his opinion, the mills needed, and to adjust all losses which occurred. He made a scientific study, the first of its kind, of the question of fire prevention in mills. Under his direction, a large reservoir was constructed on Belvidere Heights and was connected by 12-in. pipes with the hydrant and standpipe systems in all the mill yards. He also had a series of tests made of the flow of water in small pipes and through small orifices. These tests were made by Joseph P. Frisell, who, I believe, was a former member of this Society. From the results obtained from these tests, a scientific system of perforated pipe sprinklers was devised, and the mills were equipped with it. Mr. Francis' scientific efforts were to such good effect that the average yearly fire loss during the period of thirty-seven years that the mutual agreement was in effect was only five cents on each one hundred dollars of capital stock of the corporations covered by the agreement. As the property value was largely in excess of the capital stock, the actual loss per hundred dollars of value was even less than five cents. I think it can be truly stated

that Mr. Francis, in addition to being the father of hydraulic engineering in this country, was also the father of fire prevention.

4 In 1878, Edward Atkinson assumed the presidency of the Boston Manufacturers Mutual Fire Insurance Co., the largest and leading company in the Associated Factory Mutual Companies, and one of the largest mutual fire insurance companies in the world. Mr. Atkinson, while not an engineer, was of a scientific and progressive turn of mind. He very soon realized that the changing conditions in manufacturing required a more scientific study of the fire hazards connected with it than had been given in the past. He also realized that better means must be provided for the extinguishment of fires, which, due to manufacturing operations, were constantly occurring in our manufacturing plants.

5 Fires had been very disastrous in woolen and paper mill plants, resulting frequently in the complete destruction of the plants. Many insurance companies were reluctant, and some refused to grant any insurance on these two classes of property. The investigations which Mr. Atkinson instituted resulted in determining that the cause of the trouble in woolen mills was largely, if not wholly, due to the character of the oil used in oiling the stock, and in the paper mills was due largely to fires which originated from hot bearings located in dark and badly constructed basements. He then employed Professor Ordway, of the Massachusetts Institute of Technology, and one of the leading chemical engineers of the country, to make a thorough investigation of wool and lubricating oils. Professor Ordway's investigation was a most complete one and resulted in new types and mixtures of oils for use in oiling wool stock. Safer types of lubricating oils, also better methods of using them, were devised by Professor Ordway. His report was submitted to the manufacturers of oils, with the request that they carry out the suggestions contained therein. Further, it was printed and distributed to the members of the Mutual Companies with the request that they buy only such oils as it advocated, and that they send samples of such oils to the insurance company for test. The final result of Professor Ordway's work and the putting of his recommendations into effect by Mr. Atkinson was the bringing of the woolen mill into the class of preferred risks and largely reducing the hazard of fire due to friction of machinery in paper mills, metal works, and other places where heavy losses had occurred from hot bearings.

6 At about the time that Mr. Atkinson assumed the executive management of the Factory Mutual Companies, electric lighting

was invented and arc lights had been installed in some of the mills, the current being conveyed to the lights over bare copper wires. Realizing the hazard connected with this method of lighting, Mr. Atkinson employed the best electrical talent available to make a study of the problem. The result was the first set of rules for electric wiring ever devised. These rules, while crude in character, are the foundation of the electrical installation rules of today.

7 The automatic sprinkler had been invented just previous to Mr. Atkinson's entering the field of fire insurance. Investigations instituted by Mr. Atkinson clearly demonstrated the value of that piece of apparatus for extinguishing fires. At the time there was only one sprinkler construction company in the field, and the pipe sizes adopted by that company were much too small and wholly inadequate. Many of the mills were equipped with a type of fire pump which was constantly failing. Fire hose was also proving defective.

8 The Inspection Department connected with the Factory Mutual Companies at this time was made up of men who, while they had an excellent knowledge of manufacturing operations, had but little or no experience along engineering or scientific lines. The need of engineers to make a study of the many questions involved was evident. It was also evident to Mr. Atkinson's mind that the department should be reorganized along engineering lines. One of the first men brought in by Mr. Atkinson, and one to whom the success of the department's reorganization was largely due, was John R. Freeman, Past Pres. Am. Soc. M. E. Other engineers were brought into the department. Scientific investigations were made of all kinds of apparatus. Specifications for the Underwriter fire pump were devised and put into effect to such an extent that today that pump is the standard fire pump the world over. Rules and requirements for installation of automatic sprinkler apparatus were also laid down. A laboratory was organized for the testing of all kinds of apparatus, this laboratory being the first of its kind in the world. Mr. Atkinson lived long enough to see this inspection department become the largest and most successful one in the world, and he is largely, if not wholly, responsible for bringing the engineer into the insurance field.

9 The great success attending the work of the engineers in the Inspection Department resulted, as changes were made in the management of the companies themselves, in the transfer of a number of these engineers to executive positions, until today the

executive management of nearly all of these companies is in the hands of engineers. Their influence on the business of fire insurance has been most beneficial, not only to the business of the Factory Mutual Companies, but also to the stock insurance business. As previously stated, the stock insurance business of the country had been conducted largely for the purpose of getting premiums, little attention having been paid to prevention of loss by fire. Many conflagrations had occurred which resulted in the failure of many of the insurance companies, and threatened the failure of many others. Water supplies in many cities and towns were proving deficient. Construction methods were poor. These methods prevailed until the experience of the Factory Mutual Companies demonstrated to the management of the large stock insurance companies that they must, if they were to continue a successful business, reorganize their work along more scientific lines. The National Board of Fire Underwriters, made up of men connected with the leading stock insurance companies, then organized a corps of engineers for the purpose of investigating the conditions relating to water supply, construction, etc., in all the cities throughout the country. These investigations resulted in many improvements being inaugurated, with the result that the fire insurance business of the country is today in a much safer condition than it ever had been in the past. It will, I think, be acknowledged that this condition has been brought about by the work of the engineer, and his influence upon the insurance business of the country is greater today than ever.

No. 1499

HIGHER STEAM PRESSURES

By Robert Cramer, Chicago, Ill.
Member of the Society

The history of the development of the steam engine presents a gradual but continuous and persistent rise of steam pressures, so that it seems natural that engineers should ask the question: "How far can steam pressures be practically and profitably increased?" Pleas for higher steam pressures have several times resulted from investigations of this subject; one of the most remarkable of these was Prof. R. H. Thurston's paper on the "Promise and Potency" of High-Pressure Steam, read before the Society in December, 1896.

2 Professor Thurston naturally had a very clear idea of the thermodynamic principles which make possible better economy by the employment of higher steam pressures, but for several reasons he could not go as far as we are able to today in presenting the practical aspect of the problem. Our progress in this respect can briefly be summarized by referring to the conditions which existed in Professor Thurston's time:

a Superheated steam had hardly been used, and the real advantage of using superheated steam was, of course, not understood.

b The real cause of such cylinder condensation as is not a natural accompaniment of adiabatic expansion was not clearly understood, and it was believed that in order to attain a high ratio of expansion in a piston engine it was necessary to use multiple expansion cylinders.

c The advantages of the unaflow cycle had not been proven.

d Steam turbines of considerable size were in the experimental stage.

e The very best shop methods known did not permit the construction of high pressure steam boilers without involving prohibitive expense.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

3 The following is an effort to present the matter of high steam pressures simply, but clearly, in the light of such knowledge as we possess today.

4 Since the science of thermodynamics has been recognized by engineers as a sure guide to improvement of heat engine economy, there has existed a tendency to increase the temperature range of the working fluid, that is, to increase the temperature at which the working medium absorbs heat and to lower the temperature at which it rejects heat. In steam engine practice this has led to the recognition of well defined limits: a maximum steam temperature of about 600 deg. fahr., above which lubrication of piston engines and maintenance of valve parts and packings is difficult, and a minimum condenser temperature of about 80 deg. fahr., corresponding to 29 in. of vacuum or ½ lb. per sq. in. back pressure. This temperature is so near the usual cooling water temperature that a higher vacuum would require disproportionately large quantities of circulating water. The necessary size of the circulating pump and also the necessary increase in the size of the air pump render higher degrees of vacuum unprofitable.

5 If, taking these limits, all heat were transferred to the steam at 600 deg. fahr., or $600+460=1060$ deg. fahr. abs., and all heat were rejected at 80 deg. fahr., or $80+460=540$ deg. fahr. abs., the Carnot cycle would be realized, and the efficiency of the engine would be $(1060-540)/1060=0.491$. Thus, nearly one half of the heat transferred to the steam would be transformed into mechanical energy.

6 While the condition of the rejection of all heat at the lower temperature can be closely approximated, it is clearly impossible to cause the steam to absorb all heat at the higher temperature. In the boiler, the water must be evaporated at rising temperature, and if the steam is superheated the process of superheating must also take place at rising temperature. Only that heat which converts water at a certain pressure and temperature into steam of the same pressure and temperature is absorbed at constant temperature. This heat is the larger part of the total heat transferred from the furnace to the water.

7 It is apparent from the foregoing that the temperature limits do not alone determine the efficiency of the process, but that the degree of approximation to the ideal Carnot cycle is of great importance. In the best present day practice, except for slightly higher pressures in some few isolated cases, the maximum steam

pressure is 200 lb. per sq. in. abs., and the superheat is 200 deg. fahr. The corresponding temperature of evaporation is 382 deg. fahr. and the bulk of the heat is absorbed at a temperature 200 deg. fahr. below the maximum. It seems reasonable to expect that the approximation to the ideal Carnot cycle, and simultaneously the economy, would be improved by using higher pressure and less superheat, that is, by increasing the temperature at which the bulk of the heat is absorbed, without increasing the maximum temperature.

8 Even a casual reference to steam tables and diagrams confirms this expectation and reveals the remarkable fact that the higher the steam pressure, the less the total heat in the steam if the final temperature be kept constant and correspondingly, the superheat is reduced with advancing pressures. These differences, while not great, are decidedly noticeable, as Table 1 will show. (The values throughout this discussion are taken from Marks and Davis Steam Tables.)

TABLE 1 TOTAL HEAT OF STEAM AT VARIOUS PRESSURES

Temperature of Steam, 600 Deg. Fahr.

Steam pressure, lb. per sq. in.	100	200	300	400	500	600	1574
Temperature of evaporation, deg. fahr.	328	382	417	445	467	487	600
Superheat, deg. fahr.	272	218	183	155	133	113	0
Total heat, B.t.u. per lb.	1323.3	1317.6	1310.7	1305.6	1301.8	1298.8	1176.0

9 An examination of the Mollier total heat-entropy diagram shows that the amount of heat convertible into mechanical energy in adiabatic expansion to any given back pressure is considerably higher for high pressure and little superheat than for lower pressure and more superheat, if the maximum temperature of the steam is the same in both cases. We have thus two causes making for better thermal efficiency with increasing steam pressures at constant maximum temperature: the decreasing total heat of the steam and the increasing amount of that part of the heat which is convertible into mechanical energy in adiabatic expansion.

10 In order to show clearly what gains can be expected from the increase of steam pressures, the following tables have been compiled and the values contained in them plotted in the accompanying diagrams. The tables show the total heat above the feed water temperature, which latter is assumed to be equal to the temperature in the condenser, and the amount of heat convertible into mechanical energy. They also show the efficiency which would be realized if the ideal Rankine cycle were carried out, that is, if the expansion of

the steam were strictly adiabatic without any losses due to either

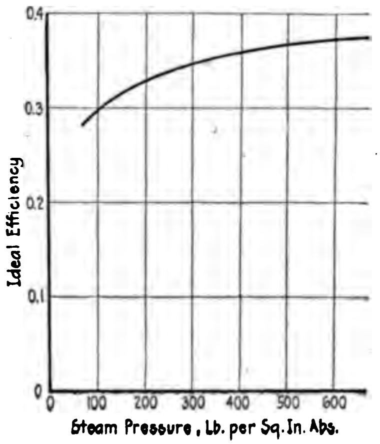


Fig. 1

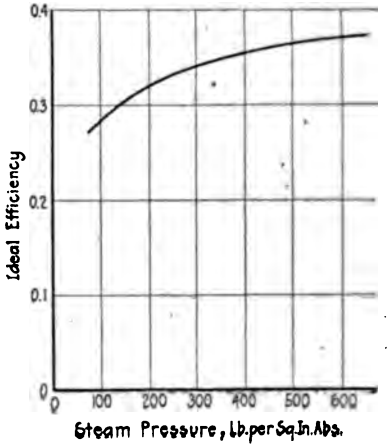


Fig. 2

Fig. 1 Ideal Efficiency. Max. Temp. of Steam 600 Deg. Fahr. Vacuum 29 In.

Fig. 2 Ideal Efficiency. Superheat 100 Deg. Fahr. Vacuum 29 In.

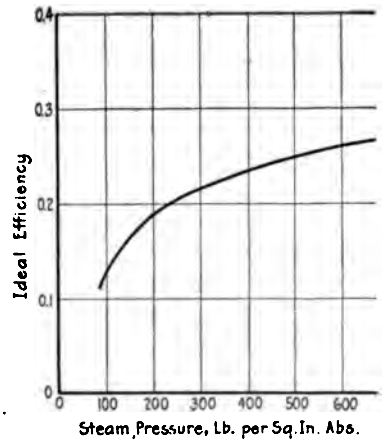


Fig. 3

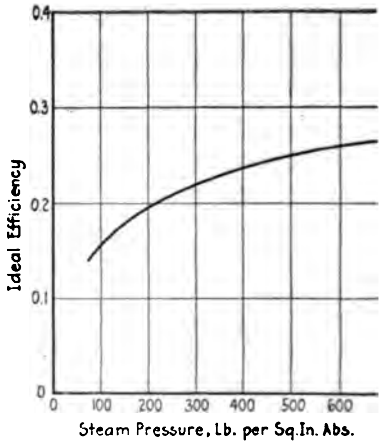


Fig. 4

Fig. 3 Ideal Efficiency. Max. Temp. of Steam 600 Deg. Fahr. Atmospheric Exhaust

Fig. 4 Ideal Efficiency. Superheat 100 Deg. Fahr. Atmospheric Exhaust

heat transfer or friction and if the expansion were continued to the condenser pressure. The tables include in addition the percentage of moisture in the steam at the end of the expansion, which will be discussed later.

11 Table 2 and corresponding Fig. 1 are based on the assumption that the temperature of the steam is 600 deg. fahr. in all cases, and that the back pressure in all cases is ½ lb. per sq. in., corresponding to 29 in. of vacuum. The steam pressure and the amount of superheat are variable quantities.

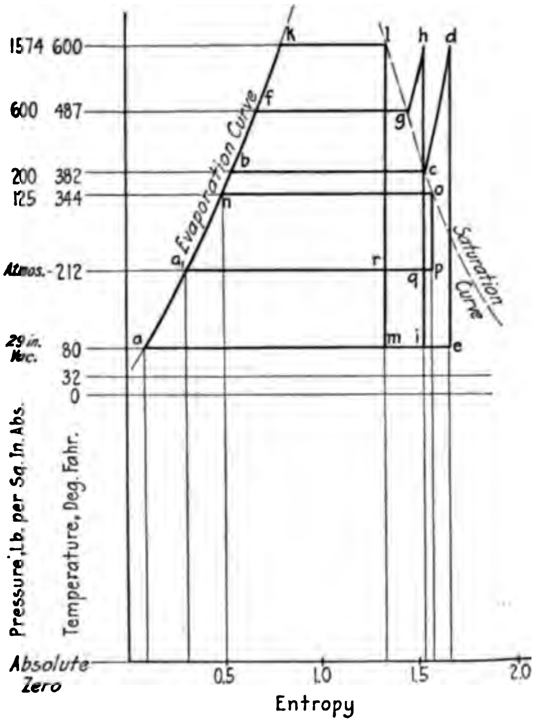


FIG. 5 TEMPERATURE-ENTROPY DIAGRAM FOR STEAM

12 Table 3 and Fig. 2 are based on a constant condenser pressure corresponding to 29 in. of vacuum, and on a constant superheat of 100 deg. fahr., so that both steam pressure and temperature of the steam are variables.

13 Table 4 and Fig. 3 are based on the same conditions as in Table 2 and Fig. 1, except that the back pressure is 14.7 lb. per sq. in., corresponding to atmospheric exhaust.

14 Table 5 and Fig. 4 are based on the same conditions as in Table 3 and Fig. 2, but on atmospheric exhaust.

TABLE 2 EFFICIENCY AT VARIOUS STEAM PRESSURES

Temperature of Steam, 600 Deg. Fahr.

Back Pressure ½ Lb. per Sq. In. (29 In. Vacuum)

Pressure in Lb. per Sq. In. Abs.	Superheat in Deg. Fahr.	Heat Units in 1 Lb. of Steam		Ideal Efficiency	Per Cent of Moisture in Exhaust
		Total above 80 Deg. Fahr.	Converted in Adiabatic Expansion		
100	272.2	1276.4	380	0.298	14.4
200	218.1	1268.6	417	0.329	18.6
300	182.5	1262.5	436	0.345	21.1
400	155.2	1257.9	454	0.361	23.2
500	132.7	1253.6	459	0.367	24.1
600	113.4	1251.0	467	0.373	25.1
1574	0	1128.0	454	0.403	35.7

TABLE 3 EFFICIENCY AT VARIOUS STEAM PRESSURES

Superheat, 100 Deg. Fahr.

Back Pressure ½ Lb. per Sq. In. (29 In. Vacuum)

Pressure in Lb. per Sq. In. Abs.	Temperature of Steam in Deg. Fahr.	Heat Units in 1 Lb. of Steam		Ideal Efficiency	Per Cent of Moisture in Exhaust
		Total above 80 Deg. Fahr.	Converted in Adiabatic Expansion		
100	427.8	1191.7	330.7	0.285	0.187
200	481.9	1209.1	387.1	0.320	0.215
300	517.5	1220.2	414.2	0.340	0.230
400	544.8	1228.9	432.9	0.353	0.240
500	567.3	1237.0	450.0	0.364	0.248
600	586.6	1244.0	461.0	0.370	0.253
684	600.0	1250.0	473.0	0.379	0.258

TABLE 4 EFFICIENCY AT VARIOUS STEAM PRESSURES

Temperature of Steam, 600 Deg. Fahr.

Atmospheric Back Pressure (14.7 Lb. per Sq. In. Abs.)

Pressure in Lb. per Sq. In. Abs.	Superheat in Deg. Fahr.	Heat Units in 1 Lb. of Steam		Ideal Efficiency	Per Cent of Moisture in Exhaust
		Total above 212 Deg. Fahr.	Converted in Adiabatic Expansion		
100	272.2	1144	196	0.171	2.3
200	218.1	1137	224	0.197	5.9
300	182.5	1130	255	0.226	9.8
400	155.2	1126	273	0.243	12.0
500	132.7	1122	282	0.251	13.4
600	113.4	1119	291	0.260	14.7
1574	0	996	305	0.306	28.8

TABLE 5 EFFICIENCY AT VARIOUS STEAM PRESSURES

Superheat, 100 Deg. Fahr.

Atmospheric Back Pressure (14.7 Lb. per Sq. In. Abs.)

Pressure in Lb. per Sq. In. Abs.	Temperature of Steam in Deg. Fahr.	Heat Units in 1 Lb. of Steam: Total above 212 Deg. Fahr.	Heat Units in 1 Lb. of Steam: Converted in Adiabatic Expansion	Ideal Efficiency	Per Cent of Moisture in Exhaust
100	427.8	1059.7	148.7	0.1405	0.062
200	481.9	1077.1	203.6	0.189	0.100
300	517.5	1088.2	235.2	0.216	0.121
400	544.8	1096.9	256.9	0.234	0.134
500	567.3	1105.0	275.0	0.249	0.144
600	586.6	1112.0	289.0	0.260	0.152
684	600.0	1123.0	306.0	0.272	0.158

TABLE 6 COMPARISON OF EFFICIENCY WITH DIFFERENT PRESSURES

Exhaust 29 In. Vacuum

Initial Condition of Steam: Pressure Lb. per Sq. In. Abs.	Initial Condition of Steam: Superheat Deg. Fahr.	Initial Condition of Steam: Tempera- of Steam, Deg. Fahr.	Heat Units in 1 Lb.: Total above Exhaust Temperature	Heat Units in 1 Lb.: Converti- ble in Adiabatic Expansion	Ideal Efficiency	Ratio Compared with 1st Line	Cycle in Temp-entropy Diagram Fig. 5
200	218	600	1267	417	0.3285	1.000	*a b c d e a*
600	113	600	1251	467	0.3730	1.135	*a f g h i* a_1
1574	0	600	1128	454	0.4025	1.226	*a k l m a*
Atmospheric Exhaust							
125	0	344	1010	157	0.1552	1.000	a_1 *n o p* a_1
600	113	600	1119	291	0.2600	1.675	a_1 *f g h g* a_1
1574	0	600	996	305	0.3062	1.972	a_1 *k l r* a_1

TABLE 7 RELATIVE GAIN IN THERMAL EFFICIENCY DUE TO INCREASING STEAM PRESSURE TO 600 LB. PER SQ. IN. ABS.

Final Condition of Steam	29 In. Vacuum		Atmospheric Exhaust	
Initial Condition of Steam	Constant Temperature 600 Deg. Fahr.	Constant Superheat 100 deg.	Constant Temperature 600 Deg. Fahr.	Constant Superheat 100 deg.
As against 100 lb. initial pressure	25 per cent	30 per cent	52 per cent	85 per cent
As against 200 lb. initial pressure	13 per cent	15½ per cent	32 per cent	37½ per cent

15 In all the comparisons based on constant final temperature the steam pressures are carried up to 1574 lb. per sq. in., which is the pressure at which saturated steam has a temperature of 600 deg. fahr. At the present time such high pressures should hardly be advocated, but the figures clearly indicate the gain which might be realized could designs of engines and turbines which are suitable for such steam pressures be successfully worked out.

16 Fig. 5 illustrates these theoretical gains due to high steam pressures by means of the temperature-entropy diagram. This shows clearly how the relative amount of convertible heat increases with higher pressures, and the tabulations in Table 6 give exact numerical values.

TABLE 8 RELATIVE GAIN IN THERMAL EFFICIENCY DUE TO INCREASING STEAM PRESSURE TO 1574 LB. PER SQ IN. ABS.

Final Condition of Steam	29-in. Vacuum	Atmospheric Exhaust
As against 100 lb. initial pressure, 100 deg. fahr. superheat. .	41½ per cent	187 per cent
As against 200 lb. initial pressure, 218 deg. fahr. superheat. .	22½ per cent	105 per cent

17 Tables 7 and 8 give the theoretical percentage of gain for certain comparisons, selected on account of their relation to conditions prevailing in present day practice.

18 These tables show that, even in case of high vacuum in the condenser, the gains, while not overwhelmingly large, deserve careful consideration, and that in case of atmospheric exhaust, these gains are so large as to fully justify an endeavor to realize high steam pressures in practice. Referring back to Tables 2 and 4, we find that, theoretically, the efficiency of high pressure noncondensing engines should be as high as that of many condensing engines under present day conditions.

19 The following considerations will show how far the benefits of high steam pressures may be realized under practical conditions, as far as our present knowledge and experience enable us to realize them.

20 The ideal Rankine cycle cannot be realized in practice. In steam turbines the expansion of the steam cannot be effected without friction losses which are converted into heat and to that extent make the expansion deviate from the adiabatic. It is also impossible to fully extract the mechanical energy which manifests itself in the velocity of the steam, the residual velocity representing a loss,

21 In piston engines of the double-flow type there are large losses due to initial condensation at the time of steam admission and to heat transfer between the steam and the cylinder walls. In the unaflow type of piston engine, properly designed, these losses can be almost entirely avoided. In this latter type of engine it is, however, impossible to carry the expansion down to the condenser pressure, because if such were done the compression, which commences at the same point of the stroke where the expansion ends, would simply be an exact reversal of the expansion. Thus, the compression curve would retrace the expansion curve and no work would be done in the cylinder. In fact in this case no steam could enter the cylinder, because the clearance would contain as much steam of the same pressure and volume as that admitted for expansion.

22 These sources of loss, friction in the turbine, condensation in the double-flow piston engine and incomplete expansion in the unaflow engine, determine the practical limits of the possibility of realizing the Rankine cycle. In large steam turbines this approximation, or the Rankine cycle efficiency, has been carried to about 76 per cent, and it is a remarkable fact that an efficiency closely approximating this has been realized in unaflow engines, even in small sizes.

23 There is no reason why the same relative efficiency should not be realized with higher steam pressures. Such losses as are simply due to temperature differences must be the same if the initial and final temperatures of the steam remain the same, because the temperature changes through which the steam passes in performing the cycle are of the same magnitude in both cases.

24 In a steam turbine higher steam pressures will mean either higher velocities or more stages, both introducing higher friction losses. It is, however, to be expected that by careful design the percentage of these friction losses as compared with the total amount of energy available in adiabatic expansion can be kept the same as in present day practice. In unaflow engines it is quite possible to keep the percentage of loss due to incomplete expansion as low as the corresponding percentage in present-day practice.

25 The foregoing seems to justify the employment of higher steam pressures. Standard boiler designs, however, do not permit the production of steam of a pressure higher than about 200 lb. per sq. in. without sacrificing safety and without calling for an investment in the boiler plant high enough to offset the gain in economy caused by higher steam pressures.

26 The solution of the problem of boiler safety under high steam pressures demands two fundamental changes in boiler design:

- *a* The boiler must be constructed entirely of tubes of relatively small diameter. All drums and vessels of large diameter, as well as all flat surfaces (even if stayed) must be abandoned.
- *b* Expanded, beaded or riveted joints exposed to the action of the fire must be avoided. That part of the boiler which receives the heat of the furnace must be practically a one-piece structure.

27 It is important that in meeting these requirements the essential characteristic of water circulation in the boiler be retained in order to make possible a control of the steam pressure and of the water content of the boiler by simple means. Flash boilers, while permitting high steam pressures, are not desirable because they require complicated automatic regulating devices necessary on account of the interdependence of feed and fire control.

28 In considering boilers for high steam pressures, it must not be overlooked that the mass of water in the boiler is at a higher temperature than that in a low-pressure boiler. This difference amounts to approximately 100 deg. for 600 lb. pressure as compared with 200 lb. Even if the heating surface is made very large, or, in other words, the evaporation per square foot is kept very low, the stack gases will leave the high-pressure boiler at a temperature 100 deg. higher than the corresponding gases of the low-pressure boiler. The boiler efficiency, other things being equal, is correspondingly reduced.

29 It is, of course, possible to meet this difficulty by making the heating surface of high-pressure boilers larger in proportion to what is current practice in boilers working under pressures used today. It is also possible to increase the effectiveness of the heating surface by proper design. Boiler designers are realizing the possibilities in this direction, even with steam pressures not higher than 200 lb. This is demonstrated by the fact that modern boilers show an evaporation per square foot of heating surface twice as high as was customary only a few years ago, at the same time realizing a better efficiency than formerly.

30 It therefore seems reasonable that highly efficient high-pressure steam boilers can be made, even without resorting to special heat saving apparatus to offset the effect of the higher temperature of the water in the boiler. The possibilities of these extraneous heat

saving devices should, however, not be overlooked. Even though the waste gases leaving the boiler have a temperature of 700 deg. fahr. or more, a properly installed economizer will reduce this temperature to any desired degree within the limits given by the temperature of the feed water. Even a very small economizer will save enough heat to offset the loss due to the higher temperature corresponding to higher pressure. In some instances, the waste gases might profitably be used to heat the air which supports combustion in the furnace.

31 These remarks are merely rough indications of the possibilities, but they point out the fact that the higher temperature of the water in the boiler should not render it impossible to obtain the very best overall boiler efficiency, even with very high steam pressures.

32 The question of strength of constructional parts outside of the boiler to withstand high steam pressures is of great importance. In reciprocating engines this question can be answered from practice, because pressures far exceeding 600 lb. per sq. in. are used successfully in gas engines and Diesel engines, and therefore the design of cylinders for high steam pressures should not present any difficulties which have not already been overcome in other types of machines.

33 In steam turbines, especially in those of larger size, the casings of large diameter would need to have disproportionately heavy walls were they required to withstand pressures much higher than those employed today. The solution of the problem of adapting turbines to high pressures is found in the principle underlying most present designs of large steam turbine units. According to this principle, the pressure of the steam is greatly reduced in the first nozzle and the resulting high velocity is utilized in several rows of blades of the velocity stage type. Thus, it is possible to confine high pressures to the steam piping and to the "steam belt" carrying the steam to the first nozzles, but to have comparatively low pressure even in the first part of the casing. The lesser efficiency of the velocity staging is not so serious at the high-pressure end as it would be at the low-pressure end, because the loss manifests itself in a somewhat higher superheat of the steam entering the succeeding stages and is therefore partly recovered.

34 With high pressures, the difficulties with piping and fittings are greatly reduced on account of the reduced specific volume of the steam. Even if lower rates of flow than those customary in present practice are permitted in high-pressure steam lines (on account of the

greater density of the steam), the fact that one pound of steam at 600 lb. pressure occupies approximately one-third of the space required by one pound at 200 lb. pressure reduces the required size of the piping and fittings to such an extent that both difficulties in design and cost for a given capacity are, if anything, less than for lower pressures.

35 In piston engines the question of cylinder lubrication is important. It is apparent that high steam pressures will permit neither slide nor Corliss valves. The advent of higher pressure will cause the poppet valve to come into its own in America, where it is now seldom used, in spite of the great success it has had in Europe for many years, lately the flexible seat type especially.

36 The largest part of the lubricating oil now used in piston engines is required for the steam distributing valves. That part required by the piston and piston rings is very small and the possibility of sufficient piston lubrication is not affected by higher initial steam pressures, because even in case of very high mean effective pressures in single-acting engines, where the piston serves as crosshead and therefore requires constant lubrication, the piston can be made long enough to keep the pressure per square inch within proper limits.

37 For a given output the cost of a unaflow engine suitable for high steam pressures should be less than that of double-flow engines, as soon as it can be manufactured under economical conditions of manufacture. Such an engine, if single-acting, has but one very simple organ of steam distribution and on account of its high mean effective pressure its weight per horsepower is low. Even in a single-acting engine of this type the mechanism is utilized twice as efficiently as in a four-stroke cycle Diesel engine.

38 Steam turbines for high steam pressures would probably cost a little more than those using lower pressures and giving the same power, on account of the extra stages required. This extra cost would be offset by a considerable saving in the required condenser cooling surface on account of the larger percentage of moisture in the steam entering the condenser and the reduced steam consumption due to better thermal efficiency. Table 9, based on the values of Table 2, will make this clear.

39 The question of stuffing boxes can be entirely eliminated in both turbines and piston engines—in turbines because high steam pressures need not be carried beyond the first nozzle, and in piston engines because the single-acting type of engine is from many points

of view the logical design for high pressures. In such a piston engine the stem of the one valve can be provided with a "labyrinth" packing, as has been successfully done in practice.

40 In the design of power houses the question of the proper arrangement of the auxiliary apparatus, such as feed pumps, air and circulating pumps, fans and stokers, would have to be considered from the point of view of high steam pressures. In very large plants all auxiliaries are often driven by electric power, and the current for them is furnished by a separate power unit. Inasmuch as these "house service units" in modern power plants have capacities of 2000 kw. and over, it is a simple matter to operate them directly with high-pressure steam.

TABLE 9 REDUCED STEAM CONSUMPTION DUE TO BETTER THERMAL EFFICIENCY

Initial Condition of Steam	200 lb. 600 Deg. Fahr.	600 lb. 600 Deg. Fahr.
Ideal efficiency with 29 in. vacuum	0.329	0.373
Percentage of steam in exhaust	0.814	0.749

Ratio of condenser cooling surface $=0.329\times0.749/0.373\times0.814=0.812$. Thus, a saving of about 19 per cent of the condenser cooling surface might be expected.

41 Where it is desired that the auxiliaries be driven entirely independent, it should be possible to drive them with high-pressure steam turbines if the size of the auxiliary unit warrants this, or by high-pressure unaflow engines for the smaller sizes. In this latter case, the objection might be raised that the exhaust of the unaflow engine would introduce a certain amount of lubricating oil into the condenser. This question can be met in different ways. If the unaflow engine is of the vertical single-acting type, the lubrication can be reduced to a very small amount, and could possibly be accomplished with graphite only, so that the amount of lubricant introduced into the condenser would be insignificant and harmless.

42 Another way of meeting this question would be the noncondensing operation of the unaflow engines, using the heat of their exhaust in a feed water heater. This manner of operation appears to be very attractive, because the economy of noncondensing unaflow engines. as compared with noncondensing engines of other types, is extremely high. Where clean feed water is readily available, the method would be readily applicable, or the unaflow auxiliary units may be connected to an independent small jet condenser.

43 There are other possibilities, and it seems clear that the question of the auxiliary drives can easily be solved satisfactorily.

44 In the foregoing only some of the important points in the design of prime movers utilizing high steam pressures have been touched upon, and these only in the most general way. It seems clear, however, that if steam pressures are increased to, say, 600 lb. per sq. in. without using temperatures higher than those employed in modern practice, the difficulties encountered by the designer are not formidable and are more easily met than in the case of some types of explosion engines which have been successfully designed. The result to be attained by adopting such high steam pressures appears to be fully worth the effort, because thermal efficiencies closely approaching those of explosion engines can be realized with simpler and less expensive apparatus and consequently better overall economy, at the same time retaining all the great practical advantages which steam utilization has over any other method of producing power.

DISCUSSION

William Kent (written). This paper contains some interesting figures of the gain in steam engine efficiency that may be obtained by increasing the steam pressure up to 600 lb. per sq. in. absolute. They supplement very conveniently some figures for the efficiency of the Rankine cycle which I calculated recently for pressures up to 250 lb. per sq. in. and superheat up to 300 deg., and which will be found on page 1091 of the 9th edition of my *Mechanical Engineers' Pocket Book.* Taking the figures corresponding to 29 in. vacuum:

	Superheat, deg. 0	50	100	150	200	250	300
Press. lb. Absolute.	Efficiency of Rankin Cycle, per cent.						
200	32.2	32.3	32.6	32.8	33.1	33.4	33.8
225	32.7	32.9	33.1	33.4	33.6	34.0	34.3
250	33.2	33.4	33.6	33.9	34.1	34.5	34.8
Superheat corresponding to temperature 600 deg.				113.4	132.7	155.2	182.5
300			34.0	...	...	...	34.5
400	(Mr. Cramer's		35.3	...	...	36.1	
500	figures)		36.4	...	36.7		
600			37.0	37.3			

Taking 225 lb. per sq. in. and 200 deg. superheat, corresponding to a temperature of 591 deg. and a Rankine cycle efficiency of 33.6 per cent, as about the standard of the most recent practice, the improvement that may be made by using a pressure of 600 lb. and a temperature of 600 deg. is (37.3 — 33.6) or 3.7, or 11 per cent of 33.6 per cent. In order to obtain this gain it will be necessary to build stronger boilers than we now have, to use economizers to reduce the flue gas temperature, and to take extra precautions against steam leakage. These improvements are quite within the range of practicability, and it may be worth while to make them to gain 11 per cent in efficiency, especially in locations where fuel is expensive and in plants having a high load factor.

Mr. Cramer says that it is possible to increase the effectiveness of the heating surface by proper design, and that modern boilers show an evaporation per square foot of heating surface twice as high as was customary only a few years ago, at the same time realizing a better efficiency than formerly. It does not appear that the improvements in rate of driving and in efficiency at high rates of driving have been due to any changes in the design of proportions of the heating surface; they have been due to greatly enlarged combustion chambers, mechanical stokers, and control of the air supply according to the indications of gas analyses. The highest efficiencies have been obtained with many different forms and proportions of boiler, and no evidence has yet been obtained that any new form of boiler will give greater effectiveness per square foot of heating surface than the forms of boilers that were built forty years ago.

R. J. S. Pigott. This interesting paper has presented the theoretical side of the question rather more strongly than the practical side. The sources of loss, especially in the turbine, have been passed over perhaps a little too lightly.

Those of us who have been interested in turbine designing for the last few years have noticed the very marked difference in the efficiency ratio of the various stages, between high and low pressure end of the machine. We know that the high pressure stages are very much less efficient. For instance, in a turbine working at 200 lb., with 100 deg. of superheat, the efficiency of the first stage is possibly 55 per cent or a little higher, depending upon the conditions, and that figure increases as we reach the lower stages, finally reaching 85 per cent, or even more, in the low pressure end.

The average efficiency ratio is ordinarily 70 to 75 per cent, including generator losses. The figure quoted by Mr. Cramer, of

77 per cent, must be quoted for overall efficiency, including generator loss. The actual efficiency of the Interborough Rapid Transit Company's turbines is over 80 per cent, if the losses to the generator are not considered, representing a high pressure efficiency of 70 per cent, and in the final stages somewhat over 90 per cent.

It is a fact that the efficiency of the stages of a turbine depends chiefly upon the density and quality of the steam, other conditions being equal. The reason for such low efficiencies in the high pressure stages is the high friction loss due to the high density of the steam, and in the case of turbines running on saturated steam, the additional friction due to the moisture. It is certain that if pressures are increased, the high pressure stages are going to become less efficient,—more especially as in substituting pressure for superheat, we are increasing the amount of moisture developed in the turbine and advancing the dew point to higher stages. In other words, the efficiency of each stage will be somewhat decreased, due to the presence of a greater quantity of moisture.

The author's expectations of the efficiency closely approximating present efficiency would not be reached with actual turbines. It is quite evident from our experience with up-to-date turbines that we cannot get very high efficiencies with the very dense steam to be expected.

Another point from the practical side is that of steam piping and pipe joints. It is generally felt that the best joint is one in which there is no gasket. Very good success has been made with ground joints, but the character of workmanship and pipe construction required for the ground joint is almost too high for ordinary construction and cannot be obtained in the average plant. Therefore the forms which require metallic, or asbestos, or other soft gaskets, must be used for these temperatures and pressures, and there is trouble enough already with 200 lb. pressure.

In the plants of one large western company, the joints are made by fusing the edges of the lap flanges with the acetylene torch, so as to avoid gaskets. This, it seems, is the first step in the change that would be required for higher pressures. The next step would be to abandon flanges almost entirely, and weld the pipes in place without joints. We are now using large quantities of welded material under very severe conditions. With the degree of skill now developed, satisfactory welding could be readily accomplished the field. It would mean the substitution of a welding gang for steamfitters, on the larger work.

G. I. Rockwood. In regard to the facility of making pipe joints, I have been rather surprised, in working out the problem of making pipe union joints out of ground bronze seats, to find what a poor job of grinding is usually done. If the grinding is done properly, as I find it can be done with a properly developed machine, the joint will stand all pressures up to 10,000 lb. per sq. in., and the union will burst before it will leak.

Pipe unions will probably replace flanges in the larger sizes, and if higher steam pressures are to become at all general, the practical requirements of pipes, built in sections rather than welded together, will call for pipe unions properly made.

Richard H. Rice. The subject of utilization of higher steam pressures in motive power apparatus has been before the designers of such apparatus for some time. The first active proposal in that direction was made, I think, by Mr. Ferranti of England, some two or three years ago, and a great deal of investigation on this subject has been stimulated.

The author has very clearly set forth the theoretical possibilities involved in the use of higher pressures. The practical consideration is that of temperatures. Several years ago the writer constructed and operated an engine utilizing a maximum temperature of 800 deg. Although this engine operated successfully, we find, in investigating the materials to be used for these pressures and temperatures, at 600 deg. practically all the materials we would ordinarily use in the construction of such apparatus begin to lose strength. Therefore, we are limited to that critical temperature, or else have to choose special materials, or, as a third possibility, must greatly increase the proportions of parts in order to keep down the stresses.

Expansion of pipes would also be troublesome at high pressures on account of the rather extreme thickness of the pipes and their consequent stiffness.

The effect on turbine design has been clearly brought out by Mr. Pigott. In addition to the difficulties in realizing the highest efficiency which he has pointed out there is the very important one of leakage. If we attempt to use very high pressures on the upper stages of turbines, we will certainly have to develop improved forms of packing.

All these considerations do not by any means indicate difficulties which cannot be surmounted; on the other hand we are fitted at the present time to undertake the development of apparatus to meet these conditions. A considerable amount of development work will be necessary, however, and the new conditions will give rise to radical changes in the design of power stations. For instance, it is obvious that if these expansion stresses in pipes are to be difficult, we should limit the length of the pipe as much as possible, which means putting the boiler and turbine close together.

The question of higher pressures is entirely a commercial one. Assuming that the gains indicated in the paper, or even half of them, can be realized, can apparatus be constructed of such cost and placed in our power stations under operating conditions, which will involve operating expenses in such amounts that the total operating gain, counting interest on the investment, operating cost, and general reliability, will make the proposition a profitable one for our power stations?

Walter N. Polakov. The question of advantages of higher steam pressures may be regarded from two angles of view. One is purely a theoretical consideration, setting aside the commercial advantages, and the other, the commercial advantages under the present state of the art.

If the steam is used at high pressure, and saturated, the condensation may have very quick deteriorating effects on the blading of the turbine. This is one of the financial questions which ought to be primarily considered before it is decided whether at the present state of the art the high pressure without superheat is advantageous.

Another consideration is whether the added cost of construction of the steam vessel for the higher pressures will be warranted. Incidentally, in European practice, temperature of superheated steam is not limited to 600 deg. fahr., but goes up to 800, and in some cases to 1000.

Carl C. Thomas. In regard to turbine design for higher pressures, the high pressure stage would necessarily be of the impulse type, because of the otherwise extensive leakage past the ends of the blades with steam of such high density as that at 600 lb. The turbine would therefore seem to require one or more impulse high pressure stages, velocity compounded, probably followed by Parson's *stages.*

Professor Carpenter's tests on the White steam car showed that at high pressures very good economy could be attained, even with small reciprocating engines. An engine of the general type of the Westinghouse single-acting vertical engine, with the lower end of the cylinders closed so that a vacuum could be maintained, but fitted with an exhaust port like that of the Stumpf engine, might be very efficient for use with these high pressures.

The questions of boiler material and design, however, for large units and very high working pressures present difficulties which may well cause the boiler builder to hesitate before making guarantees of endurance. There is a very considerable difference between the relative degrees of danger in handling large quantities of steam and water under 225 lb. pressure and 150 deg. superheat as compared with dry or slightly superheated steam at 600 lb. pressure. If the higher pressures are to be used, we must expect to wait some time before entirely satisfactory boilers of large capacity can be developed to suit the new conditions.

D. S. Jacobus. The statement is made in the paper that standard boiler designs do not permit a construction for higher pressures than 200 lb. per sq. in. This is below the pressure often carried in present practice. There are standard designs operating successfully at high capacities at over 300 lb. per sq. in., and for special conditions as high as 500 lb. The figure of 200 lb. is therefore too low to set as the limiting pressure for standard designs.

The Author. Professor Kent has pointed out that the improved boiler efficiency in recent designs is not due to any change in the boiler itself, but rather to the furnace. If the point is conceded that high efficiencies can be obtained with high pressure boiler designs, then whether the necessary changes are made in the boiler or in the furnace is immaterial. I concede that most of the recent improvements in boilers have been made in the furnace, and not in the boiler proper, and the most efficient boiler designs used today are practically the same as those of twenty years ago. The question brought up in the paper is, however, entirely different. If steam pressures as high as 600 lb. are used, it will be necessary simply from that point of view to change the boiler design, and as pointed out in the paper, to do away with large drums and a small factor of safety.

Mr. Pigott raises the question of efficiency of the high pressure stage in the turbine. In the first place, it will almost be impossible in large turbines to utilize some of the principles employed today. It will not be possible to carry the high steam pressures right into the turbine. As a matter of fact, in most of the present large turbine designs, with the exception of the straight Parsons type, there is a large drop in pressure before the steam enters the turbine proper; in other words, in the very first drop of pressure, a good part of the energy of the steam is converted into velocity. The steam after it leaves the first nozzle, has now to be utilized in a scheme of velocity staging, and in that respect the turbines for high pressure will be practically identical with the present types of turbines, but the velocity stage at the beginning will be carried out to a greater extent.

It is quite true that with high pressures, losses in the high pressure stage of the turbine will be greater, but it must be remembered that all the losses occurring in the individual stages reappear in the steam in the shape of added heat, because there is no other way of disposing them. While this loss of heat decreases the efficiency of the high pressure stage, it is partly utilized in the lower stages. Of course the losses that occur at the high pressure end, say between 600 lb. and 200 lb., will not be fully recovered in the low pressure end of the turbine, but by proper design they can be recovered to a very great extent.

Regarding piping, in the western plants which Mr. Pigott mentions, they have a welded seam between the flanges. The strength of the joints is given by the flange bolts, and the welding is used merely as a seal, and that works out in practice very well. Of course it is necessary in case of repairs to cut the seal, but on the other hand most of the repairs in these cases are caused by failure of the gasket, and that is the very thing the welding avoids.

I do not mean to point out that it is not possible to depend on the strength of the weld. In high pressure design with which I have experimented and carried out a great deal of practical work during the last few years every joint is welded. I have tested welded joints with pressures between 9,000 and 10,000 lb. per sq. in. and have burst the tube, being unable to injure the weld. It is therefore possible to depend on the strength of the weld if it is properly made.

At higher pressures, it is possible with a pipe line less than

12 in. in diameter to supply all the steam necessary to operate a 20,000 kw. turbine, and from that point of view the question of strength of joints is disposed of, because it is certainly easier to make 12-in. joints and welds for 600 lb. than to make 18 and 20-in. joints for 200 lb.

Mr. Rice has pointed out that loss of strength of the materials occurs in temperatures higher than 600 deg. The paper does not advocate raising temperatures while increasing pressures. For that reason the question of the loss of the strength in the material at high temperatures is eliminated.

In reply to Professor Thomas, the unaflow design of reciprocating engine is specially fitted for the adaptation to high pressures. It is a remarkable fact that such knowledge as we possess today points to the possibility of realizing fully in a unaflow engine efficiencies given by theoretical considerations. Such engines, too, are much simpler, and are also less bulky than other engines used on lower pressures to obtain the same results.

In the light of Doctor Jacobus' criticism, I must modify the statement that the present day boiler designs are not suitable for higher pressures than, say, 200 lb. At higher pressures, however, say 250 lb. and over, standard boiler designs become expensive enough to offset all the gain which would otherwise result from the employment of these high pressures.

Boiler designs are possible which will utilize high steam pressures without necessarily increasing the cost of the boiler to such an extent that the theoretical gains are almost wholly wiped out. The question is not merely a theoretical one; although the paper attempts to present only the theoretical aspect, it is primarily an economic one. Most of the discussors concede the possible theoretical gains, so the question remains: Can high pressures be realized in practice, without involving such high charges, due to initial cost of installation, interest on the investment, repairs, and up-keep, that the theoretical gains are offset? It has been my contention that these extra expenses necessitated for high steam pressures are not high enough to offset the theoretical gain, even to a small extent. As the problems of design are more clearly understood, and are properly met, we will within a very short time see the general adoption of higher steam pressures, especially in large power plant work.

No. 1500

DESIGN OF FIRE TUBE BOILERS AND STEAM DRUMS

BY F. W. DEAN, BOSTON, MASS.
Member of the Society

The appointment of a committee by this Society to devise rules for the construction and care of steam boilers, which are to serve as a standard which may be adopted by any state or municipality, shows the importance now attached to the use of such materials and designs as will promote safety.

2 The rules[1] now drawn up seek to unify boiler practice in this country and Canada; they embrace specifications for materials of the various kinds used in boiler construction, stresses to which the most important parts should be subjected and features of design. One of their objects is to compel boiler makers in small and obscure places to adopt proper methods of construction, so that the feeling that proximity to some boilers is dangerous may be eliminated.

3 While the rules formulated are excellent, it is not possible for any set of rules to cause good designing in all respects; and I have thought it best, in order to extend this work, to consider what makes steam boilers dangerous, and how certain parts should be designed in order to make them safe. I mention *design* rather than *material,* because I do not think there have been many boiler explosions caused by inferior materials.

CAUSES OF BOILER EXPLOSIONS

4 It is easy to recall the time when the causes of boiler explosions were mysteries and various theories were advanced to account for them. One commonly assumed cause was the introduction of water on hot plates. It is now known that hot plates cannot vaporize sufficient water to create the necessary amount of steam to produce pressure, and

[1] Trans. Am. Soc. M. E., Vol. 36, p. 977.

Presented at the Annual Meeting, December 1915, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

that the contact of hot iron or steel with cold water has no appreciable physical effect upon the metal.

5 In a general way it can be said that boilers explode because they are not strong enough to stand the pressure to which they are subjected, but such a statement as this needs explanation. When boilers are new they are usually strong enough to stand the pressure for which they are built, but in service something weakens them. Corrosion sometimes does this, but not often sufficiently to cause an explosion.

6 The great and usual cause of weakness is such a form of some part that, with the application and removal of pressure or with variations in pressure, this part bends first one way and then the other, and this causes cracks. It is in general the shells at the longitudinal joints which act in this way, and it is safe to say that nearly all explosions of fire tube boilers are caused by the bending and resultant cracking of this part of the shell. Occasionally a boiler corrodes sufficiently to weaken it to the breaking point, but this is very uncommon.

7 As is now generally understood, the most prolific cause of explosions of fire tube boilers has been lap longitudinal joints, and the use of butt joints with inside and outside covering plates has, so far, done away with such explosions. The lap joint makes the boiler non-circular at and in the vicinity of the joint; and when pressure is applied the plate, in its effort to become circular, bends somewhat, and on the reduction or removal of pressure, it tends to return to its original form. The frequent repetition of these actions causes the plate to crack and finally to become too weak to stand the stress caused by the working pressure. Boiler plates with lap joints have often cracked entirely through for a greater or less distance, and the escape of steam has given warning in time to prevent an explosion.

8 While I have said that butt joints have prevented explosions, I should not fail to mention that there have been cracks in the butt joints of four horizontal return tubular boilers, and that their leakage gave warning so that the boilers were taken out of service before they exploded. These occurrences are causes for anxiety, and the future may bring occasional explosions of butt joint boilers as thus far built in this country for we cannot expect warnings in all cases.

THE DESIGN OF BUTT JOINTS

9 It is of the utmost importance to design butt joints that will not bend in service, and it is of no less importance that the defects of the butt joint generally used in land boilers in the United States and Canada should be most impressively pointed out. I refer to the joint

which has a narrow butt strap on one side of the shell and a wide one on the other. This joint is one-sided, and its center of resistance does not coincide with the center of pull of the shell plate. That part of the wide strap which extends beyond the narrow one is riveted to the shell, and this outer part of the joint is a lap joint with its peculiar defects. The rivets in this part of the joint are overhung and in

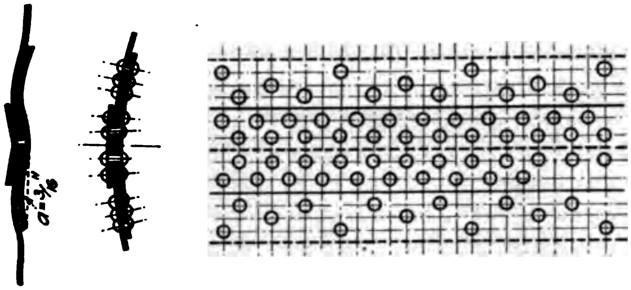


FIG. 1 BENDING OF BUTT JOINT TESTED TO DESTRUCTION

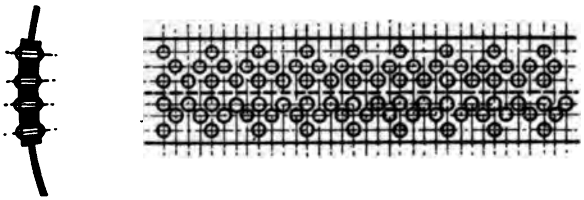


FIG. 2 BUTT JOINT USED IN MARINE PRACTICE. EFFICIENCY ABOUT 85 PER CENT

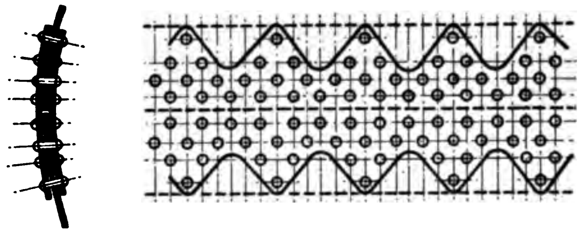


FIG. 3 BUTT JOINT WITH EFFICIENCY OF 92 TO 94 PER CENT

service tend to tip over and bend the joint. This is only another way of stating that the joint is a non-central resisting device, and must cause the plate in and near the joint to bend when the boiler is subjected to pressure. Fig. 1 shows how this bending occurs when such a joint is tested to destruction, this being the result of several actual tests.

10 In order to prevent a joint from bending, a butt joint with straps of the same width and with all rivets in double shear should be used, as is always done in Europe. The center of resistance of such a joint coincides with the center of pull in the shell. The joint when tested to destruction remains straight, without bending; and by its use the only probable cause of boiler explosions as far as the design of the joint is concerned is eliminated. There are various designs of such joints ranging from the simplest joint with double covering plates of equal widths, to the joints shown in Figs. 2 and 3.

11 Fig. 2 illustrates a design which is used largely in marine practice in this country and abroad and which has a theoretical efficiency of about 85 per cent. Fig. 3 shows a more efficient joint, having a theoretical efficiency of from 92 to 94 per cent. In the joint in Fig. 3 the outer strap is cut away between the rivets in order that it may stand caulking; the high efficiency is secured by the wide pitch of the outer rows of rivets.

BENDING PLATES

12 There is still another detail of boiler making that needs more careful attention than it has usually received. On account of the difficulty of bending plates to an exact circular form out to the edges, it has been customary to hammer the edges to this form. The result is that the plate is injured and the shape is only approximately exact, and this occurs at a part of the boiler that is in every respect the weakest and most troublesome. In some shops methods are taken to minimize this defect, and the rules of this Society require that the ends of the plates shall be pressed into shape, and hammering is prohibited. In England presses are made to do this work properly and press the whole plate, instead of rolling it. One of these presses, as made by Fielding and Platt of Gloucester, is illustrated in Fig. 4.

BRACES

13 Boiler head braces of the crow foot or similar types are usually designed so thin that they bend in service where the foot joins the rod, and are likely to break finally at this point. This detail should be made stiffer than is commonly done.

14 Through rods above the tubes of horizontal return tubular boilers should be supported so that they cannot vibrate, and the supports should be stiff enough to prevent movement in any direction, instead of merely supporting the weight. If any braces are used below the tubes, through or head-to-head braces, rather than braces riveted

to the shell should be employed. Such braces should not, however, pass through the back head on account of the nuts being in contact with the fire. The rods should be secured to angles riveted to the back head but separated from it 2 to 4 in. by ferrules around the rivets, in order to permit the removal of dirt between the angles and the head.

15 Although not related to safety, it is best not to have the rods above the tubes pass through the back head, because the nuts interfere with an efficient method of covering the back connection with fire

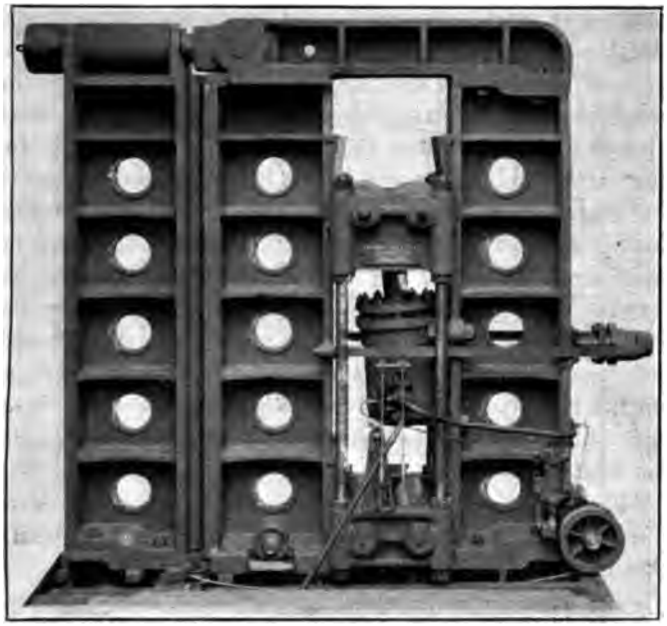


FIG. 4 FIELDING AND PLATT PLATE PRESS

brick. There are various ways in which nuts can be avoided, and Fig. 5 shows methods of staying both above and below the tubes. In the former case diagonal braces are used for staying the upper parts of the tube plates in order to give more room for inspection, and the heads are stiffened by riveting on thick plates. In England plate gusset stays are preferred to through rods or diagonal stays, but in this country they are seldom used. There is, however, no reason to doubt their efficiency, although any head stay attached to the shell tends to bend the latter and throw it out of equilibrium. Nevertheless I have never heard of a rupture coming from this cause.

RIVET HOLES AND RIVETING

16 Riveting is now generally done by hydraulic machines, which have shown themselves superior to any other kind. In consequence of the slow movement of a hydraulic plunger the rivets have time to enlarge and fill the holes; and from the solidity of action and steady holding power of the machine the plates are firmly pressed together, with the result that joints riveted by this type of machine are tighter than those made by any other.

17 It is the common practice in this country to punch the holes of boilers $\frac{1}{8}$ in. or $\frac{1}{4}$ in. small and then to drill them to size with all plates and covering plates in place. This is a great advance in practice over punching to size, but it is not satisfying to the imagination and it may be one reason why plates exposed to the hottest gases crack between the rivets and their edges. Another reason for such cracking may be bulging caused by too much pressure by the riveting machine on the rivet. Sometimes such bulging is very apparent. A still further advance in practice is to punch one butt strap for each joint with small holes and use it as a template for drilling not only itself but the main plate and the other butt strap. Similarly, the holes in one plate of a circular seam may be punched small and used for a template for drilling the other holes. The best way, however, and the one which I hope to see adopted everywhere, is to drill all holes from the solid. I think it will be found that with proper tools this is the cheapest method.

18 The conical rivet head is being displaced by one of more or less spherical form which has the advantage over the former in having a thicker edge and increased holding power.

REINFORCING PLATES FOR THE OUTSIDE FIREBOXES OF VERTICAL BOILERS

19 Above the staybolt level of vertical boilers the outside fireboxes are subjected to the full stress that comes from the steam pressure, unreduced by any connection to the inside firebox by the staybolts. In some designs the whole outside firebox is made of the thickness required to stand the pressure as if unstayed; in other designs there is a short course of increased thickness just above the staybolts, and in still another design the outside firebox plate is thin for its whole height and is reinforced above the staybolts by riveting a band of steel around the inside of the plate.

20 I disapprove of the last of these methods and mention the fact here because the object of this paper is to point out the causes of

boiler explosions and to advocate methods of construction that will reduce, if not do away with them. I have already stated that shell explosions are nearly always caused by the bending of the plates, and the inside reinforcing plates of vertical boilers, just described, can do no good without bending. They cannot then prevent the main plates from being overstrained, and they are therefore possible causes of explosions.

REVERSED FLANGES IN VERTICAL BOILERS

21 Still further in accordance with the object of this paper, I shall describe the action of that type of vertical boiler which is

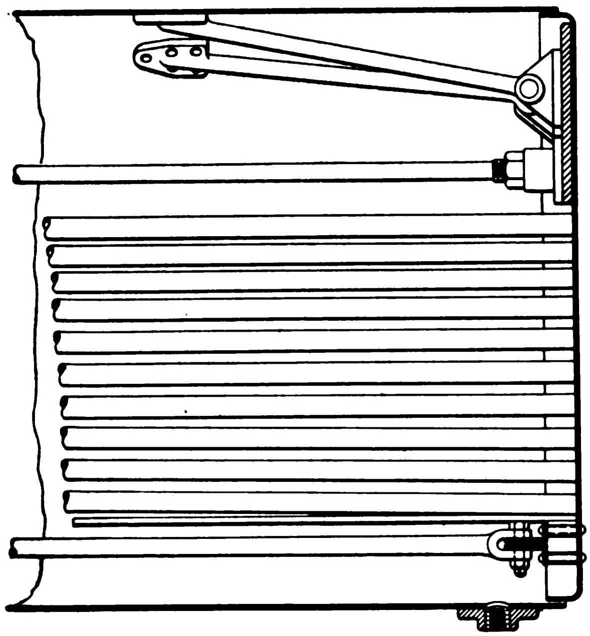


FIG. 5 DIAGONAL AND LONGITUDINAL STAYS WITHOUT OUTSIDE NUTS

changed in diameter above the firebox by means of a reversed flange, Fig. 6. On account of the ease with which this flange bends, this type of boiler elongates when subjected to pressure, and, under test pressure, to a considerable extent. Even the pulsations in pressure coming from the opening and closing of the inlet valves of steam engines cause the boiler to change its length each time, and this action and others have caused many of the reversed flanges to crack. The effect has been reduced by making this flange of a less flexible form and increasing its thickness. It is not a good plan, however, to have flexible means of connecting the ends of boilers, and when the flange is made so thick that it is not flexible its object is no longer accomplished.

22 Another harmful effect of the elongation of boilers of this type is the bending of the lower tube plate upward and the upper one downward. This tends to pull the outer tubes from the tube plates and may cause explosions.

23 In order to obviate the two defects described, the reduction in

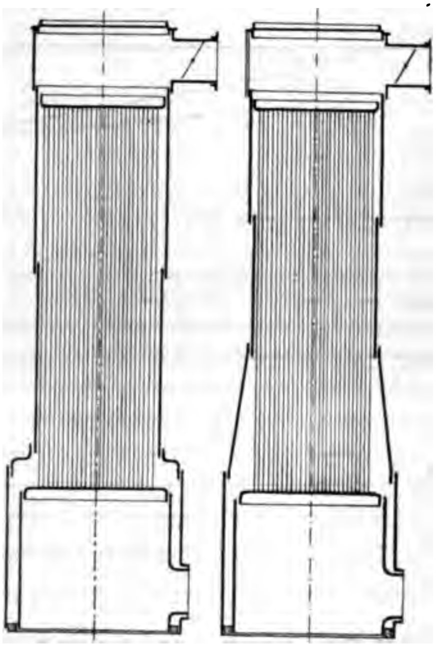


Fig. 6 Fig. 7

Fig. 6 Vertical Boiler with Reversed Flange
Fig. 7 Vertical Boiler with Conical Course

diameter should be made, as it has been many times, by a conical course, Fig. 7, instead of by a reversed flange. An incidental advantage of this is that the circulation of the water is a little freer.

DISHED HEADS FOR STEAM DRUMS

24 Several explosions have been caused by dished heads cracking around the edges where they join the spherical portions. Apparently

it has been thought by designers in general that if a head is pressed into spherical form and flanged, the radius being made equal to the diameter of the drum and the thickness equal to that of the drum, nothing more is necessary. There is no doubt, however, that such heads breathe and that the cracking is due to this. It is the same phenomenon that causes the rupture of lap joints and breaks staybolts. I believe that such heads should be made of thinner plates than usual and braced like flat heads. The braces should be strong enough to carry the total pressure on the heads, and thin plates would stand the flanging process better than thick ones and with less liability of being injured at the corners.

25 In regard to the method of bracing such heads I am inclined to think that radial plate gussets would be best, and, if placed at equal distances completely around the drum, the latter would not be distorted by supporting the head. I think that anybody who has a boiler with unbraced drum heads should view them with anxiety.

FLUSH HEAD HORIZONTAL RETURN TUBULAR BOILERS

26 In New England, horizontal return tubular boilers are always built with the front tube plate flanged forward, but in other parts of the country it is frequently if not usually flanged backward. The latter is known as the *flush head* or *New York* boiler. I consider the New England method the better because all the riveting can be done by machine. With the other construction, one circular joint must be riveted by hand or pneumatically. Another advantage of the New England method is that it makes a tight smokebox, while the other, especially if the boiler has a brick smokebox, which is usually the case, is likely to leak air. A leaky smokebox diminishes draft and cools the gases, and thus diminishes the effect of an economizer if one is used.

27 The object of the flush head boiler is to have the joint between the front head and the shell plate always in contact with water so that if it is not protected by brickwork it will not be injured by heat. In the New England design, there is, however, no difficulty in keeping the joint protected, and there probably has never been a case of burning the joint.

FACTORS OF SAFETY

28 At present it is customary in this country to use a factor of safety of five in the construction of stationary boilers; in locomotives the factor is four. I hope that by careful designing four will sometime be used in stationary boilers, for no boiler ever explodes because

it was originally too weak, but because, as already explained, it was designed or built so as to bend with subjection to, or changes in, pressure, except in the case of boilers having crown or stayed sheets. The explosions of locomotive boilers are almost exclusively confined to the blowing down of overheated crown sheets due to low water. In stayed water legs, it is extremely uncommon for enough staybolts to break to allow the sheets to bulge and rupture.

BRICKWORK

29 A desirable feature of brickwork enclosing boilers is the absence of cracks. This not only improves the appearance but, by preventing leakage eliminates the harmful effect on draft and economy and reduced efficiency of economizers when used. Generally speaking, brickwork will stand without cracks in boiler settings unless it is pushed by the boiler; and it is only necessary to so place it, and have such details about the boiler, that it will not receive any serious pushing. The brickwork should not touch the boiler anywhere, and the space between it and the boiler should be stuffed with asbestos fibre. Although this filling may tend to leak air, the covering over the top of the boiler, which rests on the brickwork, prevents this. The front end of the boiler should be fixed, and the other end should have attached to it some back connection covering device which will slide and not tend to push the back wall over. Such a device has been made by Orosco C. Woolson, Mem.Am.Soc.M.E., and is reproduced in Fig. 8.

30 The vertical thickness of the brickwork on the sides of the boiler, and almost in contact therewith, should not, I think, be more than 12 in. It has been made 24 in., and in such cases I have seen the part raised bodily, apparently by the expansion of the boiler. This part of the brickwork is usually above the center of the boiler, but it should be equally above and below the center as thereby there is less chance of any of its weight being supported at all by the boiler. If the boiler supports it there is a chance that its weight will bend the plates, and the arguments against this have already been given.

31 It is common to use buckstays on the sides of boiler settings, but I also use them on front and back in order to prevent cracks. Buckstays should be usually of 8-in. I-beams of the lightest section, instead of cast iron which is unreliable.

USE OF AIR SPACES IN BOILER BRICKWORK

32 It has always been customary in this country to build the side and back walls of boiler settings with air spaces in order to diminish

loss of heat by radiation. It is probable that these spaces cause loss of heat by convection and leakage, and it has been proved by experiments carried out by the United States Bureau of Mines (Bulletin 8) that this is true. If it is advisable to use spaces in the walls in order to prevent cracking of the brickwork, it is best to fill them with some material, such as ashes, crushed brick, sand or other loose material, which will entrap air but diminish its movement. Solid brick walls form a better non-conductor than walls with air spaces.

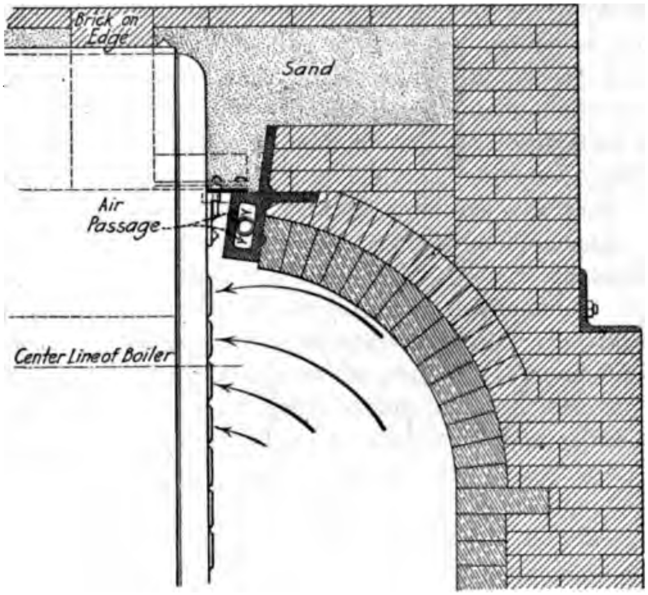


FIG. 8 WOOLSON'S GAS TIGHT BACK ARCH CONNECTION

METHOD OF SUPPORTING HORIZONTAL RETURN TUBULAR BOILERS

33 The method of supporting horizontal return tubular boilers is of more importance than is usually realized. Such boilers, no matter what their length or size, should be supported at no more than four points. If boilers are long it is common to support them at six points. In order to prevent the end supports from leaving their bearings, springs are often placed under the middle brackets, but this does not render the pressures on the supports equal and is only a makeshift.

34 It is a principle in mechanics that if a body rests on three points the pressures on these points can be determined and will not change even if the points change their positions or levels. A 3-legged stool always rests properly on its legs and with unchanging pressure

on each, even when it rests on an irregular floor, but a stool with than three legs never presses equally on each, and if its feet were fully fitted to bear on the floor, a change in position would destro fitting.

35 This shows that in supporting a horizontal boiler the t point principle should be applied. To obtain this effect and yet the boiler held up at four points, two points at one end are supp in the usual manner and the other two are connected by links equalizing lever working on a pin passing through overhead suppo

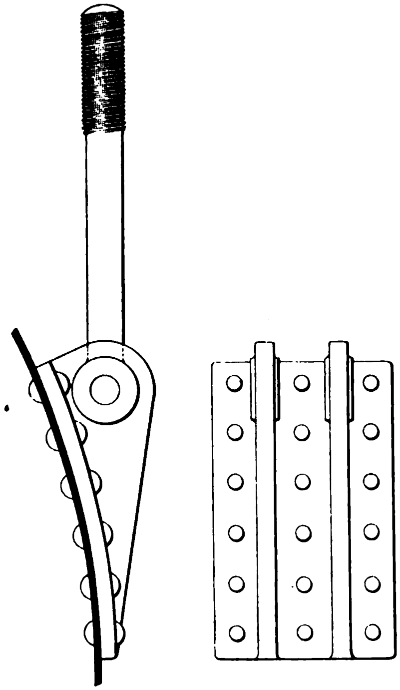


FIG. 9 BOILER SUPPORT, SHOWING GOOD DESIGN OF LUG

beams. This was first done by Mr. Woolson, as described in a by him before the Society[1] in 1898. When a boiler is supported i way the stress in the plates due to the weight can be made prope will never change no matter how much the brickwork may settle, this system is not used the stress at one point may be sufficiently to be a serious matter.

[1]Hanging and Setting of Horizontal Fire-Tube Boilers, O. C. Wc Trans. Am. Soc. M. E., Vol. 19, p. 781.

36 Brackets or lugs used on boilers for suspension from above should have the point of support close to the boiler and the direction of the link nearly or quite tangential to the shell, as shown in Fig. 9. With such a bracket and a tangential link, it is not important to rivet a plate inside the shell to distribute the supporting strains.

DISCUSSION

W. F. Kiesel, Jr. (written). The ills of boilers and some remedies, as described by the author, are worthy of considerable thought. The remedies proposed seem to indicate a general rule, that is, to distribute unavoidable distortions or deflections over as much space as possible, thereby avoiding concentration of strains in isolated locations, at which points, due to stresses approaching or exceeding the elastic limit, detailed fractures will occur.

The author has pointed out possibilities of change in design which tend to distribute the distortions. Distortions due to varying pressures can generally be eliminated in this manner. Changes in shape due to temperature must receive careful consideration by the designer, who usually estimates the possible extent of the distortion and utilizes the best available means to counteract the resultant strains within sufficient limits of stress per square inch. He also makes it a point to leave as much freedom as possible for the parts to expand and contract, realizing that such obstruction to expansion and contraction creates undesirable strains in the plates.

It is undesirable to introduce holes or stays unnecessarily for which reason the writer cannot agree with Mr. Dean in his recommendation to eliminate dished heads. It is true that some such heads have failed, but in connection with these failures there is chance for grave suspicion that either the metal in the head has been damaged, or that the design of the head was faulty. A well-made and properly-designed dished head will give no trouble, and thousands of them are in use. A dished head should have sufficient metal in the cylindrical flange to resist the pull of the spherical dish. Another important feature is the transition radius at the circumference of the flange.

The calculations as given by Rankine and others are not easy to handle by ordinary inspectors and laymen, but the engineer has no difficulty in properly designing a dished head. The formulation of rules for the use of inspectors must, necessarily, be simple. Such rules are, therefore, inclined to err on the safe side, and result in dished heads which are stronger than necessary.

For reasons given, we believe that, within prescribed limits, dished heads should have preference for small drums and boilers. This permits eliminating unnecessary stays and rivet holes, which stays at the same time oppose natural expansion and contraction due to heat and thereby create unknown internal stresses in the material.

Thomas E. Durban. There is one point in the paper I would like to bring out, and that is in reference to the recommendation that all holes be drilled through the solid. Our experience is that in drilling a number of plates, such as a horizontal boiler, with butt straps, it is impossible to drill them through the solid clear through and make as good a job as you can make if a pilot hole is punched, and the drill run through the pilot hole. As a matter of fact, in the case of a multitude of plates drilled together, it is impossible to run the drills through with the rapidity necessary in order to do the job quickly and not have the drill run.

The Author. Concerning Mr. Kiesel's view of the safety of dished heads, it is of course to be expected that mine would not be acceptable to everybody. I was not aware that the proper method of designing safe dished heads had been determined and hardly thought it likely to be. My view was therefore that it is best to have heads that are unquestionably safe.

In regard to Mr. Durban's view, I feel that his opinion would not be shared by all boiler makers, for there are some who make a practice of drilling rivet holes, and who, in fact, have no punches capable of punching holes through anything but plate about thick enough for boiler uptakes. I have always felt that the machines used for drilling holes in boiler plates are flimsy and unstable, and I think that in order to drill plates properly it is merely necessary to have machines that are of good stiff design. When such are used I feel certain that the difficulties mentioned by Mr. Durban will not exist.

No. 1501

APPLICATION OF ENGINEERING METHODS TO THE PROBLEMS OF THE EXECUTIVE, DIRECTOR AND TRUSTEE

By Hollis Godfrey, Philadelphia, Pa.
Member of the Society

The problem considered in this paper is the expression of an endeavor to determine the methods by which the service of a consulting engineer may be made of greatest value to the executive of a corporation, or to a board of directors or trustees, during an interregnum in the executive office. In taking a more or less comprehensive view of problems which have confronted certain executives, and mapping out plans coöperatively with them which they could present to their directors, to their subordinates, and to their constituents, I have been forced to a regular method of procedure made practicable through coöperation of the executive and the consulting engineer. This method, which, after an experience of several years, has proven its value in money and in service, is briefly as follows:

a The defining, on the basis of facts shown by engineering study, of the policies of a business.

b The expression in simple usable form of the definitions obtained.

c Construction on the basis of the studies made.

THE NEED FOR DEFINITION

2 The need for an executive to have definitions before he can make decisions is often forgotten. No executive is doing the most he can do for his business until he has obtained his definitions, by having his business fully surveyed and his fields of action selectively mapped, until he has located to the best of the present state of

Presented at the Annual Meeting, December, 1915, of The American Society of Mechanical Engineers. Contributed by the New York Section.

engineering and of industry the places where waste exists and where the greatest possibility for profitable development lies.

3 Such broad study, to be fully effective, must come from the coöperative use of the best methods known to engineering and to industry, through such coöperation between the industrial executive and the consulting engineer as shall combine the best of the inside industrial view of the business in question and the best of the outside view of applied science. Such a study must include or be followed by the vital but often neglected comparative and selective studies, which show what pays and what does not. To define what is to be kept and what dropped is essential to a successful business. Too often the rule is to keep everything; such a policy may have been possible in the old days of big profits, but it is no longer possible today.

THE NEED FOR EXPRESSION

4 To express a definition is almost, or quite, as necessary as to define, for an executive has especial reason to make his meaning clear. First and foremost, his policies must be so clear in his own mind that he can properly place them before the board of directors or trustees, if these bodies are to legislate wisely and well upon the policies placed before them. Again and again problems involving very considerable sums of money come before boards of trustees and directors, where the time of discussion of the question is limited to from one to three hours. The executive is the person to bring these problems before the board; and in order to choose wisely between different policies, in order to enable the directors to see what should be done, and to lay down a policy for the institution, the facts used must be so well chosen and selected that there can be no question of their value, and so clearly and so completely presented that their meaning can be given to a board in the shortest possible space of time.

5 Expression to the board of directors is but one of the problems. An executive has another duty which is often forgotten,—whether he will or no, his immediate task must be largely that of training his subordinates, of educating the men who are to work with him in the different policies of the business so that the whole staff shall work in harmony and each one understand what is behind the different policies which affect the sales, operating, and accounting departments. If the executive has not defined and expressed his *policies in so* simple a fashion that all his force is working in harmony,

the lost motion ensuing means a loss in money which is amazing to contemplate. In many corporations, the executive has another duty, because in this day of the interest of the public and of the Government in corporation affairs, it is again and again essential not only that proper facts should be chosen, but also that those proper facts, once chosen, should be properly translated and expressed in a form which will reach a given audience, especially the broad audience of the public. Too often today such selection and expression of facts are wanting, and instead a theory or a case is presented, which may be met and conquered by another theory or another case.

COÖPERATIVE CONSTRUCTION

6 Money saving and money making—time saving and improvement of service—all these are intimately concerned with proper definition and expression of the facts about a business. When such definition and expression have been made, strong constructive work can be done and all three factors, definition, expression, construction, can best be upbuilt through the application to industrial needs of the best that modern science and scientific methods has produced.

7 It cannot be too strongly emphasized that such applications of science to industry as are here outlined can best be accomplished through the coöperative efforts of engineers trained for years in science, in various types of industry and in the application of science to industry and of corporations willing to devote to such coöperative endeavor the cordial assistance of employes trained for years in the details of their own business. Neither from without nor within can the best results be secured. Advances can best be made through the welding together of the two types of knowledge possessed by two groups of men, one possessing general knowledge of science and industry and one possessing specific knowledge of a given business.

METHODS OF PROCEDURE

8 There are six divisions, as I have seen them develop, which very much concern a consulting engineer who is serving any corporation along these lines. The final thing is, of course, the decision, and it is the executive or the board who knows five or ten per cent more than the other man who decides promptly, and in most cases decides rightly. It is, then, a function of the consulting engineer to

present such facts to the executive that the basis of the decision made may be as sound as possible; he should place before the executive:

a A general preliminary coöperative study of the field to determine what lines are most worthy of study.

b A coöperative determination of what facts should be known about the lines selected and a definition of the periods most worthy of study on the basis of the preliminary study.

c A careful collection and intensive study of the facts existing in the lines chosen for the periods determined.

d The translation of the facts collected and studied in the light of their relation to the other departments of the business.

e The expression of the facts studied and the results obtained, in the briefest and clearest way.

f A method for the constructive use of the facts obtained and where necessary, an expression of the facts in a form intelligible to the special audience which needs most to know them.

THE TASKS OF EXECUTIVE AND ENGINEER

9 Turning to the first heading, the need for a broad, comprehensive view of the whole problem, it seems to me that every executive must realize the fact that it is his task to determine the internal policy and the external policy of his company, and that those policies, which make up the policy of the business, must rest primarily upon the history of that business and upon the facts of the business and its relation to its world. He should also realize that he can best obtain the necessary data as to both history and facts through the coöperative assistance of the engineer. No consulting engineer can possibly know as much of a given business as can men who have been trained for years in that business; but it is equally true, though sometimes forgotten, that no business executive can know as much of scientific methods as applied to industry as the engineer—each has his own knowledge, his own training, his own experience. One has the inside viewpoint; the other, the outside viewpoint. Neither can do a complete job alone. The peculiar value to industry of the engineer who possesses training and experience comes largely from the fact that such an engineer can choose the most vital physical facts and bring them into such relation that they may be clearly

and briefly expressed, and because the engineer's knowledge makes possible the discarding of useless statistics. The need for training and experience in the determining of what facts to collect and what lines to study can hardly be made too emphatic.

10 The problem, then, for the engineer is to use his viewpoint, to use the scientific method undimmed by too much technique of the business, coöperatively with the executive in the collection and translation of the facts, to bring the essential factors from the cloud of non-essentials, and to define and express the clear-cut, sharp, comprehensive policies which should be pursued.

11 The problem for the executive is to aid the engineer by giving him the facts about the business, and by assisting him in determining the practicability of his suggestions. It must be remembered also that both executive and engineer must be open minded, if coöperation is to succeed. It is comparatively seldom that an engineer is asked to lay out a wholly new plant. He is generally doing the best he can with an old one.

12 One may often see in the records of many corporations the rings of age showing as many periods of growth as the rings around a giant sequoia. To determine the meaning and value of these different rings, to understand their translation, to find out what is orderly growth and what is disease excrescence, to determine what should be fostered, to devise means of loosening cramping bands which have grown up around the business,—that task should be done best by the methods outlined here.

13 There is another reason why the obtaining of a broad comprehensive view is essentially a function of the outside engineer, even where the executive is himself an engineer. I personally believe that no man can get along without an outside view upon his problems, and that the executive is always dimmed by his nearness to the problem. I question whether any sincere man will deny that he has certain preferences for the work he is doing himself, or question that no matter how completely he endeavors to gain an unprejudiced view of his daily routine he is somewhat affected by a glamor of personality.

14 As I indicated before, the internal view is but one of the factors in the situation. No business lives alone. Not only within the business, but outside the business, any individual institution or corporation whose aims are clearly defined and rightly defended and expressed can remove to a large degree any unpleasant relations with other corporations, with legislatures, with public service com-

missions, and with the public generally. Such questions as the following are constantly arising: What is the competitor's policy as compared with ours? What is being done in his field, and how does his field relate to ours? What will a legislature or a public service commission do that affects our future? What will the public reaction be upon a given policy? All uncertainty as regards the answers to these questions is largely cleared away when any branch of industry has a clearly set forth and definite policy, clearly expressed, and based upon proper study of the facts not only of its interior but of its exterior relations.

A BROAD COMPREHENSIVE STUDY

15 An illustration of a rather broad and comprehensive study made for an executive and a board, the results of a year's study of a problem, which included a survey of the whole field and a selection of those lines which seemed most profitable for investigation, is stated as follows:

16 *Given a city with one hundred miles of streets and public places to be lighted and with a maximum population of a quarter of a million, provide the best light at the lowest cost.*

17 In this investigation we first made a preliminary study, and laid out a work chart, Fig. 1, showing certain statistical and experimental studies that we desired to follow out. Under *space* we studied the geographical areas, and made a classification of the streets,—widths, pavements, nature of travel, and character of abutting buildings, taken from the highway plans of various streets, and arranged in groups. We then classified the street intersections and public places, found the areas, angles of intersecting streets, character of abutting buildings, particularly with regard to automobile travel and possibilities of bad lighting at the intersections. We determined the present condition in the case of the streets and intersections and public places.

18 We then studied the densities of population, and also studied with relation to these densities of population, the movement of building in the last five years and how the building had progressed in different periods and under different conditions in the city. As a matter of fact, the popular estimates of what building had occurred were largely incorrect until we made this study, and the people of the city thought the facts about the city were quite different from what they actually were. We found that while the movement

PRELIMINARY WORK

- **A. Statistical Studies**
  - Space — Study of Geographical Area
    - Classification of Streets
      - Widths
      - Pavements
      - Nature of Travel
      - Character of Abutting Buildings
    - Classification of Street Intersections and of Public Places
      - Area
      - Angles of Intersecting Streets
      - Character of Abutting Buildings
    - Movement of Building Development
    - Present Conditions
  - Men — Study of Densities of Population
  - Money — Study of Unit Cost per Street Mile for Different Classes
  - Miscellaneous Facts — Study of Location of Existing Lamps
    - Personal Inspection
    - Official Records
    - Contractors' Records
- **B. Experimental Studies**
  - Areas — Illumination per Unit Area for Different Classes of Streets and of Street Intersections
  - Population — Traffic Census
  - Intensity of Light and Light Sources — Unit Intensity per Unit Population per Unit Area
- **Correlation of A and B** — The Second Step

Fig. 1 Chart Indicating Statistical and Experimental Studies Used in an Investigation of a City Lighting System building was supposed to be greatest in one direction, it was actually greater in another direction.

19 We then went on with the question of money, and studied the unit cost per street mile for the different classes of lighting for the different streets.

20 In regard to miscellaneous facts, we studied the location of existing lamps, from personal inspection, from official records, and from the contractors' records and comparing the city records with the contractors' records. The lighting had gone on as most lighting goes, in a varying manner; one year the appropriation had been large and there had been a lot of lamps put in as a result of new developments started. The next year the development died out, for some reason, and a lot of lamps had not been placed, and some of those which had been put in place were not effectively used. The result was that there was no regular distribution of lighting.

21 The heading "experimental studies" represents definite experimental work. We put up adjustable standards similar to those used by the Edison Company, the Electrical Testing Laboratories, and by Mr. Lacombe in the New York City tests, and also similar to various other experimental plants elsewhere. Under "areas," we studied the illumination per unit area for different classes of streets and of street intersections. We studied the population and traffic census, especially with reference to foot traffic and automobiles, and incidentally let me say that in a similar study to this we found one interesting thing where we made a very considerable mistake, one which would have made much trouble if we had not been working coöperatively with the officers of the city, and had not invited them to criticize freely our studies. We took the traffic census on a given road and found there were only eighteen vehicles going up that road. We checked it up, and one of the persons familiar with the situation said, "There is a big mistake there, because that is where the early morning trucking goes." We went back and made the study between two and six o'clock in the morning and found over a hundred trucks coming in over that road, which went out late in the afternoon. In the late morning, when the study was taken, there was almost no traffic on that street. There is an illustration of the necessity of coöperating with the people in the place, to find out what the facts are about the actual conditions. If we had not checked that study street by street with the people who knew the conditions, we should have based our lighting for

that road on the morning study, and found no traffic worthy of mention.

22 Under "intensity of light and light sources," we determined the unit intensity per unit population per unit area, and in getting that together there is only one conception, practically, that one

- SECOND STEP
  - DESIGN AND CONSTRUCTION OF TRIAL INSTALLATIONS
    - TYPE
      - GAS
      - ELECTRIC
    - REFLECTORS
      - CLEAR
      - SEMI-OPAQUE
      - TRANSLUCENT
    - POLE HEIGHT
      - LIMITS FOR ARCS
      - LIMITS FOR LOW UNIT
  - STUDIES OF RESULTS OBTAINED FROM TRIAL INSTALLATIONS
    - TESTS
      - C. P. OF LIGHT SOURCE
      - EFFICIENCY AT OBJECTIVE POINT
    - COSTS
      - RELATIVE MAINTENANCE COST OF VARIOUS TYPES
      - RELATIVE INSTALLATION COST OF VARIOUS TYPES

FIG. 2 STUDIES OF LIGHTING EQUIPMENT AND RESULTS FROM TRIAL INSTALLATIONS

should give the executive to use, and that is the conception of the unit, the concept of the candle, because after all our light unit comes back (with full recognition of the carefully drawn and admirable work of the Bureau of Standards), to a concept of a candle, just as density of population or unit population goes back to the conception of one man in one house, and that unit concept can always be given simply and directly. Almost the only thing used in the preliminary work, which needed any translation, was the word "unit," (and we only used that because we had to use the unit concept) and we found that was easily translated, even with the men on the street.

23 We then took up the question of design and construction of trial installations for both gas and electric lamps, Fig. 2. In the case of reflectors, we studied the clear, semi-opaque, and translucent. We also studied the matter of pole heights, limits for arcs, and limits

- ANSWER TO PROBLEM
  - THE DRAWING OF SPECIFICATIONS
    - INTERCHANGEABLE WHERE POSSIBLE
      - GAS
      - ELECTRIC
    - CONSISTENT WITH LOCAL CONDITIONS
    - COMPREHENSIVE
  - THE OBTAINING OF BIDS
  - THE FINAL CONTRACT
    - FAIR AND EQUITABLE TO CITY AND CONTRACTOR

FIG. 3 CHART INDICATING ANSWER TO THE PROBLEM OF CITY LIGHTING

for low units. We made studies of results obtained from trial installations, and made tests to discover the candlepower of light sources and their efficiency at given points.

24 In considering the cost of this installation, we then proceeded to a study of the relative maintenance and costs of various types.

25 The answer to the problem was the final drawing up of specifications, Fig. 3, making the gas and electric units interchangeable where possible, and making specifications consistent with the local conditions which did not require tests of an expensive nature

to be carried on, and which would be comprehensive. It is impossible in any chart to show all of the different factors which concern specifications in general. Perhaps there is no more dangerous fallacy than the concept that a specification which can be used with great satisfaction in one plan can be used with equal satisfaction in another. The main thing which we desired to obtain was a final contract which should be fair and equitable to both the city and the contractor.

26 Such a study as I have shown here is a brief representation of my first concept which concerned the possibilities of engineering work done coöperatively with an executive in obtaining broad, comprehensive views for the education of associates, for the public, and for the authorities.

A SELECTIVE STUDY

27 The second division of my subject concerns the application of the theory of selection to the different parts of a problem in such a way as to enable any executive to understand those factors in a business which are more valuable and those which are less valuable. Certainly if there is a branch of engineering which, from the standpoint of money returns, needs most to be applied to industry, it is such studies as shall give readily to the executive and to legislative bodies and corporations that concept of the selective values of their business. There is no way I know by which this can be obtained save through engineering studies of the type illustrated here,—for there are very few businesses which are so simplified that one can tell offhand at a glance what line will pay and what line will not. Such studies as those concerned with the possibilities of obtaining raw material, of development of transportation, of the labor market, and of selling fields,—all come into this group and concern such comparative information as shall make selection between different problems possible. Such comparative information as will enable the executive to know which part of the business will be mainly effective, provides the means by which the executive may select what shall be pushed and what shall be held, what shall be dropped, and what shall be left for a future time or for future stimulation. It is upon such selective decisions that again and again large decisions may hinge.

28 I had made a number of studies of this nature when I went to the Drexel Institute, Philadelphia, as president, and found there

what is right to keep and what is right to eliminate. The educational example used here is chosen as an illustration.

33 I have already spoken of the question of the time-saving factor in the preparation of reports. The average director or trustee is a busy man. The meetings of the boards of directors or trustees which I have attended in different parts of the country have almost always been brief, a condition which calls for such rapidity and such pointedness of statement, as shall bring conviction to the eye as well as to the ear, and will make possible wise legislative action upon the

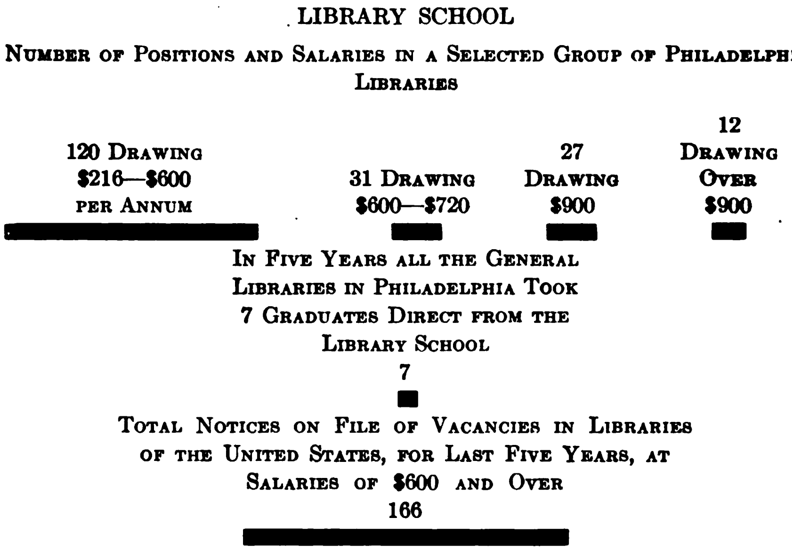


Fig. 7 A Study of Library Positions and Salaries

part of the board. If the proper ends are to be obtained, clarity as well as brevity is essential.

34 Since 1906, I have been largely interested in the study of engineering reports from the standpoint of clearness. The inadequacy, diffuseness, and obscurity of many of these reports is remarkable. Reports to be read by laymen are either couched in technical language difficult of comprehension, or so much material is laid before the public that they are commonly unable to come to the final, sharp, clear understanding of what is meant, and such lack of expression is a great handicap to our profession.

A STUDY IN CLARITY

35 Fig. 8 shows the summing up of the results accomplished through a study of the type outlined above and through construction on the basis of the study, and illustrates the principle of clarity which it is desired to show.

36 The results concern the regular delivery of thousands of papers called for daily on the requisitions of clerks. No mention is made of the emergency system of delivery which was constructed

THE RESULT IN SERVICE

Average Time Required to Fill a Given Number of Requisitions

Delivery Time

The Old Way

5 Hours

The New Way

3 Hours

Regularity of Delivery

Irregular Number of Deliveries by Clerks from Various Departments or Messengers, Resulting in Waste Time by Clerks of 65 Hours per Week

Replaced by

Regular Deliveries by 2 Messenger Boys at Frequent Intervals

Saving of Time of Department Clerks

+65 Hours Used by Department Clerks

2 Boys

2 Boys

Completeness of Delivery

Incomplete Deliveries

Replaced by

Complete Deliveries All Requisitions Sent to Sections at One Time Are Filled or Returned Checked at One Time

Fig. 8 One of a Group of Charts Expressing Results from a Study of a Large File Room Chart

in order to care for all such requisitions. The problem here so far as the regular delivery from the file room is concerned resolved itself into four phases:

a Diminishing of the time required to fill requisitions. The work done reduced the time in question from five to three hours according to the records made.

b Regularity of delivery. The installation of a controlled system of transportation replaced irregularity of delivery with controlled and inspected regularity.

c Reduction of irregular messenger service carried on by department clerks. This irregular service was abolished and the whole messenger service carried on by the two boys who had previously done but a part.

d Completeness of delivery. This was provided and assured by a complete system of inspection. The entire differences in service noted here were accomplished without increase of working force but actually with a decrease.

A STUDY IN BREVITY

37 Brevity is also essential in an engineering report. It is necessary to attack the problem in such a way as to give definite and complete information to a limited, definite and complete group. The engineering report is a translation of physical facts to be used in many cases for the information of groups. The group of people for whom that report is written should be considered quite as much in an engineering report as in any other translation of physical facts on paper.

38 To come to a specific case in Fig. 9, I have constantly found that the question of prices was one of the most difficult things to go into in a short space. I have spent a considerable amount of time in endeavoring to get all price facts about several businesses on a single sheet of paper. The facts were on 34 pages of catalogue when I went to the Drexel Institute, and one would have to read over 34 pages of this catalogue to get all the facts about the different fees and different items of cost information one would naturally desire. To solve this problem we standardized on fees, and put them all into a single place, and for the sake of clarity stated in all cases three things: the amount, what the student obtained for the amount, and when it was payable. These are the three questions that are always asked by prospective students and their parents.

REQUIREMENTS OF AN ENGINEER WORKING IN THIS FIELD

39 There is no question that there is some opposition to any question of engineering study, or especially of any engineering investigation. Most of the present justified feeling against the very word *investigation* has come because of the defects of the investigator. *Much of that* opposition has come because it was assumed that the

A STUDY IN CLARITY

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36 The results concern the regular delivery of thousands of papers called for daily on the requisitions of clerks. No mention is made of the emergency system of delivery which was constructed

THE RESULT IN SERVICE

AVERAGE TIME REQUIRED TO FILL A GIVEN NUMBER OF REQUISITIONS
DELIVERY TIME

THE OLD WAY

5 HOURS

THE NEW WAY

3 HOURS

REGULARITY OF DELIVERY
IRREGULAR NUMBER OF DELIVERIES BY CLERKS FROM VARIOUS DEPARTMENTS OR MESSENGERS, RESULTING IN WASTE TIME BY CLERKS OF 65 HOURS PER WEEK

REPLACED BY
REGULAR DELIVERIES BY 2 MESSENGER BOYS AT FREQUENT INTERVALS

SAVING OF TIME OF DEPARTMENT CLERKS

+65 HOURS USED BY DEPARTMENT CLERKS

2 BOYS

2 BOYS

COMPLETENESS OF DELIVERY
INCOMPLETE DELIVERIES

REPLACED BY
COMPLETE DELIVERIES ALL REQUISITIONS SENT TO SECTIONS AT ONE TIME ARE FILLED OR RETURNED CHECKED AT ONE TIME

FIG. 8 ONE OF A GROUP OF CHARTS EXPRESSING RESULTS FROM A STUDY OF A LARGE FILE ROOM CHART

in order to care for all such requisitions. The problem here so far as the regular delivery from the file room is concerned resolved itself into four phases:

a Diminishing of the time required to fill requisitions. The work done reduced the time in question from five to three hours according to the records made.

b Regularity of delivery. The installation of a controlled for policies is of value only as it is vitally concerned with the future.

b To omit many engineering refinements that cost much money and lead to a "false and delusive accuracy," as one of my old instructors used to put it; to avoid, so far as possible, in all cases doing work that "costs more than it is worth."

c To check his work with the practical needs and limitations of the business, keeping an open and perceptive mind.

d To change his methods for the better with the advance of the engineering and industrial art—in short to keep abreast of the "best of the art."

e To have a wide acquaintance with those men of his profession and of other professions and callings, who know most of special lines of which he may need to know.

f To bring the best that science and industry has yet produced to bear upon the problem before him.

40 Those are serious requirements to lay down. Yet there is no small number of engineers who I know could meet them all. They can hardly be made materially less, for in work like this the danger of construction turning to destruction in untrained hands, presenting ill digested, wrongly collected, wrongly translated, and wrongly expressed facts, is too great to be readily assumed.

FACTS VERSUS THEORY

41 There is another matter which affects very considerably, as I said at first, any problem in engineering investigation, and any problem of expression of engineering facts, and that is the fact that people today are looking, it seems to me, for facts properly expressed rather than for legal cases. Again and again in this country we have suffered from one case placed against another, one theory placed against another theory; and I believe that the period is coming more and more when the facts are going to settle problems; but facts alone are not enough. It is only when facts are expressed in such a fashion as to reach the definite audience which it is to reach that facts really gain the day.

42 On this whole question, it would seem to me that it is of vital importance to the profession to show the industries and institutions of this country the value of such engineering service as I have outlined here, and perhaps also the need to have engineers with trai

engineer was working to investigate or find fault with somebody, instead of working coöperatively with the men in the business to do the most that could be done for the business. Moreover, to make savings of money, service, and time, the coöperative engineering adviser to an executive must be able—

FEES AND STUDENT EXPENSES OF THE DREXEL INSTITUTE

TUITION	$100.	RECEIVED IN INSTRUCTION DURING YEAR

PAID TO REGISTRAR OF THE INSTITUTE

$50. Paid on 1st registration day.	$50. Paid on 2nd registration day.

INSTITUTE FEES Paid to Registrar	STUDENT EXPENSES Paid elsewhere than to the Institute
SUPPLIES AND LABORATORY EQUIPMENT. $20. Returned in experience with visible materials used in class during year. Paid to Registrar first registration day.	BOOKS AND SUPPLIES. $25. Received in visible property largely left at end of year. Expenditure made by student at various times during year.
STUDENT ACTIVITIES. $10 Returned in better health and social experience during year. Paid to Registrar first registration day.	MATERIAL FOR APPAREL Amount largely dependent on student. Received in visible property. Expenditure made by student at various times during the year.
DEPOSIT $5. Returned in cash if nothing is broken or lost. Paid to Registrar first registration day and covers keys and breakage.	INCIDENTAL EXPENSES, LUNCHES, CAR FARE, ETC. Amount largely dependent on student. Expenditure made by student.

FIG. 9 CHART INDICATING METHOD OF BRIEF AND COMPREHENSIVE GROUPING OF COST DATA

a To distinguish clearly between records which are vital to the future policies of a business and those which are merely historical. The past in industry as a determinant

No. 1502

OPERATION OF PARALLEL AND RADIAL AXLES OF A LOCOMOTIVE BY A SINGLE SET OF CYLINDERS

By Anatole Mallet, Paris, France
Honorary Member of the Society

In locomotive construction, it is often necessary to introduce, at one or both ends, carrying axles with radial play, in order to facilitate the handling of curves. Such construction reduces, however, the amount of adhesive weight, which is a serious inconvenience if considerable play is required. Hence, for a long time, arrangements have been sought which would make it possible to transmit power to convergent axles without, however, increasing the number of steam cylinders. An examination of the principal arrangements proposed for this purpose forms the subject of the present paper. These various systems of transmission may be divided into two classes: *First,* those which involve elements having rotary motion; *second,* those which involve elements having reciprocating motion.

TRANSMISSION BY ROTARY MOTION

2 This class includes gear transmission, transmission by endless chain and transmission by universal joints.

3 *Gear Transmissions.* Although the first use of gear transmissions dates back to the very origin of locomotives,[1] they appear to have been first utilized for operating locomotive axles having freedom of radial movement in 1838, in a locomotive built at Heath Abbey for the Rhymney foundry in Wales. This locomotive was carried on two trucks with two axles each, as shown in Fig. 1, which is reproduced from The Engineer, November 15, 1867. The two trucks could turn

[1]Trevethick, 1803.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Railroads.

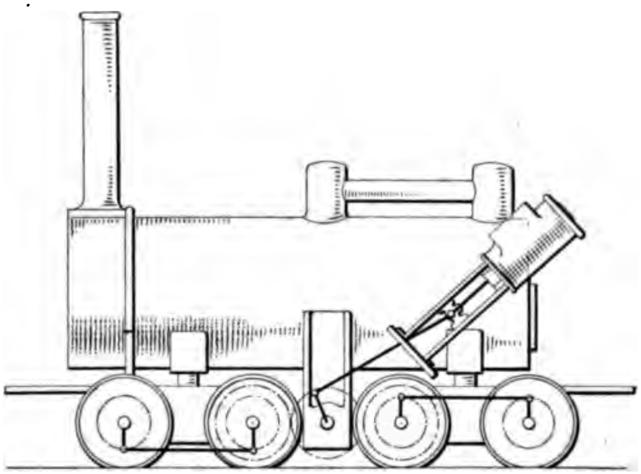

Fig. 1 Gear Transmission Locomotive

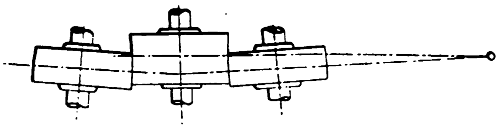

Fig. 2 Plan of Gears of Locomotive in Fig. 1

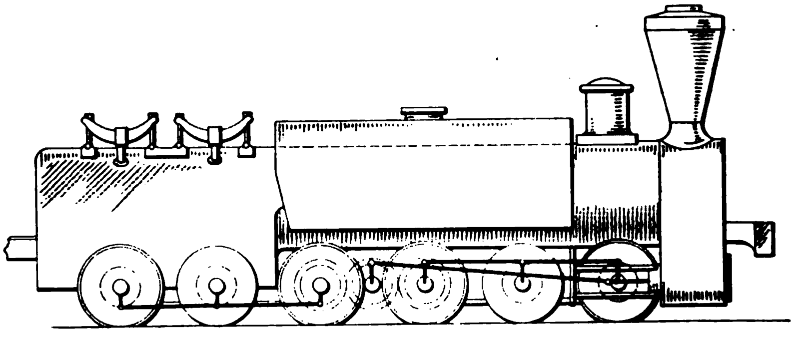

Fig. 3 Six-Axle Gear Transmission Locomotive by Tourasse

so that they were at an angle with each other without throwing the driving gears out of mesh, as illustrated in Fig. 2.

4 In 1841, the Baldwin Locomotive Works[1] built a locomotive in which the rear axles were driven by means of a countershaft and connecting rods, and the axles of the front truck were operated by a gear transmission located on the longitudinal axis of the machine. This locomotive weighed 13½ tons and was designed for use on a quarry railroad. The results, I am told, were satisfactory, but the type was afterwards abandoned.

5 The French engineer, Tourasse, presented at the Competition

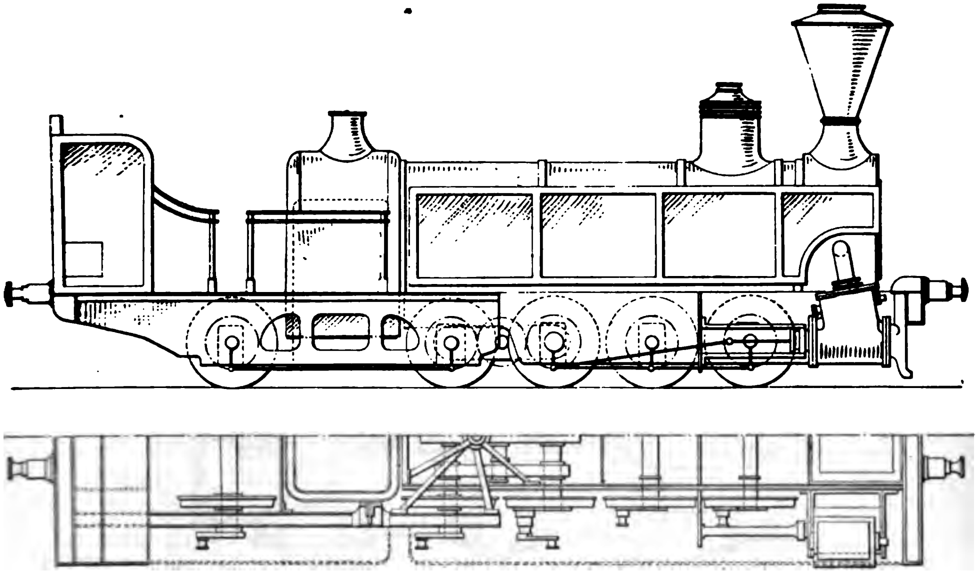

FIG. 4 ORIGINAL ENGERTH LOCOMOTIVE

of Semmering in 1851 a design of locomotive with six axles (Fig. 3) and similar to the Rhymney locomotive. It may be seen that from the cylinders is operated a countershaft carrying a toothed gear which engages with toothed wheels carried on the nearest axle of each track, these axles in turn being coupled to the other axles by outside connecting rods. This locomotive was to weigh 60 tons with the water carried in a saddle tank on the boiler. The cylinders were 0.50 m. (1.64 ft.) in diameter with a 0.60 m. (1.97 ft.) stroke; the wheels were 1.20 m. (3.93 ft.) in diameter and the heating surface, 250

[1]Notice sur les locomotives des établissements Baldwin, p. 19, édition de 1881.

sq. m. (2590 sq. ft.). The power developed would have been extraordinarily large for that time, since, according to the author of the design, the locomotive was to be able to start with a load of 250 tons over a grade of 2½ per cent, although a capacity of only 140 tons was required. Fig. 3 is reproduced from the Atlas zu die Lokomotive des Staats Eisenbahn über den Semmering; pl. XXVII.

6 The Locomotive Works of Winterthur, Switzerland, built in 1883, for an industrial railroad in the south of France, a locomotive

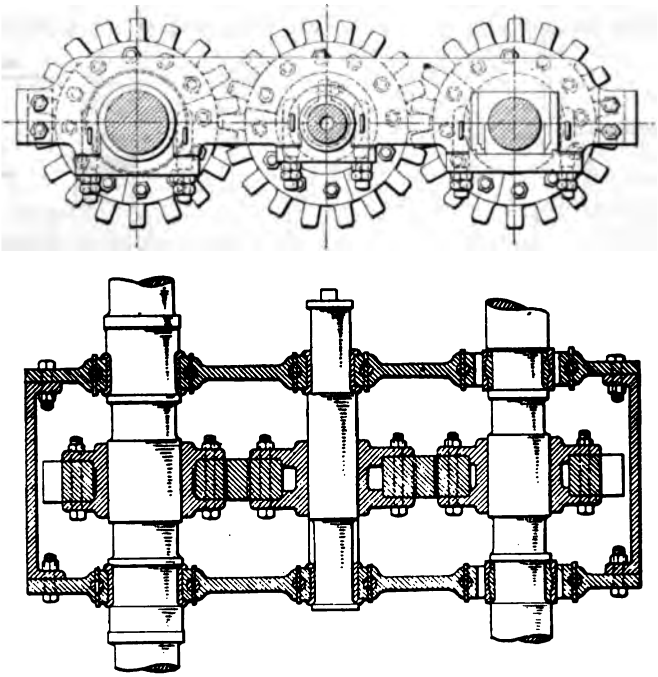

FIG. 5 GEAR ARRANGEMENT OF LOCOMOTIVE IN FIG. 4

similar to the one just described. It was supported on two trucks of two axles each, and weighed 22 tons. It is described in the Organ für die Fortschritte des Eisenbahnwesens, 1883, p. 4, pl. I. It appears that this type was unsuccessful.

7 The famous Engerth locomotive (Fig. 4), built after the Semmering Competition from which no practical results were obtained, was at first characterized by the use of gear transmission for connecting the last axle of the locomotive to the forward axle of the tender. The arrangement of these gears is shown in Fig. 5. It may

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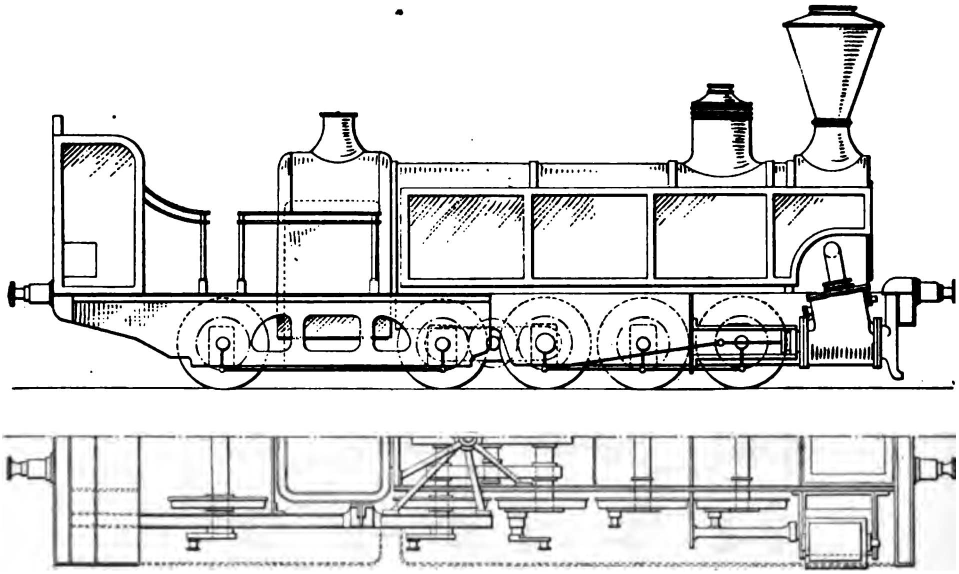

Fig. 4 Original Engerth Locomotive

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[1]Notice sur les locomotives des établissements Baldwin, p. 19, édition de 1881.

the Climax and Heisler locomotives have actually been built in sizes of 75 to 80 tons and the Shay locomotives, up to 135 tons.

12 *Transmission by Endless Chain.* The use of endless chain for coupling axles which may be thrown out of parallelism appears to have been adopted for the first time in 1851, by S. A. Maffei, of Munich, in the construction of the locomotive Bavaria, presented by him at the Semmering Competition. This machine (Fig. 6) had seven axles, driven by two cylinders. The axles were divided into three groups and the wheels of each group were coupled by external connecting rods, while the groups were connected by endless chains made of links and studs. The engine weighed 68,000 kg. (74.8 tons) with its equipment and it was given the first prize at the Competition after having satisfactorily passed all the tests, having shown a very remarkable regularity with respect to speed, pressure and combustion

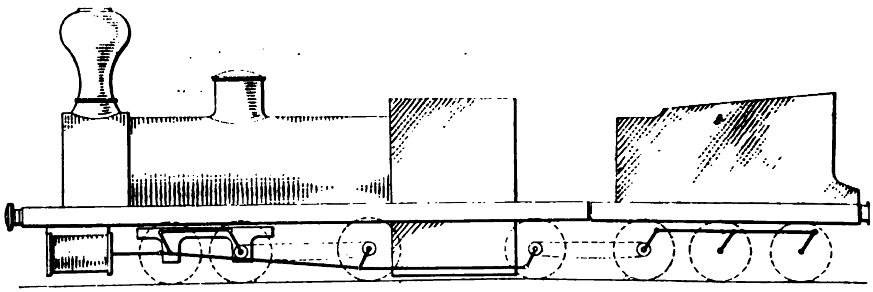

FIG. 6 LOCOMOTIVE "BAVARIA" BY MAFFEI

of fuel, as well as exhibited a constant superiority in regard to fuel economy. It was said that the victory was due only to the very brief duration of the tests, and that this locomotive could be maintained in good operating condition only by constant repairs to the chain transmissions. As a matter of fact, the Bavaria has never been reproduced in full or in part.

13 The designer of this machine, Maffei, presented to the same Competition several designs based on the use of chains, and other engineers have considered this method. Lievescy, for instance, in 1860, devised an arrangement of chains and sprockets, not mounted directly on the axles as in the machine of S. Maffei, but carried on a spherical ball joint in such a manner that the toothed wheels and chains remain always in the same plane, even when the axles vary in position with respect to each other. Thouvenot proposed to connect the various axles of a train by means of elements of this kind, but

be noticed that the intermediate shaft, carrying the middle toothed gear, is arranged to slide longitudinally in its bearings if necessary, to cut out the connection with the wheels of the tender.

8 Quite a large number of Engerth locomotives were built. As a rule, they had six driving wheels under the locomotive proper and four under the tender, or a total of ten in all. As the gears did not give satisfactory results in actual practice, however, they were eliminated and the machine reduced to the type of locomotive and tender with six coupled wheels. The complicated gear transmission type has long since entirely disappeared from practice.

9 Within recent years, a locomotive builder in Lyons has built some small narrow gage locomotives which are supported on four axles—all driving. The three rear axles are coupled by external connecting rods, while the front axle, which has radial freedom of motion, is connected with the axle next to it by a train of gear wheels located in the longitudinal axis of the machine, just as in the Engerth type. It does not appear, however, that this system has found an extensive application.

10 Before 1830, W. N. James, of Birmingham, proposed to connect not only the axles of the locomotive and tender but also those of the cars by means of gear wheels operated by a longitudinal shaft running the length of the train and provided with ball and socket joints to give them the flexibility necessary for making the curves. By this arrangement the inventor proposed to obtain sufficient adhesion to handle the train on grades without recourse to the Blenkinsop rack, and he stated that experiments made on a small scale showed that he could make grades of three inches in a yard, or 1 in 12. In the History and Progress of the Steam Locomotive, by Elijah Galloway, London, 1831, pp. 587 and following, is found a description of the arrangement proposed by James; this has furthermore been proposed several times later, for instance, by Cernurchi in 1857, for mountain railroads.

11 There are, in actual use in the United States, locomotives in which the axles of the engine and of the tender are coupled together by gears and a longitudinal shaft fitted with ball sockets. These locomotives are of the Climax, Shay and Heisler systems, which differ from one another in the arrangement of the details of transmission and the location of the steam cylinders. These systems are too well known to necessitate their description here, but it may be of interest to state that they have been used even for very large units. Thus,

the Climax and Heisler locomotives have actually been built in sizes of 75 to 80 tons and the Shay locomotives, up to 135 tons.

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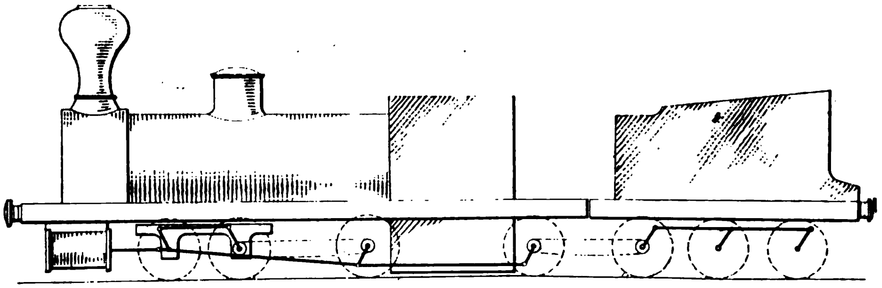

FIG. 6 LOCOMOTIVE "BAVARIA" BY MAFFEI

of fuel, as well as exhibited a constant superiority in regard to fuel economy. It was said that the victory was due only to the very brief duration of the tests, and that this locomotive could be maintained in good operating condition only by constant repairs to the chain transmissions. As a matter of fact, the Bavaria has never been reproduced in full or in part.

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shown in Fig. 8 (only one-half of the machine is shown, the other half being exactly similar). From the figure it may be seen that only the central carrying axle of each truck is hollow, containing a rigid shaft *a* acted upon by the steam cylinders. The other axles are coupled by external connecting rods, in the middle of each of which is provided a slot in which glides the crank pin of the shaft *a*. The shaft *a* is carried on external supports and is connected with the hollow axle by a central universal joint.

18 The inventor utilized the peculiar idea of operating both trucks from the same cylinders in order to simplify the general construction of the machine. To accomplish this, each piston rod was arranged to pass through both covers of its cylinder and to engage at each end with a connecting rod. Due to the obliquity of these connecting rods, however, their midpoints of stroke did not correspond to each other, and a sliding of the wheels on the rails twice in each revolution resulted. This difficulty could have been avoided by the use of two pistons in each cylinder, one for the forward truck and the other for the rear truck, but that would have complicated the machine.

19 In 1892, Messrs. Klein and Lindner, Saxonian engineers, invented an arrangement of an axle with ball joint which they used on a certain number of locomotives, either on the front axle or on the rear axle. This arrangement is shown in Fig. 9. It does not appear to the author that this system has been used on any but locomotives of quite moderate weight.

TRANSMISSION BY RECIPROCATING MOTION

20 This class includes such systems as make use of connecting rods, equalizers, etc., for connection of the convergent axles. The author considers it advisable to call attention in a general manner, however, to the fact that this classification cannot be very rigorous because certain arrangements might belong to two classes at the same time, owing to the multiplicity of parts entering into their construction.

21 The mechanisms of this class may be divided in the following manner: Coupling of convergent axles by connecting rods located in the longitudinal axis of the engine, these connecting rods being either simple or double, rectilinear or triangular; coupling by oscillating levers or equalizers; use of a free axle coupled by connecting rods to the converging axles, and coupling of axles by means of external connecting rods of which the length varies with the radial displacement of the axles.

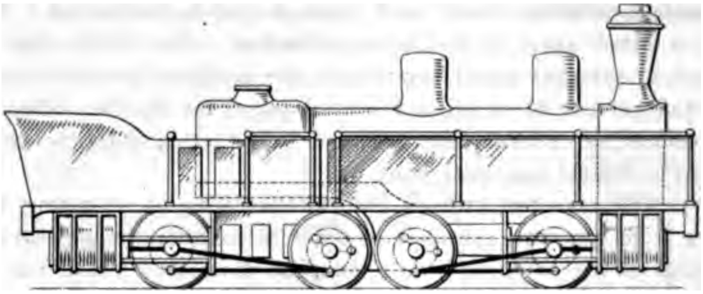

FIG. 10 SOCIETE JOHN COKERILL DESIGN

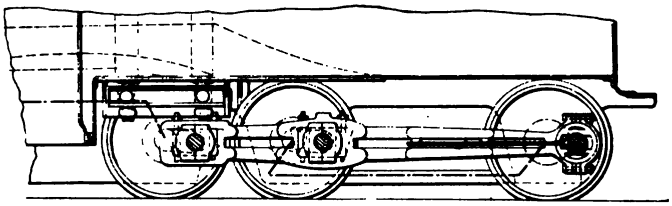

FIG. 11 LONGITUDINAL SECTION OF FIG. 10

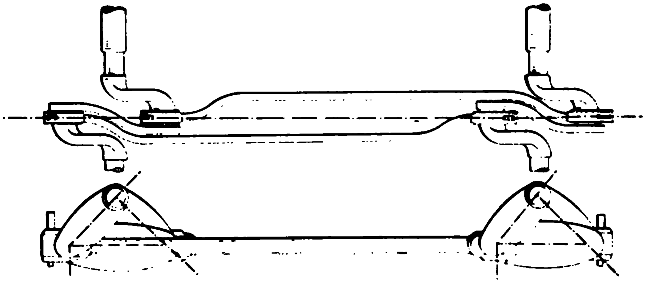

FIG. 12 THOUVENOT'S LOCOMOTIVE WITH RODS ON AXIS

shown in Fig. 8 (only one-half of the machine is shown, the other half being exactly similar). From the figure it may be seen that only the central carrying axle of each truck is hollow, containing a rigid shaft *a* acted upon by the steam cylinders. The other axles are coupled by external connecting rods, in the middle of each of which is provided a slot in which glides the crank pin of the shaft *a*. The shaft *a* is carried on external supports and is connected with the hollow axle by a central universal joint.

18 The inventor utilized the peculiar idea of operating both trucks from the same cylinders in order to simplify the general construction of the machine. To accomplish this, each piston rod was arranged to pass through both covers of its cylinder and to engage at each end with a connecting rod. Due to the obliquity of these connecting rods, however, their midpoints of stroke did not correspond to each other, and a sliding of the wheels on the rails twice in each revolution resulted. This difficulty could have been avoided by the use of two pistons in each cylinder, one for the forward truck and the other for the rear truck, but that would have complicated the machine.

19 In 1892, Messrs. Klein and Lindner, Saxonian engineers, invented an arrangement of an axle with ball joint which they used on a certain number of locomotives, either on the front axle or on the rear axle. This arrangement is shown in Fig. 9. It does not appear to the author that this system has been used on any but locomotives of quite moderate weight.

TRANSMISSION BY RECIPROCATING MOTION

20 This class includes such systems as make use of connecting rods, equalizers, etc., for connection of the convergent axles. The author considers it advisable to call attention in a general manner, however, to the fact that this classification cannot be very rigorous because certain arrangements might belong to two classes at the same time, owing to the multiplicity of parts entering into their construction.

21 The mechanisms of this class may be divided in the following manner: Coupling of convergent axles by connecting rods located in the longitudinal axis of the engine, these connecting rods being either simple or double, rectilinear or triangular; coupling by oscillating levers or equalizers; use of a free axle coupled by connecting rods to the converging axles, and coupling of axles by means of external connecting rods of which the length varies with the radial displacement of the axles.

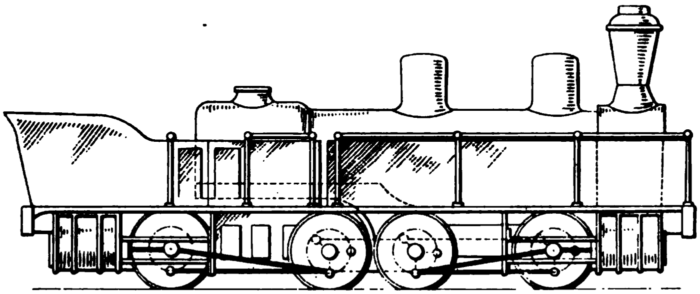

Fig. 10 Societe John Cokerill Design

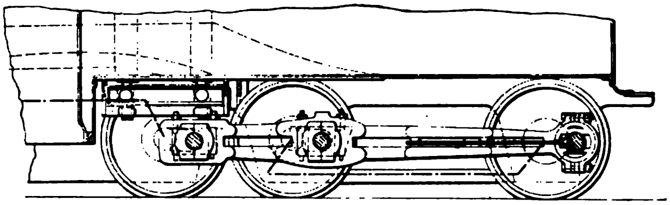

Fig. 11 Longitudinal Section of Fig. 10

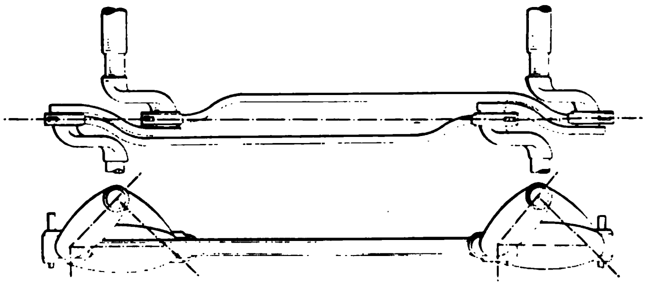

Fig. 12 Thouvenot's Locomotive with Rods on Axis

systems located very close to the longitudinal axis, with cranks at right angles to the axles.

28 *Coupling by Oscillating Levers or Equalizers.* The idea of using oscillating levers for coupling convergent axles was first disclosed about 1855 in an invention by Lucien Rarchaert, who tried very persistently to realize it. Fig. 15 shows the arrangement proposed by this inventor for coupling three axles. This system has been very favorably reported on.[1] There is a model of it in the gallery of the Conservatoire des Arts et Métiers in Paris, but the system itself has never been actually used. Larpent, engineer of the Western Railroad, of Paris, offered a design very much like the preceding one but had no more success.[2]

29 A German engineer, Christian Hagans, of Erfurt, invented an arrangement which he applied at first to small locomotives and after, with some modifications, to large five-axle locomotives with three fixed axles and two axles forming a truck, as shown in Fig. 16. The axles of the truck were acted upon by a vertical lever, *a,* oscillated through the intermediary of a longitudinal rod by lever *a'*, oscillated by the piston rod. The upper end of the lever *a* is connected to the top of an equalizer, *b,* articulated in the middle and having its lower extremity attached by a distance rod to the rear axle, *c,* of the truck. The result of this arrangement is that if, on curves, the axles of the truck are displaced, the lower part of the lever *a* has a displacement in the same direction and to the same amount, so taking care of the convergence of the axles.

30 The Hagans system was at first considered quite a success on the Prussian State Railroads on five-axle coupled locomotives weighing 72 tons in service, but it has since been entirely abandoned. As a reason for this was given the fact that the introduction of locomotives with five axles, parallel and coupled by ordinary side rods, has made the complication and expensive maintenance of the Hagans machine unnecessary. However this may be, we may say that this system has probably been supplanted by the Engerth system, no doubt one of the most widely applied arrangements for operating converging axles. A machine of this type was shown at the World's Exposition in Paris in 1900.

31 The Johnstone system, which has been applied on several large duplex locomotives built in the United States for the Central

[1] Annales des Mines, 1863.

[2] Bul. French Society of Civil Engineers, 1860.

Mexican Railroads, has also some resemblance to the preceding type. Fig. 17 shows one-half of this locomotive, and the other half is entirely similar. The piston rod (or rather rods since there are three of them, there being two cylinders placed side by side) acts on the middle of a lever *a*, which is vertical when in its normal position. The main connecting rod is attached to the lower extremity of this lever while from the upper extremity a short coupling rod connects to the top of equalizer *b*. This equalizer oscillates about its middle and operates from its lower end a connecting rod to a crank pin set at 180 deg. from the working pin of the counter-crank. From the figure it is seen that the lever *a*, to which the piston rods are attached, moves

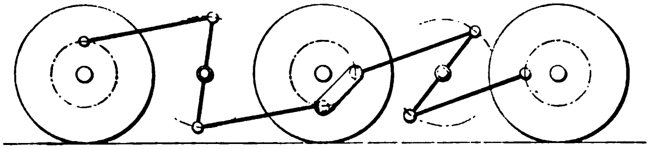

FIG. 15 BARCHAERT'S THREE-AXLE COUPLING ARRANGEMENT

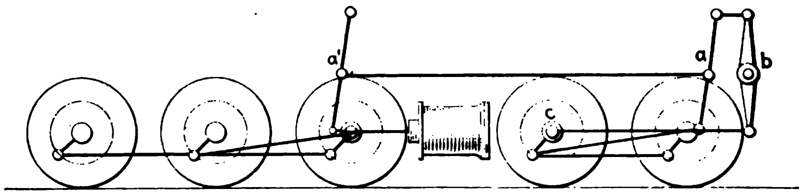

FIG. 16 HAGAN'S FIVE-AXLE LOCOMOTIVE

always parallel to itself, vertically on straight track and at a slight incline on curves.

32 In the same category may be placed also the arrangement proposed in 1860 by Behne, of Hamburg, of a locomotive set on two trucks of four axles each and having two steam cylinders, one at each side, operating through equalizers as shown in Fig. 18. These equalizers act through connecting rods to the cranks of countershafts which are coupled with cranks on the axles of the wheels. This locomotive was described in French Patent No. 26977 of November 8, 1860, and could have been used up to a weight of 100 tons with a load per axle of about 12 tons, as was usual in those days.

33 *Use of Free Axles.* The use of a free axle coupled by connecting rods with radial axles appears to date back to the Semmering Competition. Maffei there presented several designs in which the

systems located very close to the longitudinal axis, with cranks at right angles to the axles.

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[1]Annales des Mines, 1863.

[2]Bul. French Society of Civil Engineers, 1860.

axles of locomotives and their tenders were coupled by inclined or triangular connecting rods. A similar design (Fig. 19), submitted at the same Competition by a Hannoverian engineer, Kirchweger (Atlas zu die Lokomotive des Staats-Eisenbahn über den Semmering; pl. XVIII), shows a locomotive carried on two trucks having two axles each, the coupling of the trucks being effected by an arrangement of this kind. It may be seen that there is a connection between the journal boxes of the wheel axles and of the free axle.

34 The Austrian engineer, Pius Fink, tried to retain in the Engerth machine its original property of total adhesive weight by substituting for the gear train an articulated device. He built three locomotives,[1] of which one, the Steierdorf, was shown at the Universal Exposition in London in 1862 and in Paris in 1867. These locomotives were in service for several years, but are not built now. Fig. 20 shows the transmission of this locomotive, by which the first axle of the tender (carried on the truck) is connected to the free axle of the locomotive proper. The locomotive had three axles and the tender two, as in the Engerth machine. Dredge and Stein in England[2] resorted to the use of a free axle and a central triangular connecting rod for coupling the axles of a locomotive with those of the tender.

35 The inventor, Rarchaert, to whom reference was made above, after having abandoned the system of oscillating levers, designed an arrangement coming under the present category (Fig. 21). This was applied on a locomotive with two trucks having two axles each. The cylinders operated a free axle coupled with the carrying axles by a triangular central connecting rod. The pins of the cranks had spherical heads. The locomotive was in service on the railroad between Fougérès and Vitré and from Orleans to Chalons. It gave good results, but at the death of the inventor experiments with it were discontinued. It is of interest to recall that the author of this ingenious system was a watchmaker.[3]

36 The use of a loose axle may be combined with that of external coupling rods in which are provided slots and in these slots, again, glide the coupling pins of the radial axles. An arrangement of coupling rods of this type was used in the Cowles system, mentioned above, (Compare Fig. 8).

37 The Köchy system (Fig. 22) also furnishes an example of

[1]Matériel roulant et Exploitation technique des Chemins de Fer, T. 11, pl. LXXV, Fig. 1, 11.

[2]Engineering, November 15, 1867, p. 456.

[3]Annales des Mines, 1876.

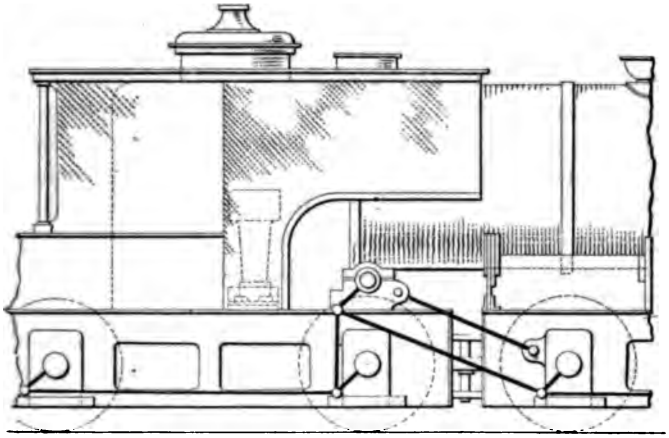

FIG. 20 LOCOMOTIVE "STEIERDORF" BY PIUS FINK

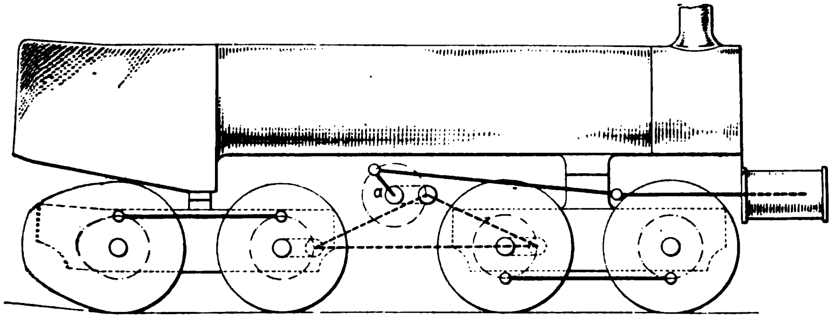

FIG. 21 RARCHAERT'S FREE-AXLE DESIGN

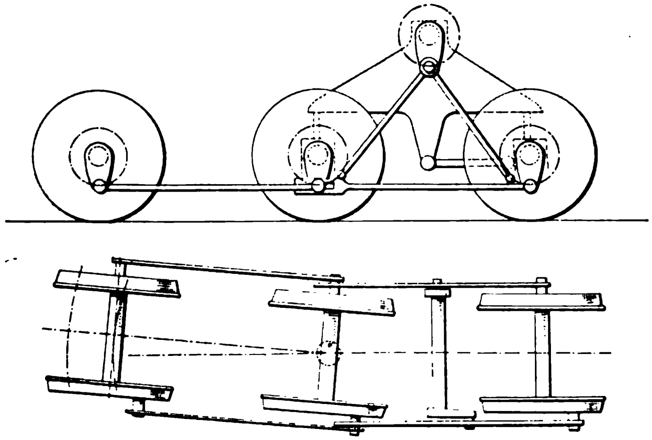

FIG. 22 KÖCHY FREE-AXLE SYSTEM

this arrangement.[1] A third example is found in the design presented by the engineer Gouin, author of the system employing oscillating levers similar to the first system of Rarchaert (Fig. 23). A model of this system exists at the Conservatoire des Arts et Métiers, in Paris, where it was placed in 1869. This model represents a combined locomotive and tender with five axles, of which the two rear ones form a pivoted truck. These axles are coupled by connecting rods having an open or slotted link in which slides the pin of a crank forming the terminal of a free axle. This cranked free axle is in turn coupled with the third axle of the machine.

38 At the Conservatoire, there is also another model showing a locomotive designed on the same principle by the same engineer in collaboration with Boutmy, former engineer of the Lyons Railroad. It is a locomotive having six axles two of which are fixed at the center and the others form two pivoted trucks located at the ends and operated by a rod transmission arrangement similar to the preceding one.

39 Finally, the well-known designer, Krauss, of Munich, proposed in 1893,[2] an arrangement permitting of the operation of the axles of a truck by steam cylinders carried on the main frame of the engine, as shown in Fig. 24. To accomplish this, the crank pins on the driving shaft carry pin blocks working in slots in the trussed connecting rods. It may be seen that the use of such connecting rods with slots is subject to serious objections. Stress is exerted on the crank pin in a vertical direction only and, moreover, the pin blocks have on curves a periodic displacement in a direction transverse to the axis of the connecting rod. As a result of this, friction and considerable wear is likely to ensue, rapidly producing play, shocks and dislocation of parts. None of these systems appears to have been utilized practically.

40 *External Connecting Rods the Lengths of Which Vary with the Convergence of the Axle.* In this connection reference will be made first to the Klose system, which has been fairly widely applied. Fig. 25 gives an idea of this system. It may be seen from this figure that the crank pin of the working axle carries a kind of rocker lever to two points of which are connected the coupling rods of the other axles. The other two points are connected to the extreme axles by a system of connecting rods and triangles in such a manner that the

[1]Zeit. des Ver. deut. Ing., August 26, 1893, p. 1054.
[2]Zeit. des Ver. deut. Ing., January 26, 1894, p. 28.

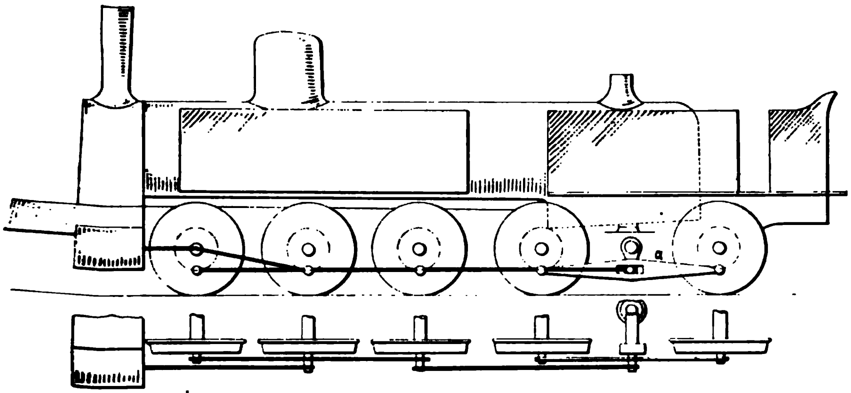

FIG. 23 OSCILLATING LEVER SYSTEM BY GOUIN

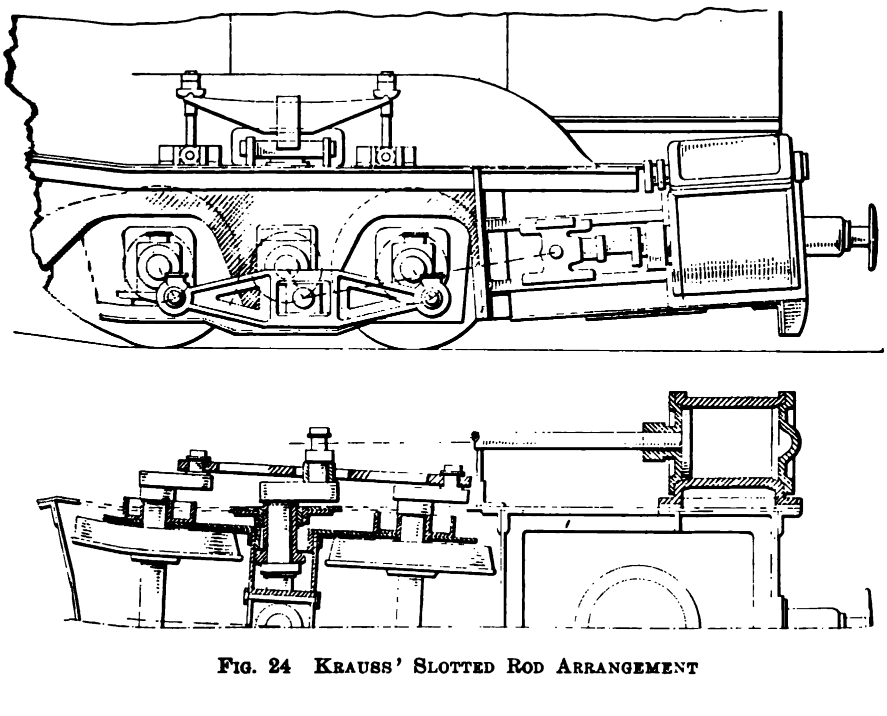

FIG. 24 KRAUSS' SLOTTED ROD ARRANGEMENT

convergence of the axles corresponds to the variation in length of the coupling rods. This system has been employed in locomotives having a gage of 1.76 m. (5.77 ft.) on the Bosnian-Herzegovinian Railroads and on large five-axle locomotives of the Würtemberg State Railroads.

41 In the Vogel system (Fig. 26), it has been proposed to couple the fixed axles of the locomotive with the axle of a pivoting truck placed under the tender. The crank pin of the wheel glides in a slot cut in the external connecting rod and the working coupling rod con-

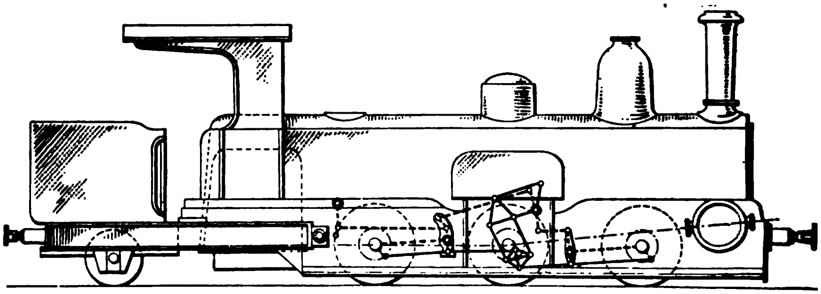

Fig. 25 Klose Variable Length Rod System

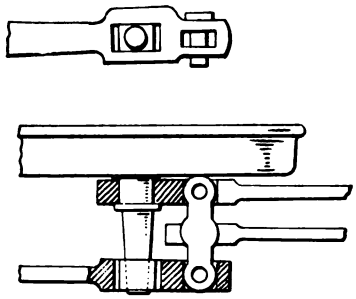

Fig. 26 Vogel Offset Rod System

nects with the spherical head of a pin connecting the two coupling rods. This system is described in the Organ für die Fortschritte des Eisenbahnwesens, 1878, p. 59, pl. V.

42 An arrangement proposed by a Brazilian engineer, G. Fretl, and described in The Engineer of September 26, 1884, p. 246, might also be cited. In this a double horizontal box engages with the crank pin of the middle axle and, by bearing on the coupling rods of the two other axles, increases or decreases the length of these coupling rods when the middle axle is transversely displaced on a curve.

43 Finally, an arrangement of coupling rods analogous to that

of Klose has been proposed by E. Neuhaus of Chemnitz,[1] to connect two trucks of two axles each, the front truck carrying the cylinders.

44 It is of interest to recall that more than seventy years ago the American engineer, Norris, described, in French Patent No. 17112, dated August 28, 1843, an arrangement consisting of a coupling rod with two parts sliding, one with respect to the other, along an oblique line coinciding with an arc of a circle described by the center of a radial axle. It appears that this arrangement cannot be executed practically.

CONCLUSION

45 In this paper the author has indicated the most interesting arrangements, so far as he knows, which have been proposed for operating the converging and parallel axles on a locomotive by a single pair of steam cylinders. If he has failed to mention any, especially those of American origin, it has been unintentionally and he apologizes for it in advance.

46 An examination of these devices gives the impression that all of them involve a serious inconvenience, and that all of them can operate in a satisfactory manner only when they are in vertical play, parallel to the longitudinal axis of the engine, *i.e.,* when the latter runs along straight sections of track. But such is not the condition on curves where the transmission element acquires a certain amount of obliquity, which necessitates the use of pins or spherical parts more difficult to lubricate. This obliquity introduces differences in length and further play between the parts, and this, in turn, leads to shocks and rapid wear of parts. Hence the maintenance of mechanisms of this kind becomes necessarily more costly than that of the ordinary locomotive transmissions.

47 Notwithstanding these difficulties, the author believes that in view of the ingenuity which has developed in the study of this question during so many years and by so many inventors, it would be hard to affirm that a system may not finally be found combining all the conditions essential to the practical operation of such a device. On the other hand, however, can it not be questioned whether researches in this direction are of any actual utility today when there is no hesitation with regard to coupling directly the largest number of parallel axles by external connecting rods and when there are other perfectly satisfactory solutions of the problem based on a different order of ideas?

[1] Zeit. des Ver. deut. Ing., May 26, 1894, p. 694.

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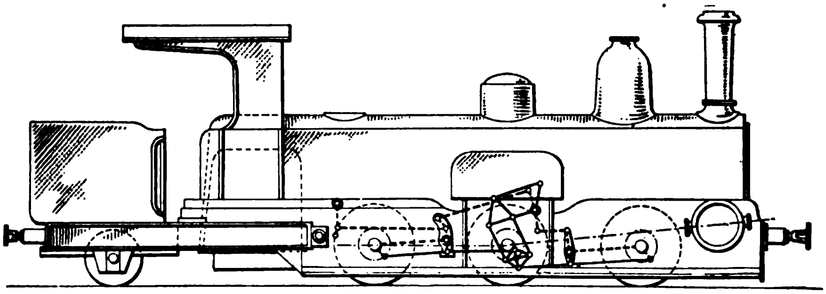
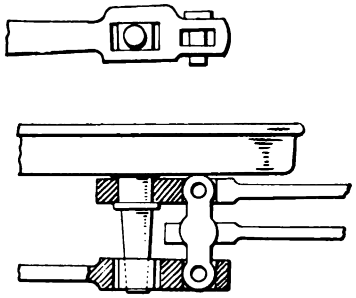


FIG. 25 KLOSE VARIABLE LENGTH ROD SYSTEM



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[1]Zeit. des Ver. deut. Ing., May 26, 1894, p. 694.

48 As a matter of fact, since 1861 and more than fifty years ago, J. J. Meyer, an eminent engineer and author of the first system of articulated locomotives which has given practically satisfactory results, wrote the following:

> "In the systems proposed for coupling in a rigid manner in whatsoever way it may be, the several axles belonging to two diverging trains, the addition of couping mechanism introduces a greater complication than the addition of two extra steam cylinders, and the maintenance of these mechanisms, as well as keeping the drive wheels rigidly to the same diameter, will be of greater cost than that of the two extra cylinders and the two mechanisms, without taking into consideration the loss in efficiency."

49 What has been said above is all the more true today, since, in addition to the Meyer machine, we now have the Fairlie system and the author's system, which was the last to come and which has received in recent years such important and remarkable application in the United States, thanks to the energy and skill of American engineers and builders.

DISCUSSION

E. A. Averill. It is possible in a two-cylinder locomotive to obtain a tractive effort of 100,000 lb., as far as the cylinders are concerned. In view of the limitations of the satisfactory factor of adhesion and safe weight on each driving axle, such tractive effort would necessitate the use of six-coupled drivers. Six-coupled drivers of 60 in. diameter means a driving wheel base of approximately 26.5 feet, which, if rigid, would be impossible for ordinary use, and almost impossible in any case. In this country we have not quite reached 100,000 lb. tractive effort, but we have reached 84,700 lb., and we have also reached a 22 ft. wheel base; and the step to 100,000 lb. tractive effort is one that is desired and probably will be taken. There is an important problem to solve in connection with the very long wheel base required, however, and this paper is particularly timely and should be given very close attention.

The paper itself is mainly historical and contains largely negative information, because the constructions shown are mostly those of failures. They certainly show how the problem cannot be solved, and that is very frequently the most important information that we have.

Carl J. Mellin (written). The excellent paper by Mr. Mallet is a valuable addition to the history of the development of the locomotive, and one from which references are readily obtained as to what has been done toward the solution of adapting driving axles to curves. This problem has been under constant consideration from the beginning of the locomotive era to the present day and has occupied many minds.

Various forms of gearing were among the first means sought to transmit motion from one set or group of axles to another, each set of axles to be capable of independently adjusting itself to the curvature of the road. This has probably been the most successful principle, through the range of the numerous designs made for this purpose, within the size and weight of the engine where one set of cylinders could supply the power.

The first application of chain transmission was also made at an early date, but probably owing to defective or insufficient strength of the chain this method was abandoned at the outset, although reintroduced at a comparatively recent time. With a more suitable design of chain, a number of log engines have been built and are reported doing satisfactory service. In a rougher roadbed they permit greater freedom of the axles than do coupling rods.

Theoretically the most correct method for self-adaptation of individual axles to curves, is probably the Haywood idea. It does not appear, however, that this can be applied to any but light engines, due to the unfavorable manner of carrying the weight on the crank axle on the ball connection between the frames. It is also a question whether the wheels will run steadily on a straight track, not being confined by any more rigid bearings than the single ball in the hollow axle. This remark holds good also for the hollow axle designs of Cowles, and Klein and Lindner.

The various other designs are not applicable to modern sizes of locomotives, but have nevertheless served as stepping stones for further progress to meet modern demands of simple and practical applications; and by the introduction of the four cylinders, followed by the Fairlie designs of double swivelling bogies and further by the application of the compound principle, the Meyer and Mallet systems, were brought into existence.

W. F. Kiesel, Jr. (written). Mr. Mallet has given us a most interesting historical presentation of attempts to provide a flexible drive for coupled wheels on driving axles which are permitted to assume a radial position. Apparently few of these designs have reached the experimental stage and none have come into general use. This is sufficient indication that the practicability of the schemes are doubtful. The further fact that other solutions to the problem, less expensive and complicated, have been found would lead to the conclusion that such schemes, as illustrated in this paper, will likely never be adopted, as their only advantage would be that all the axles can be driven from a single set of cylinders.

The weight and size of modern locomotives are so great that the cylinders are now as large as road clearances will permit. If larger locomotives are built, and some have been built, the application of two or more sets of cylinders will probably be obligatory. If the number of sets of cylinders is increased, the Mallet type of locomotive is the logical type to use, as with that type no change in customary construction of side rods, pins, etc., is necessary. In the Mallet locomotive, all necessary flexibility that may be required on account of track curvature can readily be obtained, making it unnecessary to consider further the flexible drive. For the reasons given, it is extremely questionable whether further research on the subject of flexible drive would be anything more than an interesting intellectual pastime.

Another reason why such types are not likely to come into practical use is that the loss in efficiency would be greater than the loss engendered by carrying 10 to 15 per cent of the weight of the locomotive on truck axles.

The Mallet type answers the same purpose, is more efficient and less expensive, and has deservedly established itself as a permanent standard on railroads in general.

G. R. Henderson (written). To the writer, the most interesting clause of the able paper by Mr. Mallet is that in which he quotes Mr. Meyer as considering that the extra complication of the various methods of connecting different trains of axles is less desirable than additional cylinders, with articulated pipes. It seems that there are other points in favor of the additional cylinders, more important than the cost or efficiency, thermally considered.

When a large number of axles are operated by one pair of cylinders, we have the following objectionable features: *a* Large and unwieldly cylinder proportions and parts, *b* great loads on rods, crossheads, guides and main crank pins, *c* heavy rods and reciprocating parts, *d* increased difficulty in lubricating the bear-justified by the results of operation so far obtained. Observations of the engines in service show that there is no lateral motion of these wheels on tangent track and on ordinary line curves even when the engine is working very hard at moderate speeds. The tire wear also appears to be about evenly divided between the first and the second driving wheels.

The construction is also applicable to Mallet locomotives, thus increasing the number of pairs of coupled wheels in each unit with a corresponding increase in tractive power.

George L. Fowler said that all the designs illustrated in this paper are intended for easement of wheel pressures on curves, and he had made a few investigations of this subject, with apparently astonishing results.

A seeming fact was that the truck leading a locomotive has a very material effect upon distributing the lateral thrust. For example, in the case of a Consolidation locomotive, it was found that on curves the leading truck exerts the greatest amount of pressure on the rail, then the second driver, followed by the first driver, and third driver and fourth driver, in the order named. Running the engine backwards, the rear wheel strikes a tremendous blow and the remainder travel around the curve without much pressure on the track at all.

The same thing was manifest in connection with the Pennsylvania R. R. electric locomotive, that is, that the leading driver on the rear unit was the one that put most of the pressure on the rail.

The distribution of the lateral thrust depends, of course, entirely upon the type of engine. The easiest riding engine he knew of was the old-fashioned American 8-wheel engine, followed possibly by the Pacific, if weight was eliminated, although in a Pacific engine with trailing wheel, the pressure put on the track by the trailing wheel was invariably much higher than that by the rear driver. In this connection, too, the pressure of the tender on the track was found to be very light, but strange enough in the case of the sleeping cars it was two or three times that of the locomotive.

On a Pacific engine maximum single wheel pressures of from 13,000 to 14,000 lb. were obtained, while a sleeping car would give a pressure of 32,000 to 36,000 lb. He had obtained as high as 32,000 lb. on a freight car, but this was running on an 8-deg. curve at fifty miles an hour.

He thought the speed of Consolidation engines running back-

Floating coupled axles, in which an abnormal amount of lateral play is allowed to accommodate the curving of the wheel base, have also been used abroad and to a limited extent in this country. Floating axles with lateral play will certainly allow long wheel base engines to pass sharp curves easily, but their use is open to the objection that they do not contribute any guiding effort to lead the mass of the locomotive around curves, or to steady it on tangent track until the full lateral play is taken up. Moreover, on account of such axles being free to move laterally, almost all the flange wear comes on those coupled wheels which have normal lateral play. The ruling of the Interstate Commerce Commission covering allowable lateral play between driving wheel hubs and driving boxes, is also an objection to this construction.

It is evident that the design of lateral motion coupled axles which will best meet the conditions of the case should be capable of application to any or several pairs of the coupled wheels. Thus it may be desirable to equip the first and last coupled axle with a lateral motion device, or possibly even the first, last and middle pairs of wheels.

There has recently been placed in service an arrangement of lateral motion coupled axle which meets these requirements. It provides sufficient flexibility to admit of a locomotive having a long driving wheel base curving easily and at the same time affords a definite resistance against lateral motion. This design is in successful operation on a number of heavy ten-coupled locomotives on the New York, Ontario & Western R. R. and has also been used on a similar class of locomotives of unusual weight and power just going into service on the Erie Railroad.

Briefly, the design consists of an arrangement which permits of about two inches total side play of the leading coupled wheels and boxes. This lateral motion is resisted and controlled by a constant side resistance which is obtained through the action of the load carried on the boxes. In this way, a positive gravity control is obtained against an initial side motion of the wheels and throughout the entire range of this motion up to its limit. The side rods connecting this pair of driving wheels with the second pair of wheels are arranged with ball knuckle joint pins and a special design of spherical crank pin.

The principle of applying a yielding resistance to control the motion of the driving axle having lateral play appears to be fully justified by the results of operation so far obtained. Observations of the engines in service show that there is no lateral motion of these wheels on tangent track and on ordinary line curves even when the engine is working very hard at moderate speeds. The tire wear also appears to be about evenly divided between the first and the second driving wheels.

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He thought the speed of Consolidation engines running backward and switch engines should be limited to certainly not more than 20 or 25 miles an hour. Switch engines hauled over his apparatus, running dead in the train, had a marked effect on the track.

He said it was of no use attempting to make a mathematical analysis of what happens on curves. The distribution of the wheel loads on the lateral thrust is quite different from anything in connection with the center of gravity. For example, in the case of the Pacific Locomotive the greatest thrust is by the front truck wheel or main driver, sometimes one, and sometimes the other, but the rear driver thrust is low, while that of the trailing wheel is very high.

In regard to the limitation of side motion in the driving axle, mentioned by Mr. Woodard, he did not feel that there was any difference in a large number of engines, in lateral motion on curves, where the engine is apparently laying over against the outer rail and all the thrust is put there. Side motion on tangent track was apparently a very important element, and an old locomotive with from 1½ to 1¾ in. side motion in the journals slides over a tangent track with an ease that is perfectly surprising.

He thought most of the blow caused by a tuned-up engine was due to the track. His own experience coincided with that of the Pennsylvania Railroad on the New Jersey & Sea Shore Line, to the effect that if you got a heavy blow at a certain point on the track, at any speed, or in any one particular type of engine, you were apt to get it every time an engine passed over that point. Distortion of the track, too, is a serious matter.

If there is any limitation in regard to the side play in an engine, it should be very liberal. A worn engine does not put any greater stress on the track on a curve than one freshly out of the shop, and it puts remarkably less stress on the tangent track.

E. B. Katte. Some experience derived from the development of the earlier high speed electric locomotives does not conform with Mr. Fowler's in regard to lateral motion. It was found that in the early electric locomotives equipped with two wheel guiding trucks considerable lateral motion produced a hard knock against the track rails when running into or leaving a curve. The low center of gravity accentuated the blow and at 70 or 75 miles an hour the cumulative shock was considerable, while the same locomotive with the lateral motion reduced by hub shims was run up to 85 miles per hour without producing a blow against the rails. Unfortunately, the locomotive had to be shimmed up so tightly that it would not take the short radius yard curves, and other means to prevent the lateral rail thrust had to be found.

ROY V. WRIGHT asked whether it were possible to provide a large enough cylinder on a locomotive to take care of six-coupled axles, and C. D. Young asked what were the limitations to one design. What was the maximum play that could be given the front and rear axles?

W. E. WOODARD replied that he thought this was possible. Such a locomotive would probably take a 32 or 33-in. cylinder, well under the size of those used on the Mallet engines. He did not see any obstacle in the way of our proceeding up to a 100,000 lb. tractive effort simple locomotive. With a larger cylinder the centers would have to be thrown out further, and to prevent side vibration the wheel base would have to be made considerably longer.

In reply to Mr. Young, at present we have only 1⅛ in. play on each side, but we could probably arrange for 1½ in., or possibly more.

C. D. YOUNG said that he could see no particular difficulty in reaching 100,000 lb. tractive effort in a locomotive of other than the Mallet type, but did not believe that the use of two cylinders was the only way to do it. It should be done with two pairs of simple cylinders, or with three simple cylinders, or if two cylinders are designed to keep within the clearance diagram of the right of way, by increasing the boiler pressure and limiting the cut-off. In that way the long overhang of the main pin which results from the wide spread of cylinders would be overcome, as well as cylinder clearance.

If the cylinders were made large enough to limit the full gear position to a reasonable cut-off, he thought that a boiler could be made which would develop the possibilities of 100,000 lb. tractive effort with two or three cylinders. He saw no use, however, for such a locomotive if the boiler could not supply enough steam.

There should be three or four cylinders large enough to permit the valve gear to be so arranged that the maximum cut-off would not be over 65 or 70 per cent, thus making it possible to develop full tractive effort at a reasonable water rate.

The water rate of the two-cylinder engine, working in full gear at seven or eight miles per hour, is about 31 lb. per i.h.p. hour with

125 deg. of superheat. If the cut-off is reduced to 50 per cent the water rate drops to 18.5 lb. The difference in boiler requirements between these two valve gear positions is evident. Likewise, if two cylinders are used and the boiler pressure is increased a sufficient amount to limit the cut-off to 50 per cent, corresponding results would be obtained, so far as the water rate is concerned, but with such an arrangement, some provision should be made in the valve for starting, or a special starting valve provided with the two-cylinder locomotive.

S. M. VAUCLAIN (written). I have read with great interest the author's historical account of the development of the different systems of radial axles on locomotives operated by a single set of cylinders.

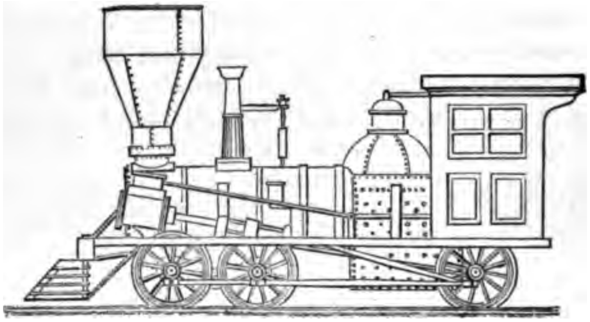


FIG. 27 BALDWIN SIX-WHEEL CONNECTED LOCOMOTIVE

Mr. Mallet's paper covers only the use of transmission adapted to curving when such a transmission is driven by steam cylinders. Electric drive lends itself to the use of radial trucks without serious complications, and the various systems employed both in this country and abroad have been described so frequently that a further description will be of no special interest.

In 1842, Mr. Baldwin took out a patent on his 6-wheel connected locomotive in which the four front drivers were combined in a flexible truck. This locomotive was especially adapted to use on American roads then existing, which had extremely sharp curves, and a large number of locomotives of this type were built and were very successful. The original engine of this type, as shown in Fig. 27, had six wheels, all connected as drivers. The rear wheels were placed rigidly in the frame and had inside bearings. The four *remaining* wheels had inside journals running in boxes held by two wide and wrought iron beams. These beams were entirely independent of each other and the pedestals formed in them were bored out cylindrically for the reception of cylindrical boxes. The engine frame on each side was directly over the beam, and a spherical pin from the frame bore in the socket between the beam, midway between

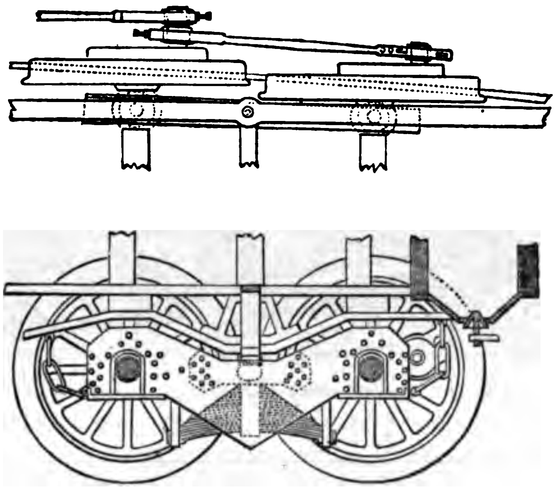


FIG. 28 TRUCK FOR LOCOMOTIVE (FIG. 27)

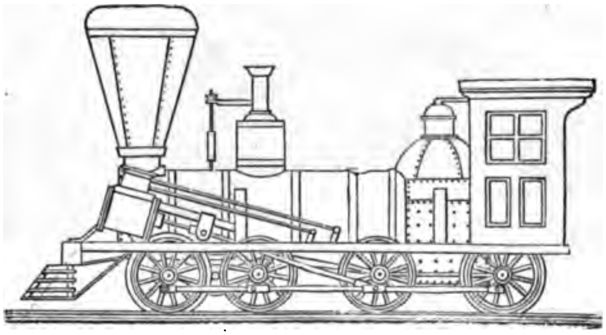


FIG. 29 BALDWIN EIGHT-WHEEL CONNECTED LOCOMOTIVE

the two axles, so that the operation of the beam was similar to the parallel ruler. A half plan of this truck is shown in Fig. 28. The coupling rods were also made with cylindrical brasses.

This design was subsequently modified into an 8-wheel connected locomotive, shown in Fig. 29. In this arrangement both pairs of drivers are rigid and connected to the boiler and steam machinery. The wheels of the front truck are connected by side rods to the drivers and the truck frames are pivoted under the engine frame to obtain a parallel movement to the axle when on curves. This

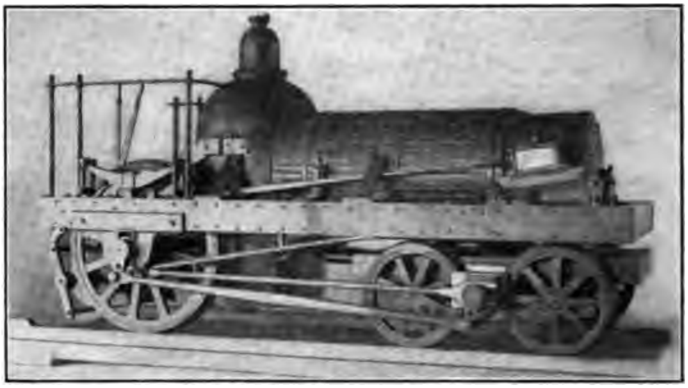


FIG. 30 GEAR-DRIVEN TRUCK LOCOMOTIVE

FIG. 31 MOTOR CAR WITH PIVOTED TRUCKS

lateral movement is possible since the axle journals and the rod brasses are cylindrical in their boxes.

Fig. 30 shows a design in which one driving wheel is coupled to a shaft driving a gear, which in turn drives truck wheels of smaller diameter. The truck is adapted to curves in the same manner as that previously described.

Fig. 31 shows a motor car with a compound engine in which the driving trucks are pivoted and steam is admitted to the cylinders through the carrying bearings. The boiler rotates in the body of the motor car as the driving truck accommodates itself to the curve. The rear truck in this arrangement is simply a carrying truck and is not driven. This locomotive was built in 1897.

The foregoing are by no means all of the American designs of locomotives with radial axles, but they serve to show different solutions of the problem, all of which have been in successful operation.

F. J. Cole (written). This paper shows in concise and orderly fashion the principal and most interesting arrangements which have been used or proposed for parallel driving mechanism arranged for operation by two cylinders, having all the weight available for adhesion, with the advantages of radial axles for traversing sharp curvature. It brings all these designs together in one paper with illustrations and descriptions, and makes available the information contained in technical literature extending over many years.

It is probable that such designs, with the exception of geared locomotives, similar to Shay, Climax, etc., attract much more serious consideration abroad than in this country. It is significant, however, that few, if any, of these locomotives are now in operation, the inherent difficulties and complications probably being so great that they more than offset the advantages gained by their use. They have also been superseded by designs having more than two cylinders in which the necessary radial action is obtained by swiveling trucks, for instance the Fairlie and Pechot. Usually such engines are built for slow service on roads with heavy grades and sharp curvature. The use of four cylinders is often more economical, all things considered, than the complication incident to these special forms of radial axles.

The author, in the invention of the locomotive which bears his name, has done more to render unnecessary such designs (except geared, similar to Shay) described in the paper, than probably any other man. These Mallet or articulated locomotives, as is well known, are adapted to the widest possible range of service. Originally they were intended for military use for very narrow gages and extremely sharp curvature, the track following the natural undulations of the ground with but little grading. At the present time in this country, we see their very highest form of development in hauling the heaviest tonnage trains in the world, such for instance, as the 2-8-8-2 type for the Virginian Railway, having a tractive power of 115,000 lb. when working compound and 135,000 lb. when working simple for short distances; weight 481,500 lb. on drivers and 542,500 lb. total in working order. It is probable that even these figures will be exceeded in the near future.

We see again modifications of this design in the Triplex, which is a triple articulated compound, with the weight of the tender utilized for adhesion. Such locomotives are suitable only for very slow service because of their boiler or steaming limitations.

Engines of the Mallet type operate with the axles of the front engine substantially radial to the track. They can be built in all designs from 0-4-4-0 to 0-10-0 or 2-10-10-0 and many other intermediate modifications of arrangement of trucks and driving wheels carrying either all the weight on the drivers or provided with leading and radial trailing trucks suitable for the road conditions under which they operate. I therefore agree with the author that it seems unnecessary to consider seriously many of the designs shown in the paper.

Five-coupled engines, having leading and sometimes trailing radial trucks with lateral play in the front and occasionally in the rear driving axles, can be operated successfully on sharp curves. It is better practice to use some form of gravity or spring resistance in axles having much lateral play in order to distribute the flange wear. Whether for safety these axles actually require centering devices or control, depends upon the speed and other conditions under which they are operated. Such devices are now available in simple form, which operate successfully. It is only in heavy grade service that locomotives having all the weight on the drivers can be operated satisfactorily. The horsepower in such service is relatively small, therefore the steam requirements are not so exacting. For fast service, in order to get sufficient boiler capacity for conventional types of locomotives, it has been found necessary to carry a certain portion of the engine on leading and sometimes on trailing axles, not only to provide for the guiding of the locomotive on curves and therefore to make the operation safe, but to permit the construction of a boiler of sufficient size to provide the steam required.

W. F. M. Goss said that from our understanding of the general problem, it would appear that the developments described in the paper are not of great practical value. Our experience seems to have passed the stage of development with which Mr. Mallet deals. The fact is, however, that we can never tell what the next step will be, because we can not predict the conditions that are to govern it. It is perfectly clear, for example, that the chain transmissions shown in these designs were destined to failure at the time they were employed, for in that day chain drives were crude; but it is equally clear that if for any reason we should desire today to couple up locomotives with chains, we could do it successfully because practice in chain gearing has passed from an experimental stage to one of great refinement. Changing conditions, therefore, emphasize the profit to be derived from study involving the historic development of mechanisms. I regard the masterful presentation of Mr. Mallet as a contribution of high value to our Transactions.

The Author was unable to present his paper in person, but it was presented by E. A. Averill, Mem. Am. Soc. M. E., who also contributed to the discussion. A letter written by Monsieur Mallet and received subsequent to his manuscript included the following paragraph which, he said, established a very clear case of priority in favor of an American designer:

The author wishes to recall here that probably the first case of transmission by convergent axles on a locomotive from a fixed steam cylinder by means of a connecting rod located in the longitudinal axis of the engine, is to be found in the locomotives built in 1832 by Horatio Allen for the South Carolina R. R. In this engine the main rod had a vertical jointing with a small stub end and large head spherical journal, so that, on curves, it lent itself to an oblique action on the central arm of the axle.

No. 1503

FOUR-WHEEL TRUCKS FOR PASSENGER CARS

By Roy V. Wright, New York, N. Y.
Member of the Society

The Pennsylvania Railroad uses four-wheel trucks under all of its passenger coaches, although the P 70 class, 70 ft. in length and having a seating capacity of 88, weigh light from 118,000 to 122,000 lb.; loaded with passengers they weigh about 135,000 lb., and never more than 140,000 lb. It is the standard practice on that system to use such trucks under all passenger equipment cars weighing less than 120,000 to 125,000 lb., except for so-called load carrying cars, including baggage-express, mail, baggage-mail, etc., which are designed to weigh over 140,000 lb. when loaded. The light weight of the bodies of the Pennsylvania P 70 coaches—and these are now standard on that system—varies from 93,000 to 96,000 lb. It is assumed that these cars regularly carry as much weight in passengers and hand baggage as coaches on other roads, inasmuch as they seat 88 persons, or several more than the maximum provided for in the standard coaches of most roads. It is the practice on the great majority of railroads to use six-wheel trucks under coach bodies weighing much less than this, comparatively few roads using four-wheel trucks under bodies weighing more than 85,000 lb. and many of them using six-wheel trucks under bodies weighing even less than this.

FACTORS IN DESIGN

2 In designing the trucks for a passenger coach four features must be kept in mind, and generally in the following order as to importance, although there may be some question as to the relative value of the last two.

a They must be designed for safety.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Railroads.

b They must ride smoothly, for travelers are particular as to this in these days and will desert a road with rough riding cars if a competitor furnishes better service. With heavy steel cars operated in long trains at high speed and with the locomotives taxed to the limit of their capacity, it is difficult to operate and brake the trains without occasional roughness and jolts, and a factor such as truck design cannot be allowed to contribute further to the rough riding.

c The weight of the truck must be kept to a minimum if for no other reason than the effect on the cost of conducting transportation.

d The truck should be designed with a view to keeping the cost of maintenance as low as possible. Here, as in the requirement for safety, it is desirable to have as few parts as possible and of simple construction.

DOES THE FOUR-WHEEL TRUCK MEET THESE REQUIREMENTS?

3 How does the four-wheel truck meet these requirements under the heavy passenger equipment on the Pennsylvania Railroad?

4 *a* The four-wheel truck of modern steel construction which has been in use on that system for a number of years has given splendid satisfaction so far as safety is concerned. As on other roads, some trouble has been experienced with hot boxes and it was at first thought that the journal bearing area was too small. The use of larger bearing areas does not seem to have materially improved conditions and it is now believed that the difficulty is entirely due to dirt or gritty matter entering the journal boxes. The problem then becomes one of improving the journal box lid and dust guard to prevent this, rather than to increase the diameter or length of the journals.

5 There has been no breakage of axles, except for three cases due to defective material when the first steel trucks were introduced many years ago. No physical weakness has developed in any of the parts in the ten years the trucks have been in service, so that as far as safety is concerned there can be no question. The possibility of accident would seem to be less with the four-wheel truck because of the smaller number of parts required.

6 *b* There seems to be a feeling on the part of some mechanical engineers that the four-wheel truck, with its shorter wheel base (7 or 8 ft. as compared with 10 to 11 ft. for the six-wheel truck) will ride

less easily than the six-wheel truck. With coil springs over the journals, elliptical springs under the bolster, and provision for lateral motion of the bolster, it would seem that there ought not to be much difference in this respect.

7 Experiments show that much of the rough riding or jolting on passenger coaches has been due to the method of anchoring the top of the dead lever to the truck frame. The unbalanced forces in the truck when the brakes are applied tend to tilt the truck frame out of horizontal alinement, thus causing a "jerky" action. By anchoring the dead lever to the body underframe this is eliminated. This development is comparatively recent and affects the six-wheel as well as the four-wheel truck. The effect of anchoring the dead lever to the truck frame has possibly been more noticeable on the four-wheel truck, because 1— to —1 dead levers are used, resulting in a greater pull on the frame than in the case of the six-wheel truck; then, too, the resisting moment is less because of the shorter wheel base of the four-wheel truck. This improvement has been patented.

8 *c* There is a wide variation in the weights of different types of steel passenger car trucks, but it is probably fair to state that a pair of four-wheel trucks will weigh from 10,000 to 15,000 lb., or more, less than a pair of six-wheel trucks having the same carrying capacity. In other words, for the same total weight of car the one with four-wheel trucks will carry 10,000 to 15,000 lb. more lading or body weight, or with the same weight of body the total weight of the car with four-wheel trucks will be from 10,000 to 15,000 lb. less than the one with six-wheel trucks. For a car weighing 120,000 lb. and equipped with four-wheel trucks this means a saving of from 8 to 11 per cent in total weight as compared with what it would be if six-wheel trucks were used. On most roads it is the practice to carry car bodies weighing more than 85,000 lb. on six-wheel trucks, which weigh fully 15,000 lb. per car more than four-wheel trucks. A locomotive that can haul eight cars equipped with such six-wheel trucks over a given division will haul nine cars of the same seating capacity having four-wheel trucks—a saving much to be desired.

9 *d* Roughly speaking, the cost of maintenance of a steel passenger car truck may be said to be very nearly in proportion to the number of its wheels and axles, these with the brake shoes being the parts subjected to the greatest wear and requiring frequent repairs and renewals. While no exhaustive data are available as to the com-

parative cost of repairs and maintenance of six-wheel and four-wheel trucks of the same carrying capacity, they are said by those who have checked these costs to be at least 50 per cent greater for the six-wheel truck.

DEVELOPMENT OF PENNSYLVANIA FOUR-WHEEL TRUCK

10 As a partial check on these conclusions, it is proposed to review briefly the development of the four-wheel steel truck for passen-

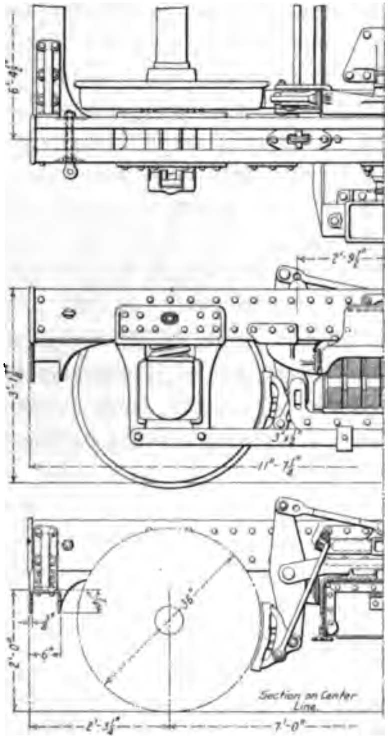


FIG. 1 ONE END OF ORIGINAL FOUR-WHEEL STEEL PASSENGER CAR TRUCK BEFORE THE APPLICATION OF CLASP BRAKES; PENNSYLVANIA RAILROAD

ger cars on the Pennsylvania Railroad. From the outset and throughout this development the aim has been to reduce the number of parts to a minimum and make the construction as simple as possible. The problem has been complicated somewhat by the necessity of providing for the application of motors to the trucks used under motor cars in electrified districts and also by the application within the past few years of clasp brakes, which are now standard on the Pennsylvania for all four-wheel trucks and all new passenger equipment trucks.

11 In designing the first four-wheel steel trucks in the early part of 1905, it was aimed to use them under the largest coach possible and keep within the M.C.B. load limits for 5-in. by 9-in. axles.

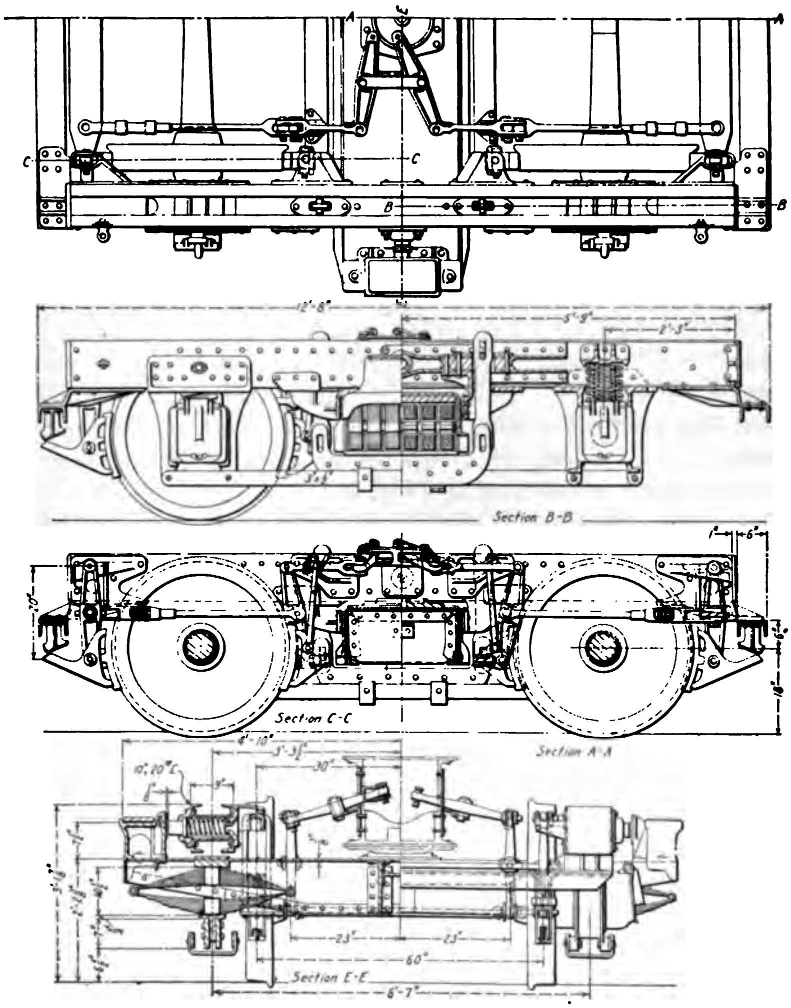


FIG. 2 ORIGINAL FOUR-WHEEL STEEL PASSENGER CAR TRUCK WITH CLASP BRAKES APPLIED; PENNSYLVANIA RAILROAD

Shortly after the trucks had been placed in service, three of the axles broke in the wheel seat where the stress is least. Investigation finally showed that the breakage was due to defects in manufacturing caused by a faulty furnace which had been discarded shortly after these axles were made. In the meantime, however, as a measure of absolute safety, it was decided to increase the axles on existing cars ½ in. in diameter and on new cars to go to the next larger size standard M.C.B. axle, the 5½-in. by 10-in. Because of hot box troubles the length of journal was afterward increased to 11 in., although experience has since indicated, as previously noted, that the trouble was probably due more to dirt getting into the journal box than the lack of journal bearing area. The 5½-in. by 11-in. journal is now standard for all four-wheel and six-wheel trucks.

12 In going from the wood to the steel construction spring planks, axle guards and brake beams were done away with, the brake levers being attached directly to the brake heads. Each side frame was formed of two 10-in. 20-lb. channels, with the flanges turned inward and forming a box girder construction. The channels were spaced so as to measure 9 in. overall. This was done to provide sufficient strength for resisting the lateral stresses, a requirement which has been overlooked in some designs. To check or limit the lateral motion or swaying of the bolster a spring arrangement was used, as shown in the drawing (Fig. 2).

13 The subsequent use of clasp brakes made it necessary to modify this design somewhat. Fig. 2 shows the details of this modified design, which in general is practically the same as the original design, other than the braking arrangement, except for changes in the end construction of the frame to provide for the outside brakes. The detail of the original end construction is shown in Fig. 1. The end rail in the original design, which was formed of a ⅜-in. plate pressed in the form of an inverted U, 6 in. in width, was changed to make room for the brake levers. The outside brakeheads in the case of the clasp brakes are attached to the lower ends of the brake levers, which are anchored at the top to castings riveted to the ends of the side frames. A 6-in. channel with flanges turned downward connects these castings and forms the end rail. It was also necessary to add brakehead tie bars because of the impossibility of connecting the tension rods for the outer brakeheads direct to the brake lever. It should be noted, however, that this brakehead tie bar is a simple rectangular bar and that the brake tension rod connects to it as close to the brakehead as possible. Obviously the weight and the cost of maintenance of this tie bar is much less than for a brakebeam where the force is applied at the middle. All of the brake levers, including the dead and live levers, are of the same size and interchangeable *except* for drilling.

14 The peculiar form of the outer brakehead is noticeable. In the first application of the clasp brakes the ordinary type of brakehead was used with springs to hold it balanced when hanging loose. These springs were difficult to maintain and were done away with by redesigning the brakehead and adding the tail piece. When the brakehead hangs loose this tail piece rests against a casting which is riveted to the underside of the end rail. When the brake is applied there is a clearance of ½ in. between the brakehead tail piece and the rest. This device has given most satisfactory results.

15 The next development was a modification of this design to provide for the application of a motor for use under motor cars on electrified divisions. To do this it was found necessary to increase the wheel base from 7 ft. to 8 ft. 6 in. Transoms were also added to support the lip of the motor and the bolster design was modified slightly; otherwise the same parts were used as in the original design.

16 The next development was a radical one, the box girder sideframe being replaced by a Bethlehem 10-in. 54-lb. H-beam, thus simplifying the design as to construction by reducing the number of parts and still providing sufficient moment to resist the side stresses. As shown in Fig. 3, the journal box pedestal casting has a projection to which the top of the lever for the outside brake is anchored and which also supports the end rail, a 6-in. H-beam. The H-beam which forms the side frame has its lower flange and web cut away over part of the journal box pedestal casting and is strongly riveted to it through both the upper and lower flanges. The casting which was formerly used on the end rail to balance the brakehead was replaced by a steel clip which is sprung over and welded to the lower flanges of the end rail.

17 Another noticeable change was the shortening of the bolster hangers, thus limiting the amount of side swing and making it possible to do away with the complicated spring mechanism which was formerly used to check and limit the lateral motion of the bolster with the longer hangers. Before making this change, the springs were gradually blocked and finally wedged solid on a number of the cars. As this had no noticeable effect on the smooth riding, it was decided to discard the springs entirely.

18 The more important of these changes, that is, the side frame construction and the change in the hanging of the bolster, were first made on four-wheel trucks for suburban cars, several hundred of which were built. These trucks, however, were of lighter construction

than those used under the standard coaches and are not considered in this discussion. The details of this improved truck as designed for use under standard coaches are shown in Fig. 3.

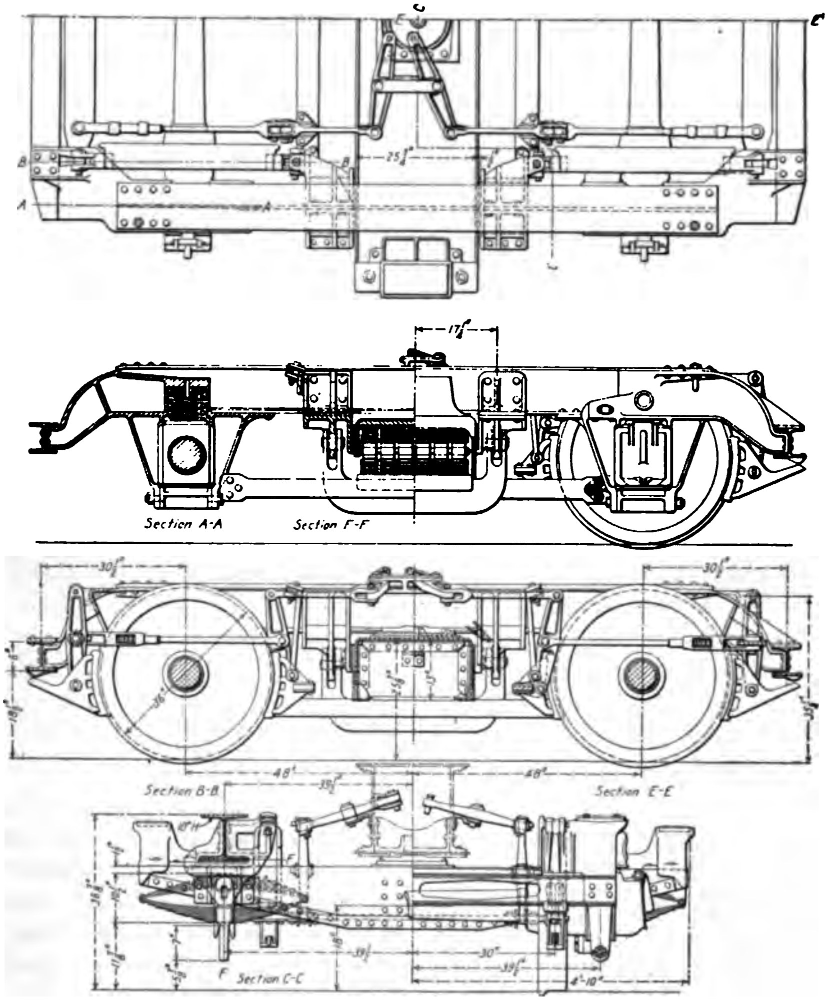


FIG. 3 PRESENT STANDARD FOUR-WHEEL STEEL PASSENGER CAR TRUCK; PENNSYLVANIA RAILROAD

19 A modification of this standard truck was made necessary by the Philadelphia-Paoli electrification and is shown in Fig. 4. The most powerful motors, 225 horsepower each, thus far used under passenger coaches are required in this service. To provide for them it

was necessary to extend the wheel base of the truck from 8 ft. to 8 ft. 8 in. Because of the great amount of room required by the motors, blower apparatus, etc., it was necessary to do away with the brakehead tie-bars, or brakebeams, and to arrange the tension rods

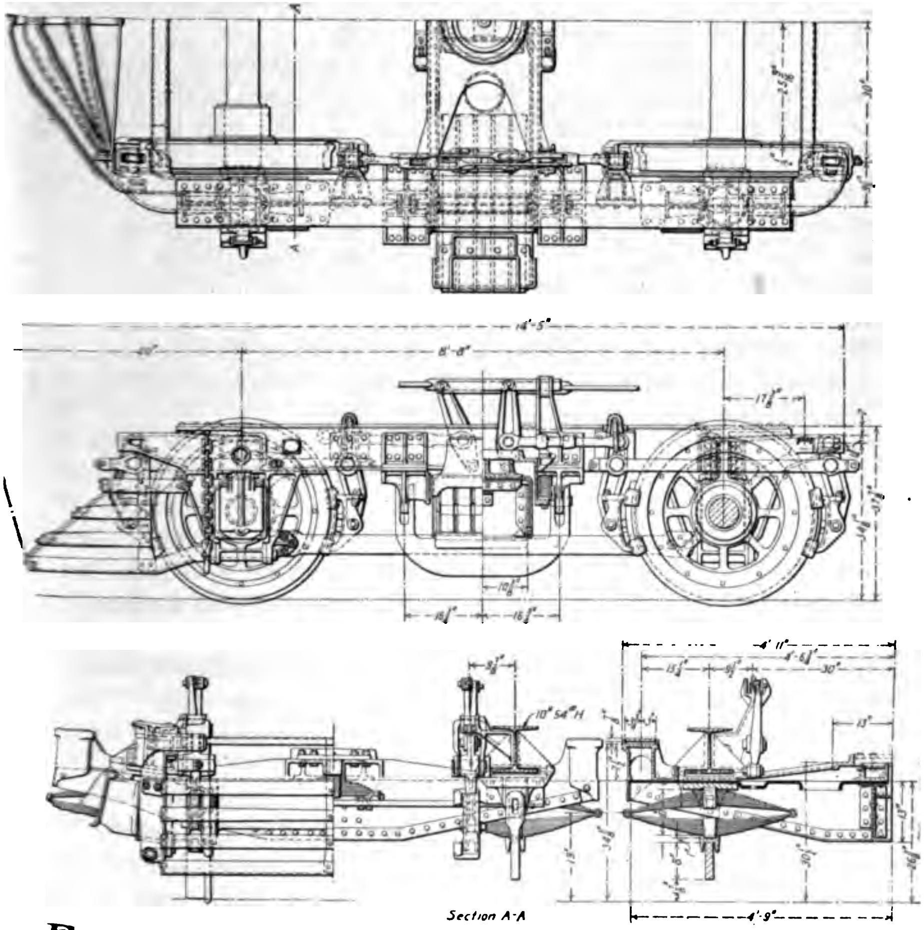


Fig. 4 Standard Four-Wheel Steel Passenger Car Truck Modified for Use Under Motor Cars on Electrified District of Pennsylvania Railroad at Philadelphia

for the outside brakeheads to straddle the wheel as shown in the drawing. This application also made it possible to change the construction at the ends of the side frames and the end rail. The details of this design are presented in order to show how the four-wheel standard truck for use under heavy coaches could be preserved in its general features and be arranged for use under the heavy modern electric motor cars. The drawing shows the use of spoke wheels; recent practice has changed these to plate wheels, and the gear wheel, which meshes with the motor pinion, is now fastened to the axle rather than on the projection of the wheel hub.

20 It is necessary to provide a blower apparatus to cool the large motors which are used. The blowers are fastened to the underframe of the body of the car near the center and the draft is carried through a duct formed by the box shaped center girder of the car between this point and the bolster. From the latter point a duct extends laterally from each side of the center girder delivering the air where it is needed. The motor leads pass through holes in the truck bolster close to the center plate so that the curving of the truck creates the least possible amount of distortion in the leads. Fortunately the necessity for making some such provision was foreseen a number of years ago and it was not necessary to make any important modifications in the design of the truck bolster.

DISCUSSION

Geo. R. Henderson (written). In reading Mr. Wright's very interesting paper, the writer's thoughts followed the abandonment of the equalizers which for so many years were considered necessary for easy riding, especially on rough track. It is well known that the Pennsylvania track is nearly perfect and it is a question whether these trucks and loads would be satisfactory on average track. This accounts for a considerable saving in weight, as the equalizer and spring seats were quite massive for heavy cars.

It is still the practice of many roads to use equalizers under tenders of passenger locomotives and also under high speed electric cars, and it is an interesting question as to just how far we can go in this respect and still not interfere with the comfort of the passengers. The condition of the track is a very important part of the problem in the writer's estimation and should not be overlooked when considering this question.

S. G. Thomson said that the clasp brake as used on the Philadelphia and Reading Railway, today, was giving most excellent satisfaction, and has done so ever since it was developed. One hundred and fifty or more cars have been in service for a number of years, and little or no difficulty has been experienced. The brake is highly efficient and the stops seem to be very much easier and shorter than with the single shoe and higher brake-shoe pressure.

In regard to the question of journal lubrication, he had found that at 80 or 90 miles per hour, some trouble was experienced from hot boxes. Careful readings of journal box temperatures seemed to indicate that the temperatures build up during closely consecutive runs, the wheels and journals not having had time to cool down before the return trips. During the rush season it was often necessary to schedule certain cars for six trips per day, equal to 300 miles or more. The cars were all-steel, weighing about 118,000 lb., which seem to be, in his judgment, about the limiting weight for four-wheel trucks at these high speeds, particularly where they are closely consecutive.

He thought, therefore, that the factor of speed seems to have a fundamental bearing on the question of the use of the four or six-wheel trucks on modern passenger equipment, and that the four-wheel truck should prove entirely satisfactory for anything less than the weights or high speeds on consecutive trips as has just been mentioned.

L. R. Pomeroy said it was astonishing how much money railroads spend to get another car on a train, to be hauled with the same engine, where the cost is more than it would be to run cars with logical trucks. He thought it was quite true that six-wheel trucks have been developed more generally in the West, and it is only in recent years that they have become general on ordinary passenger coaches.

In the West some of the reasons why the six-wheel truck was adopted on cars which would now be considered light were the very light gravel ballast, light rails (from 60 to 70 lb.), and cast iron wheels.

He presented the following figures of weights of four and six-wheel steel trucks under steel cars:

Four-wheel trucks on the Harriman Lines, 5 x 9 journals; weight 26,500 lb. complete. Western Pacific, $5\frac{1}{2}$ x 10 journals; 31,120 lb. complete. Barney & Smith special built-up truck, similar to the Atlantic Coast Line, 5 x 9 journals; 27,600 lb. complete. N. Y., N. H. & H. R. R. standard trucks, 31,400 lb. complete.

Six-wheel steel trucks on the Harriman Lines, 5 x 9 journals;

42,000 lb. complete. Rock Island, 5 x 9 journals; 41,220 lb. complete. Commonwealth, 5 x 9 journals; 40,900 lb. complete. Pullman standard truck, 5 x 9 journals; 43,720 lb. complete. New York Central latest form of steel truck, 45,000 lb. complete. The latter weighs considerably more than the average of the others, and is partly accounted for by being equipped with clasp brakes. Barney & Smith built-up truck, 5 x 9 journals; 42,000 lb. complete. Canadian Pacific trucks under 70-ft. diners, 41,200 lb. complete. Canadian Pacific under sleepers 72 ft. 8 in. long, 40,900 lb. complete; under sleepers 70 ft. 3 in. long, 41,800 lb. complete; under sleepers 74 ft. long, 41,800 lb. complete.

He did not think that even under ordinary conditions of service, there is any justification whatever for a six-wheel truck under a suburban car, and yet a great number are running on these trucks. He said he had never yet come across a 70-ft. car of the ordinary passenger coach type where the six-wheel truck was justified with the track in use today.

The Wabash has a 60-ft. all-steel mail car, 5 x 9 journals, six-wheel trucks, total weight 124,000 lb. The trucks weigh 42,000 lb. With four-wheel trucks, 27,000 lb., the total weight would be 109,000 lb. We have a great many cars that weigh as much as 124,000 lb. running successfully with four-wheel trucks.

There are two classes of six-wheel steel passenger cars on the New York Central, the earlier form with equalizers, in connection with Commonwealth frames, and the latter with half elliptic springs over boxes and short equalizers between the springs. The total weight of the car is 142,000 lb. and that of the six-wheel trucks 42,000 lb. With four-wheel trucks the total weight would be 127,000 lb. There are a number of cars of no greater weight running today on four-wheel trucks.

On the 63-ft. 78-passenger cars of the Central Railroad of New Jersey, Commonwealth trucks, with 5½ x 10 journals, are used. The total weight is 115,800 lb. With six-wheel trucks the total weight would be 131,800 lb.

The body of the N. Y., N. H. & H. R. R. 1914-schedule 70-ft. 88-passenger car weighs 89,000 lb. Add the weight of a six-wheel truck, 42,000 lb., and the total weight would be 131,000 lb. In later cars the four-wheel standard type truck was substituted. The heavy four-wheel truck weighs 31,400 lb. and has a 5½ x 10 journal, and it brings the total weight of the car to 121,400 lb. on four-wheel trucks. These cars are giving excellent satisfaction in respect to freedom from hot boxes, and comfortable riding.

The Grand Trunk composite steel frame wood finish car, 74 ft. long, seating 97 passengers, weighs 137,000 lb. Deducting the weight of the six-wheel truck, 40,000 lb., leaves 97,000 lb. Adding the weight of the four-wheel truck, 27,000 lb., makes 124,000 lb. Those figures are well within the capacity of the four-wheel truck.

The total weight of the Union Pacific 69-ft. steel truck baggage car is 106,000 lb. With six-wheel truck it would weigh 122,000 lb.

The Postal 60-ft. four-wheel truck car has a total weight of 111,600 lb., which would be increased to 127,600 with the six-wheel truck.

The Lehigh Valley has a steel well flat car, with a capacity of 220,000 lb. The light weight is 91,900 lb., making the total loaded weight 311,900 lb. The car has a six-wheel truck, with 6 x 11 journals, and yet the ratio of total weight of the car against the projected area of the journal is only 475.

The Santa Fé has a 70-ft. 83-passenger all-steel car which weighs 134,000 lb. and has six-wheel trucks; with four-wheel trucks it would weigh 118,000 lb. The 70-ft. 76-passenger chair car on the same road weighs 136,000 lb. and has six-wheel trucks; with four-wheel trucks it would weigh 120,000 lb. It will be seen, also, that these cars could be properly put on four-wheel trucks.

GEO. W. RINK. I would ask Mr. Thomson whether the type of truck which gave trouble with hot journals was that with the coil spring over the journal box. We had this type of truck[1] in service on twenty-eight 63-ft. passenger cars weighing 101,400 lb. The trucks had $5\frac{1}{2}$ x 10-in. journals and clasp brakes and weighed 14,500 lb. each.

The wheel beam is made up of an eye beam and two channels and the transoms of cast steel riveted to the wheel beam. The elliptic springs are carried in a cradle suspended by U-shaped spring hangers. Pedestals are of cast steel, riveted to wheel beams, and are provided with spring pocket. On trucks of this type it is very essential that holes be reamed and the rivets fill the holes, otherwise there will be trouble in maintenance.

We had a number of cases of hot journals on these trucks when first placed in service. There is a tendency in this construction to so distribute the pressure on top of box as to cause an excess pressure

[1] Railway Age Gazette, August 16, 1912, p. 279.

per square inch on the journal bearing. We also found a strong tendency for the journal boxes to bind in such a manner that they were thrown in or out, due to the action of the coiled spring over the box, resulting in excessive wear of box flange and pedestals. This truck also gave trouble from loose rivets.

We are now using on our 63-ft. steel coaches and combination cars, weighing 116,000 lb. and seating 78 and 51 passengers respectively, a cast steel truck[1] with 5½ x 10-in. axle, and with wheel beams, transoms and end sills cast in one piece. The truck weighs 16,100 lb., and with generator and truck suspension 17,600 lb. These trucks have cast iron pedestals of heavy pattern bolted to the wheel beams, and have the long equalizer with coil spring adjacent to the journal boxes and the same type of clasp brake as used on the built-up truck. We do not use the same type of clasp brake as used by the Pennsylvania Railroad, but use standard brake beams.

These trucks ride very smoothly and seldom have heated bearings. We now have 126 steel coaches in service with this truck giving excellent results, fully complying with the four important features as mentioned by the author—safety, smooth riding, minimum weight, and low cost of maintenance.

Alphonse A. Adler asked whether, in car truck journals, the resistance of friction is dependent on the load, or whether it is possible so to adjust the load that there is perfect film lubrication, in which case the resistance is only proportional to the size of the journal. This will perhaps have some influence in selecting the six-wheel truck in preference to the four-wheel truck if the journal sizes must remain constant.

E. B. Katte said that while they have made several attempts to design a six-wheel motor car truck, they have not arrived at a design which is satisfactory from either a mechanical or an electrical standpoint, and he did not know of any six-wheel motor car trucks in high speed service.

C. D. Young. Railroads have been too prone in recent years to use six-wheel trucks, based upon their experience with four and six-wheel trucks in the days before the advent of the steel truck. Due to the flexure in the wooden trucks it was necessary to go to the six-wheel truck. As far as the probable cost and maintenance are

[1]Railway Age Gazette, Mechanical Edition, December, 1914, p. 627.

concerned, due to the fewer parts, the four-wheel truck is preferable to the six-wheel truck, provided it gives satisfactory service.

With the use of the steel truck the total weight which will be satisfactorily carried on the four-wheel truck car can be materially increased. Ninety-eight per cent of the railroads in this country use four-wheel trucks under heavy passenger locomotive tenders, with axle loads as high as 45,000 lb., yet when a passenger car is designed there is hesitancy about putting 31,000 lb. on the same axle.

To ascertain what effect on the train resistance the two extra axles of the six-wheel truck car would have, we took the same car bodies in a train and compared the resistance of the train when carried on four-wheel trucks, similar to that illustrated in the paper, with that when these cars were carried on the six-wheel standard trucks of the Pennsylvania Railroad. The trains were run at 35, 50 and 65 miles per hour. Three round trips were made and the resistance of the cars was obtained. It was found that 13 six-wheel truck cars offered the same resistance as 14 four-wheel truck cars; in other words, if on a train of 13 cars the equipment is changed from six-wheel trucks to four-wheel trucks, one more four-wheel truck car can be hauled without increasing the resistance over that of the 13 six-wheel truck cars.

The author shows the development of the clasp brake for the four-wheel truck cars, which came about as a result of clasp brakes used on the Philadelphia & Reading Railway. In this connection, two or three points brought out in a series of tests of clasp and single shoe brakes made by the Pennsylvania Railroad, are of interest.

The single shoe brake has a total weight of 3682 lb. per car and the movable parts weigh 3084 lb. The clasp brake has a total weight of 4433 lb. per car, the movable parts weighing 3152 lb., showing an increase in total weight of the clasp brakes of 24 per cent. It was developed in our tests that it was desirable to have as low a weight in the moving parts of the brake rigging as possible to overcome the effect of inertia on the first application of the brakes, for obviously the heavier the moving parts the more inertia and the longer it takes to get full braking pressure at the wheel with a given pressure in the brake cylinder. With equal weights of movable parts, it would be expected that full braking power would be developed as quickly with the clasp brake as with the single shoe brake, but for a given braking power a higher coefficient of brake shoe friction is obtained with the clasp brake car and, therefore, the stops are shorter than with the single shoe brake car.

To illustrate this, at 60 miles per hour with 125 per cent nominal braking power the clasp brake car made a stop in 808 ft., probably the shortest stop ever made on a passenger car under that braking power. The corresponding stopping distance for a single shoe is about 1250 ft., so that with 24 per cent increase in weight of brake rigging there is, with the clasp brake car, a much larger percentage decrease in the stopping distance.

Another point which cannot be ignored is that the use of the clasp brake is economical in brake shoe material. A series of road tests of brake shoes shows a saving in brake shoe material of about 30 per cent with clasp brakes over single shoe brakes.

In reply to Mr. Adler, the friction of the journals of a car running at 50 miles per hour offers only a small resistance compared with the resistance due to wind, flange, oscillation and other causes.

THE AUTHOR. Six-wheel trucks are used extensively and there must be good reasons for this. These reasons have not, however, been brought out in the discussion. It is a severe indictment of the mechanical departments of our railroads if the use of six-wheel trucks under modern railway equipment has simply come about because in the days gone by, when wooden trucks were in use, it was found desirable to use six-wheel trucks in many cases.

Mr. Henderson's discussion brings up a point which might account for the use of the six-wheel truck in the Far West, and that is that where road conditions are poor, there is a possibility that the six-wheel truck may ride more easily. On most of the sections of the roads in the East which still have poor roadbeds, old light wooden cars with four-wheel trucks are used. There is, however, a grade of track between the magnificent roadbed, which we have on the high traffic eastern lines, and the very poor condition; and, generally speaking, on that kind of track the four-wheel trucks ride just as easily as do the six-wheel trucks.

The latest type of standard four-wheel passenger car truck, modified for use under motor cars on the electrified district of the Pennsylvania Railroad at Philadelphia, is illustrated in the paper, and it is interesting to note how skillfully the designers have taken advantage of the standard design and modified it only slightly to allow for the motors, the largest thus far used under motor cars.

No. 1504

MODERN ELECTRIC ELEVATORS AND ELEVATOR PROBLEMS

By David Lindquist,[1] New York, N. Y.
Non-Member

The elevator art has gone through quite a number of more or less radical changes, particularly in the last fifteen years. These changes have been partly due to developments in building construction, utilizing steel as building material, thereby making it possible as well as practical to erect high structures. At the same time, it cannot be denied that the simultaneous development of commercial elevators particularly for high speed, has materially aided in the development of modern buildings.

2 Elevators may be classified according to the driving power employed: steam driven elevators; hydraulic elevators, and electric elevators.

3 The first class, namely, steam elevators, is at the present time practically obsolete. There are, of course, quite a few of them still in existence and running, but there are no new plants of this type being installed at the present time.

4 Hydraulic elevators may be divided into several groups, depending upon the different methods in which the hydraulic power is transmitted to the elevator car. Some of the principal and well-known types are: The horizontal hydraulic (rope geared); the vertical hydraulic (rope geared), and the plunger hydraulic (direct connected). These types have been mentioned in the order they were introduced for high speed in comparatively high buildings. The plunger type practically superseded other types of hydraulic elevators during the period of 1904 to 1907, and in turn the plunger has now been almost entirely superseded by the gearless 1:1 traction type of electric elevator.

[1] Chief Engineer, Otis Elevator Company.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the New York Section.

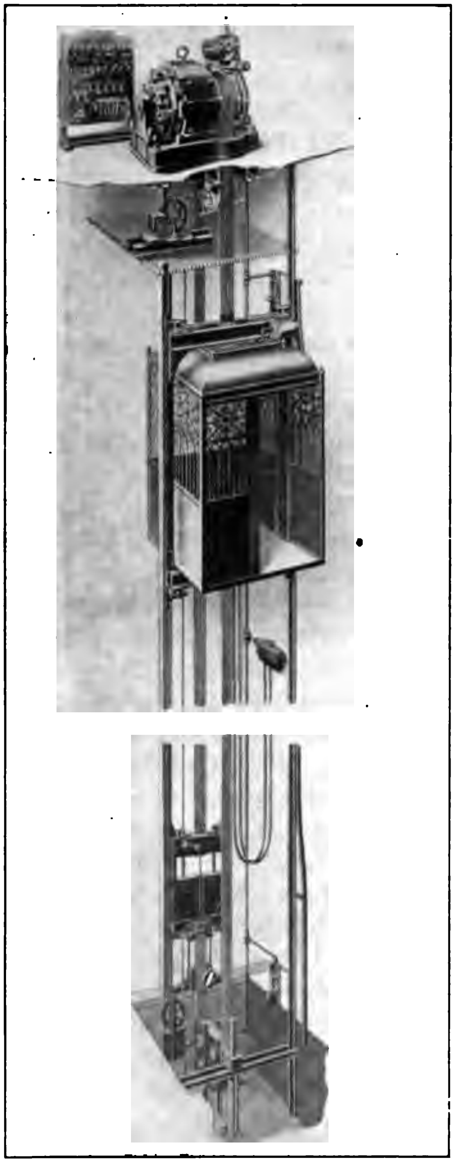


FIG. 1 ARRANGEMENT OF APPARATUS IN A TYPICAL GEARLESS TRACTION ELECTRIC ELEVATOR INSTALLATION

5 Without considering in detail the technical features of the plunger elevator in comparison with the gearless electric traction type, the principal reasons for this change in elevator practice may be summarized as follows:

a Higher initial cost of a plunger installation

b Larger amount of total space in the building occupied by the machinery

FIG. 2 DETAIL VIEW OF THE OVERHEAD INSTALLATION OF A GEARLESS TRACTION ELECTRIC ELEVATOR

c Lower car mileage and consequently more elevators required for the same service

d Higher power consumption

6 The principal features of both the gearless and gear traction type of elevators will be described, the gearless, however, being treated more in detail.

7 *Location of Machine.* When considering the installation of the traction electric elevator the first question which arises is as to where the machine may most suitably be placed. There is no question

but that placing the machine directly over the hatchway, as indicated in Figs. 1 and 2, is the most suitable arrangement. With this overhead location the best results are obtained, and the principal advantages may be enumerated as follows: better traction, less amount of rope, longer rope life, minimum space required, and higher efficiency.

8 It will be perfectly plain, if one stops to consider, that placing the machine directly over the hatchway will impose less load on the building than if the same machine were placed below. Placing the machine directly above the hatchway imposes a load on the building equal to the weight of the hoisting machine plus the loads on car and counterweight ropes, whereas placing the machine below, imposes a load on the building equivalent to twice the loads on the car and counterweight ropes. For a duty of 2500 lb. at 700 ft. per min., the hoisting machine weighs about 16,000 lb., which is considerably less than the total load on car and counterweight ropes.

9 In the above example, the machine, due to high speed, is quite heavy in proportion to the load. A machine intended for lower speed would weigh still less and make the difference in loads imposed upon the building, to which attention is called, even more apparent.

10 Locating the machine above also takes up less space in the building than locating it elsewhere. It also somewhat prolongs the life of the ropes, as by this arrangement the rope is not subjected to so many bends. Placing the machine overhead also increases the overall efficiency of the installation.

11 *Roping.* The next consideration is that of roping. The roping of a so-called gearless 1:1 traction machine located overhead is extremely simple, as indicated in Fig. 3.

12 The principle of the traction drive is no doubt very old but its commercially successful application to an elevator machine, consisting of a slow speed electric motor directly connected to the driving sheave, was first accomplished about ten years ago. Some attempts had previously been made, particularly by Duwelius, who built two, or possibly three, machines. For a number of reasons, however, they were not considered practical and he finally abandoned the attempt, for apparently he could not produce a successful gearless electric elevator machine.

13 At first glance, it would appear as if the traction drive would be rather uncertain, when considering that the ropes are not actually hitched to the driving member, that they simply go round the driving

sheave of the motor, and depend solely upon the friction, or adhesion between the ropes and the driving sheave. This, however, is not the case; on the contrary, it is safer than any other method of drive.

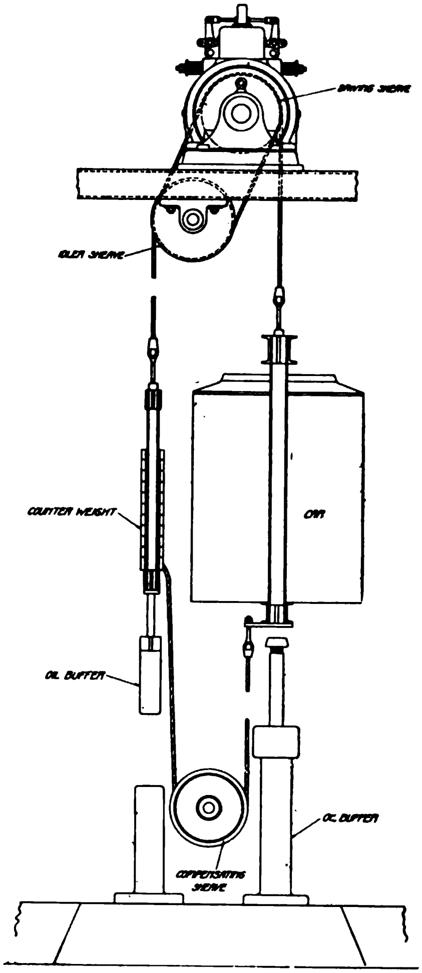


Fig. 3 Details of the Roping of a Typical Gearless 1:1 Traction Elevator

14 Before any of these traction machines were put on the market, a large number of tests were made under such variable conditions as would be encountered in actual installations. It was found that with one half wrap round the driving sheave the maximum traction re-

lation—that is, the relation between the load on the heavy side of the ropes to that on the light side—was 1.56, and the minimum traction relation with one half wrap was 1.4. With two half wraps as indicated by diagram, Fig. 3, the maximum relation is 2.43 and the minimum 1.96, the increase being of course in logarithmic proportion.

15 The maximum was obtained with a new turned sheave with practically new ropes and without any lubrication. The minimum was obtained with smoothly-worn sheave and ropes well lubricated, and with comparatively light tension and high speed. The principal

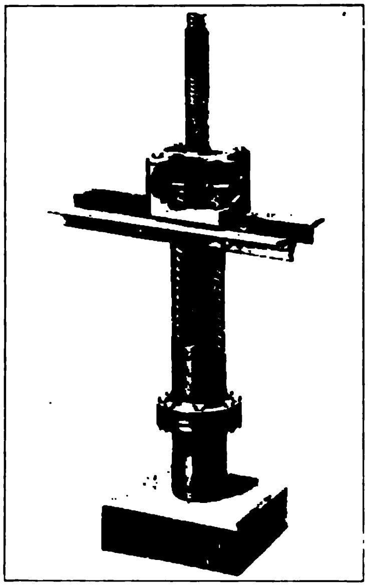


FIG. 4 DETAIL VIEW OF CAR BUFFER OF THE SPRING RETURN TYPE

reasons for this result are that with light tension and high speed there is an oil film between the rope and the sheave; moreover, the rope does not make contact fully 180 deg. with the sheave, due to the stiffness of the rope and centrifugal action.

16 The small variation or the constancy of the traction is quite remarkable, and as long as the maximum variations in the load on a traction type elevator are such as not to require more than the minimum possible traction relation above mentioned, no slipping can occur.

17 Under ordinary conditions a 1:1 traction machine is usually provided with six ropes ⅝-in. in diameter. The material is of soft

Fig. 5
A Traction Electric Elevator Machine of the 1:1 Type

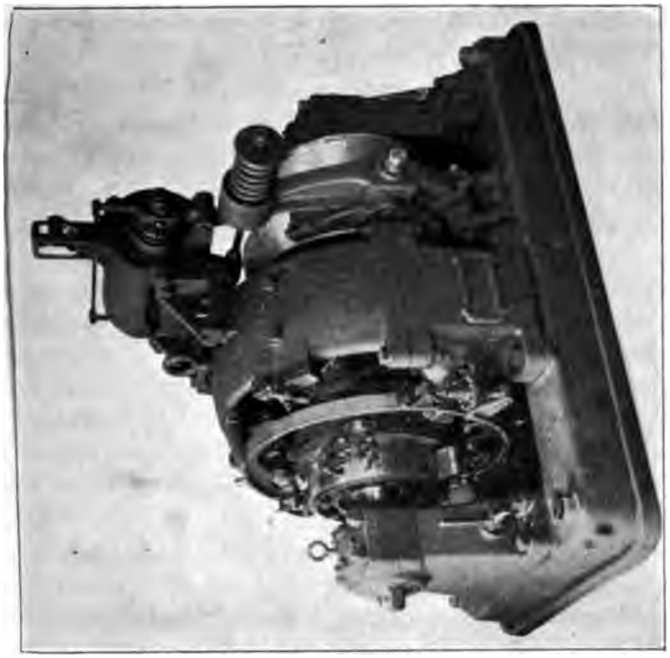


Fig. 6
A Traction Electric Elevator Machine of the 2:1 Type

steel, and in actual installations this will give a safety factor of not less than twelve.

18 Each rope is provided with a self-adjusting swivel rope hitch of the ball and socket type. This permits a gradual creeping and thus prevents any excessive twisting strain, and relieves the rope of the bending strains at the hitch, principally caused by vibration.

19 The traction method of drive has a number of inherent safety features. Traction elevators are arranged so that in case of overrun at terminals, either the car or counterweight bottoms on a buffer, thereby reducing the traction sufficiently to prevent further motion of the car and counterweight, even if the motor keeps on running. The car buffer is usually an oil buffer of a spring return type and mounted in the bottom of the pit, as illustrated in Fig. 4. As shown in Fig. 3, the counterweight buffer is mounted on the counterweight, and acts also as counterbalance. This latter buffer has gravity return, whereas the car buffer has spring return.

20 For very high rises, the great weight of the hoisting rope will cause considerable traction, even after the car or counterweight has landed on its buffer. This traction, together with the momentum of the car or counterweight, may cause either of them to travel into the overhead work, under conditions of runby clearance usually available. To prevent this, some of the exceedingly high rise buildings are equipped at the top of the hatchway with a retarding and latching device for both car and counterweight, in addition to the regular oil buffers acting in the pit. In case of abnormal overrun, the retarder brings either car or counterweight to rest and the latching device prevents subsequent downward movement.

21 *Counterweight and Rope Compensation.* The counterweight, the mechanical construction of which may be seen in Fig. 1, equals in total weight the weight of the car plus usually about 40 per cent of the maximum load. If we consider an elevator of 2500 lb. lifting capacity, 40 per cent of this equals 1000 lb. (the overbalance) and this represents about six or seven persons. Thus, with six or seven persons in the car, giving balanced condition, it is apparent there would be no *net load* to be lifted and the only power required would be for acceleration and for overcoming friction and electrical losses.

22 It is obvious that with a high rise elevator the variation in the net load on the elevator machine, due to the shifting of the weight of the hoisting ropes from one side to the other of the driving sheave as the car moves up and down, would be excessive if this were not

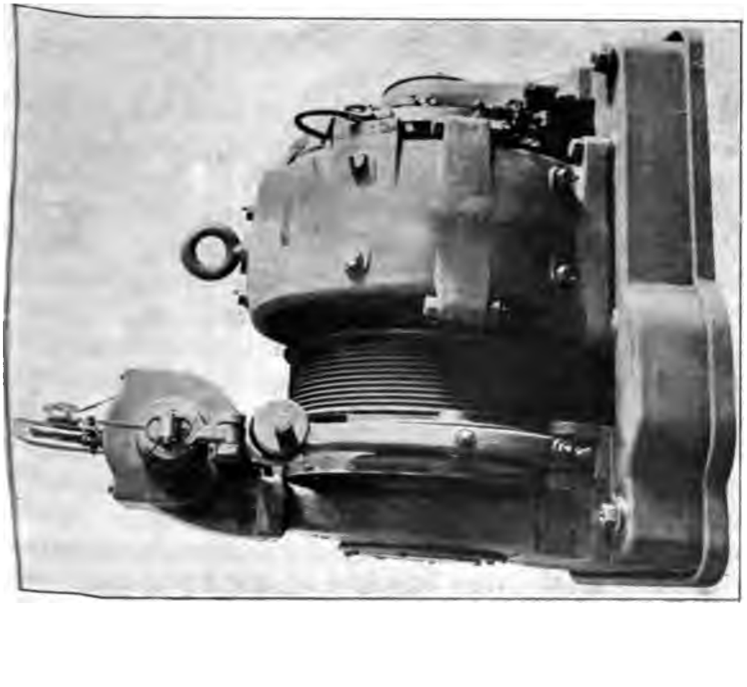


FIG. 7 A TRACTION MACHINE OF THE 1:1 TYPE FITTED WITH BALL BEARINGS

FIG. 8 A SAFE-LIFT TRACTION MACHINE FITTED WITH BALL BEARINGS

compensated for. This compensation is usually obtained by means of chains or ropes attached to the car and counterweight, and running down the hatch in a loop. In Figs. 1 and 3 may be seen the compensating ropes running down from the bottom of the car to the tension sheave in the pit and up to the counterweight. The weight per foot of these compensating ropes is such that they, together with the electric cables (that lead to the car), will compensate the weight of the hoisting ropes regardless of the position of the car.

23 For all high speed high rise elevators, compensating ropes are used. In the pit a tension device is provided for the compensating ropes. For moderate rises and comparatively slow speeds, chains may be used instead of ropes.

24 *Driving Motor.* The electric motor is of the slow speed type, generally six-pole and usually provided with shunt field only. The armature is series-wound with conductors of rectangular cross section, in order to get the maximum amount of copper in the armature. With a 36-in. driving sheave, a car speed of 600 ft. per min. corresponds to 63.6 r.p.m. of the motor. Figs. 5 to 8 show the compact design of the motor, and Figs. 7 and 8 show machines provided with ball bearings, which it may be seen are shorter than the regular babbitted bearings. The principal features of the motor with driving sheave, commutator, electro-mechanical brake, brake shoes, etc., are clearly illustrated.

25 Up to even a comparatively late date, it seems to have been the general impression that a motor of moderate duty, having a speed so exceedingly low as that required for this gearless type of elevator, would have also a low efficiency; but this is not the case. On the contrary, it has been demonstrated, and proven a number of times, that a motor with this low speed can be designed to have just as high efficiency as any high speed motor of equal output. One peculiar feature about the efficiency of this motor is that it is unusually high at light loads. This is particularly of advantage in connection with elevators where the average load on the motor is usually less than one-half its rating.

26 The curve in Fig. 9 shows the result of a test on a traction motor installed in the Metropolitan Life Tower, New York. This motor was rated at 38 h.p., at which full load capacity an efficiency of 89 per cent may be observed. At half load, the motor shows an efficiency of 91.5 per cent and at quarter load over 90 per cent. It

may be said that these are uncommonly high efficiencies for a motor of 38-h.p. capacity, designed for a speed of 58 r.p.m. at 110 volts.

27 The curve in Fig. 10 shows the result of a test of a traction motor installed at the 181st St. Subway Station in New York City. This curve shows that the motor has a maximum efficiency of 93 per

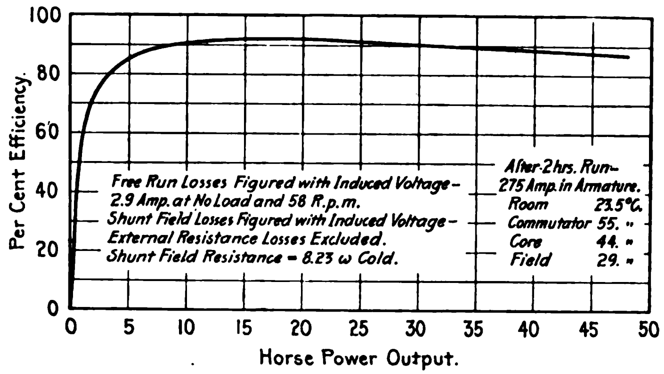


FIG. 9 EFFICIENCY TEST OF A TRACTION MOTOR IN THE METROPOLITAN LIFE TOWER, NEW YORK

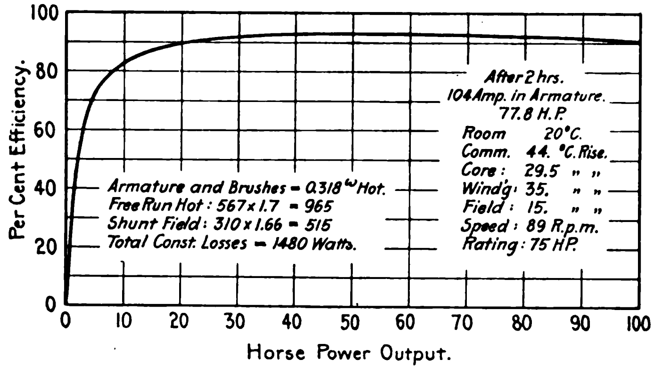


FIG. 10 EFFICIENCY TEST OF A TRACTION MOTOR AT THE 181ST ST. SUBWAY STATION, NEW YORK

cent at about half output and illustrates still better the remarkable results than can be obtained with motors of abnormally slow speed for comparatively small outputs. These efficiencies are, as a matter of fact, considerably higher than are usually obtained in high speed motors of corresponding horsepower; particularly is this true with light loads. Further reference to motor efficiency will be made in connection with power consumption.

28 The driving sheave is commonly about 36 in. in diameter, and the driving sheave and brake wheel are of the rim type, cast integral and bolted to the armature sleeve or spider. Circular rope grooves are employed. The brake is of the shoe type, and the magnet is usually provided with series-winding for quick release, and with a shunt-winding for holding. The brake shoes are provided with a lining of fabricated asbestos. A gradual and soft application of the brake is obtained by a particular method of magnetic retardation of the magnet cores, eliminating thereby the necessity for dash pots. The brake shoes were formerly, for some time, lined with leather, but after very exhaustive tests of a number of different brake lining materials, it was found that a certain kind of fabricated asbestos was the most suitable. The particular quality or characteristic of this brake lining is that the friction between the lining and the wheel is practically constant at all times. Tests have been made and the temperature ran up to about 400 deg. cent., but there was no appreciable variation in the friction. Furthermore, the coefficient of friction is very constant, varying but little with variations in speed. The variation is remarkably small as compared with leather. With leather, it was frequently noticed that the machine would come almost to a stop, and then creep, just as in the case of a hydraulic elevator with a slight leakage of water. This was due to the fact that the friction coefficient between the leather and the brake wheel was very much lower at slow speed than at high, the minimum being obtained at practically zero speed.

29 *Bearings.* The bearings are either of the ordinary babbitt lined, self-aligning type, with automatic chain-oiling, or are of the anti-friction type (ball or roller bearings). Elevators of the gearless traction type have been for some time equipped with ball or roller bearings, which are used for both the main motor, and rope sheave bearings. They were adopted primarily to gain space, and it is readily apparent that these anti-friction bearings take up much less room than the plain solid bearings. It followed therefore that a traction machine equipped with them could be designed to take up less space and thus, in many instances, eliminate the necessity for double decking an installation consisting of a number of adjacent elevator machines; by "double decking" is meant the placing of machines on two levels, when the machine is longer than the center distance between adjacent hatchways, thus preventing their installation on one level.

30 In the case of elevators with 2:1 roping, this saving in space

FIG. 11 TYPICAL OVERHEAD INSTALLATION OF TRACTION ELECTRIC ELEVATOR MACHINES WITH PLAIN BABBITTED BEARINGS

in connection with the sheaves, really becomes a necessity; for h(particularly in the counterweight sheave where space conditions always limited, a plain babbitted bearing of suitable capacity wo be practically impossible. Of course, the use of these anti-frict bearings also introduces other very decided advantages—materi reducing friction, particularly at starting, and introducing a smoc ness of operation superior to plain bearings. This is again parti larly applicable to the sheaves used with machines having 2:1 ropi for here the additional number of sheaves requires that the frict be reduced to a minimum.

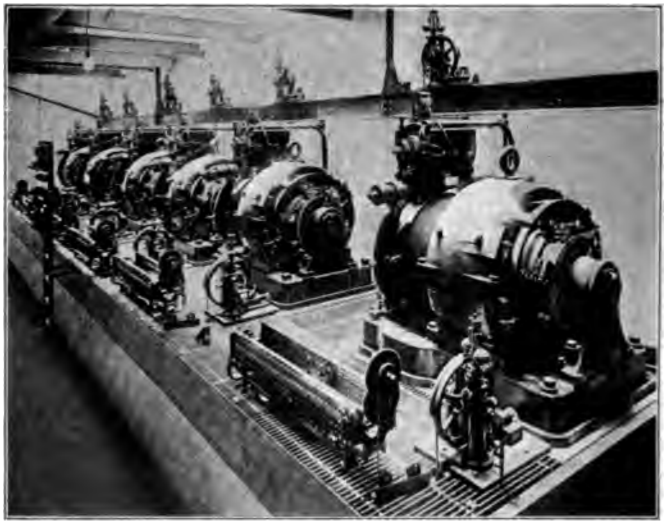


FIG. 12 AN INSTALLATION OF BALL-BEARING TRACTION MACHINES

31 As to the relative merits of ball or roller bearings, opini(differ greatly. Personally, I consider ball bearings, at the pres time, superior to roller bearings for electric elevator machines. reasons for this are based upon considerable investigation which] convinced me that ball bearings can be successfully and practica produced of sufficient capacity to withstand the service impos Further than this, ball bearings possess the inherent advantage being able to run slightly out of alignment without causing destruct strains or excessive friction. With roller bearings slightly out alignment, even though this be not sufficient to set up destruct strains, the friction will be increased very materially. As a matter fact, actual tests have shown that friction induced in this manner (

readily be in excess of the friction in a plain bearing. Furthermore, ball bearings are capable of resisting a certain amount of end thrust, quite sufficient, in the case of these traction machines, to take care of the "float" of the armature, partly due to magnetic action, and added to by the action of the hoisting ropes. Roller bearings will permit of no end thrust at all, and therefore when they are used additional means must be provided to take care of this.

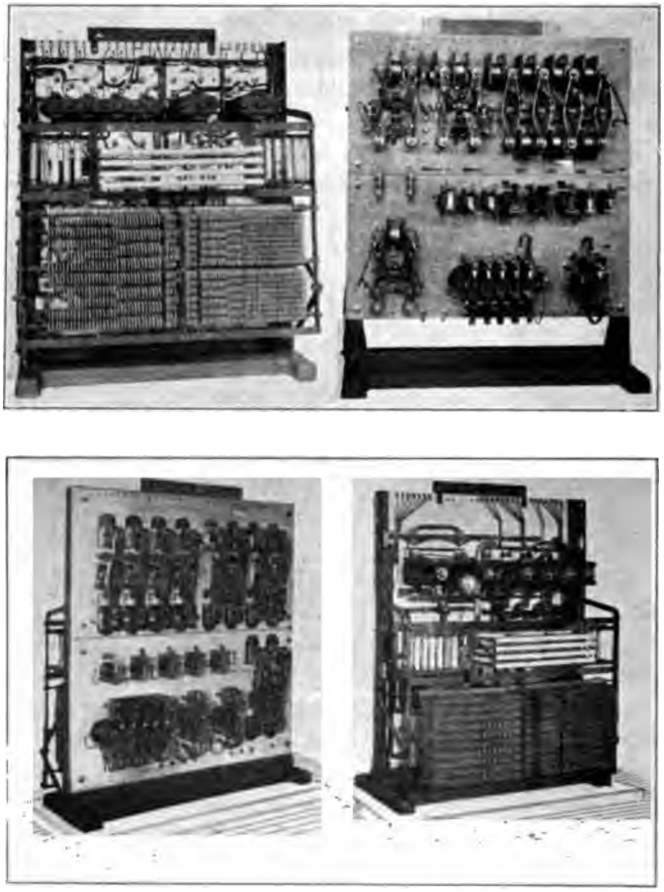
32 The introduction of ball bearings in connection with elevator machinery was something of an innovation, and entailed the use of bearings of considerable size and strength. At the start, the ball bearing manufacturers apparently and unfortunately did not fully appreciate the special features of design that were required, with the result that some of the earlier bearings of this type were not as suitable as they might have been, or are today. However, even taking this fact into consideration, there have been, in proportion to the large number used, very few that have proved unequal to the duty imposed upon them. It is pleasing to record here that while at the start it was supposed that the foreign-made ball bearing represented the highest state of development, today the American product has proved itself by far the best that can be obtained.

33 The total losses in ball bearings may be considered due to three distinct causes, namely, rolling friction, sliding friction and losses caused by the lubricant. The rolling friction loss is that due to the slight deformation of the balls and their races; this loss is exceedingly small. The sliding friction occurs between the balls and their cage, and to a slight extent between the balls and the races; this latter is occasioned by the differing velocities at various points of contact between the balls and the races, due to deformation. The sliding friction between the balls and their cage in a radial type of bearing is also very small, and consideration of the sliding friction between balls and races may usually be neglected unless the difference in velocities is considerable, as may be the case in the use of ball bearings with close fitting races, heavily loaded.

34 Paradoxical though in sound, it should be noted that the principal loss in ball bearings is that caused by the use of a lubricant. This loss, of course, depends upon the viscosity of the lubricant and increases greatly with the speed. It is due partly to the churning of the lubricant and partly to the force necessary to squeeze it out of the path of the balls. The main reason for its use is for protecting the bearings from corrosion. The best protection for slow speed bearings

is obtained by the use of a moderately heavy grease which will n flow. Protection from corrosion is important since a speck of ru may soon ruin a bearing. The lubricating properties also redu wear.

35 *Method of Control.* Figs. 13 and 14 show controllers used



FIGS. 13-14 TYPICAL CONTROLLERS USED WITH TRACTION ELECTRIC ELEVATOR MACHINES

connection with 1:1 and 2:1 traction elevators and Fig. 15, an i stallation of a large group of controllers. The speed variation nece sary in connection with high speed traction elevators is obtain partly by field regulation and partly by series and by-pass resistan in the armature circuit. The field regulation is usually capable of reducing full speed down to a speed from 60 to 40 per cent of full speed; further slow-down is obtained by resistance control.

36 The combination of the two methods mentioned is necessary for obtaining sufficiently slow car speed (about 90 ft. per min.). This slow speed is required for making accurate stops both at intermediate and terminal landings, and also in order to make a very short travel, or to "inch" up or down, to the landing. The exceedingly slow speed automatically obtained when approaching terminal landings is necessary not only to secure accuracy of stops with varying loads, but to provide a fundamental safety feature. It may be mentioned that anti-friction bearings not only materially aid in obtaining smoothness of operation but greatly facilitate the "inching" up to the floor landings.

37 As a matter of fact, the 1:1 electric traction machine with ball bearings has practically the smoothness of motion when running, starting and stopping, so much appreciated in hydraulic elevators; yet it comes to a positive stop without the objectionable oscillation and subsequent creeping, inherent in hydraulic elevators.

38 In connection with the regular operating features of the control apparatus, there are also a number of other features introduced for the purpose of safety. Some of these may be merely mentioned without going into too great detail:

a Automatic return of car switch to "off" position.

b Automatic stopping switch on car for automatic stopping at terminal landings.

c Final cut-out limit switches in hatchway operating independently of the automatic stopping switch.

d Switches operated by centrifugal governor automatically stopping the elevator in case of overspeed; the first switch cuts the power off from the machine and applies the dynamic brake effect to the armature, and applies the mechanical brake to the brake pulley; the second switch applies the light retarding force to the car safety referred to later under the subject of electro-mechanical safety.

e The safety switch in the car under the control of the operator also performs the same function as the two switches operated by the governor.

f Regulation of shunt field by centrifugal governor to maintain constant full speed with variable loads.

g Oil buffers as previously mentioned, capable of independ-

FIG. 15 A LARGE GROUP OF TRACTION ELEVATOR MACHINE CONTROLLERS IN

ently stopping the fully loaded car when descending at 50 per cent excess speed without discomfort to the passengers.

h For high rise elevators the use of a mechanical retarding and latching device.

39 Fig. 16 shows the switch located on the car to effect the automatic stop at terminal landings. This switch is provided with a lever actuated by two cams in the hatchway, one at the upper landing and one at the lower.

40 *Loads and Speeds.* Gearless traction machines, utilizing 1:1 or 2:1 roping, have been built for loads varying from 2000 lb. up to 11,000 lb. at car speeds from 350 up to 700 ft. per min. Of these duties the most generally used for the modern high office building, utilizing 1:1 roping, is about 2500 lb. at a speed of 600 or 700 ft. per

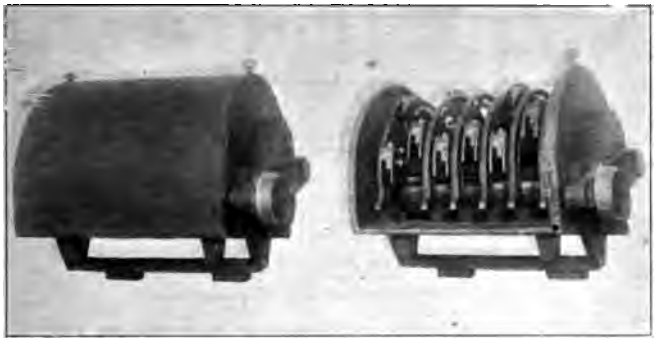


FIG. 16 AUTOMATIC STOPPING SWITCH LOCATED ON CAR

min., although in many instances, a speed of 500 to 550 ft. per min. is found suitable.

41 The speed of the high rise elevators in the Woolworth Building, New York, is 700 ft. per min., which is at present the highest elevator speed permissible in this city. In the new Equitable Building in New York, certain of the elevators are arranged to run a portion of their travel on express service at a speed of 650 ft. per min. and the rest of their travel on local service at 550 ft. per min. The change in car speed is automatically accomplished at the point in the hatchway where the character of the service changes. The point at which this change in speed occurs can readily be altered at any time to suit the requirements of the building. The owners of this building considered that a change in speed as just described would be advantageous, and no doubt this will prove to be the case. Reference to

the curves in Fig. 17 will make the fact apparent that a reduction in speed from that of the express run is desirable when the elevator is running in local service. The advantage is found in a reduction of power consumption without an appreciable loss of service time.

42 The curves in Fig. 17 were produced from tests of the elevators installed in the Metropolitan Life Tower. They show the relation between time and speed, both for express runs and for stopping at every floor; ascending and descending with full load. It will be noted that the maximum speed attained when stopping at every floor is only about 70 per cent of the full speed attained during express

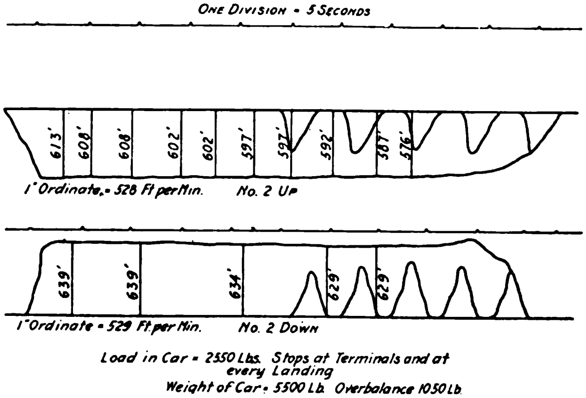


FIG. 17 SPEED CURVE OF HIGH-RISE ELEVATOR IN THE METROPOLITAN LIFE TOWER, NEW YORK

run. This is, however, a rather extreme case as the car was very heavy, the ropes were long and consequently the masses to be accelerated were unusually great.

43 For more moderate speeds and heavier loads, a 2:1 roping of car and counterweight is utilized (shown diagrammatically in Fig. 18), which retains the safety features and general characteristics of the 1:1 equipments. For moderately high buildings, this 2:1 roping is suitable for a car speed not in excess of 450 ft. per min.

44 *Geared Machines.* The traction principle is also applicable to elevator machines employing moderately high speed motors with some form of gearing between the motor and driving sheave. Machines of the geared type are most suitable when lower speeds are involved or where the service conditions are not so severe. Under these conditions the power consumed will be comparatively small on account

of the lesser mileage, and hence the more expensive gearless machine with its reduced power consumption may not be necessary.

45 As far as smoothness of operation is concerned, a geared machine can obviously never equal that of a gearless, on account of the

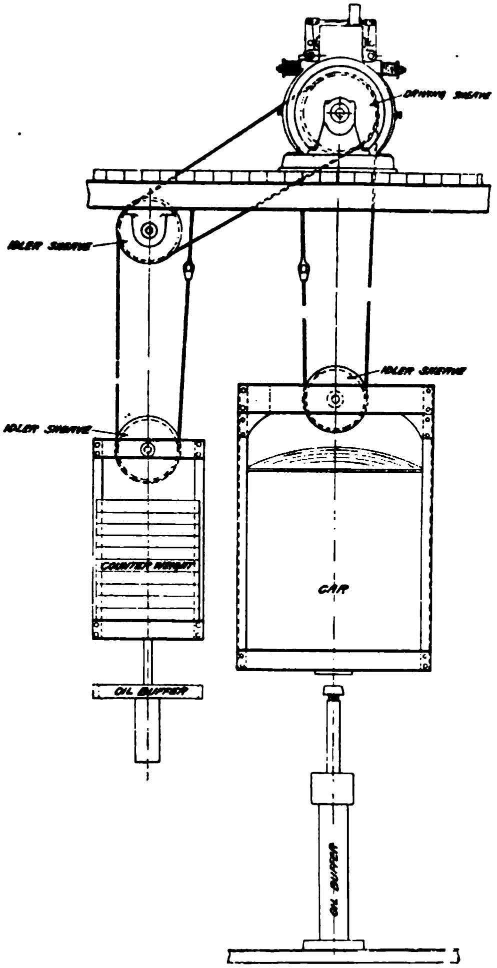


FIG. 18 2:1 ROPING OF A GEARLESS TRACTION ELEVATOR

fact that it is practically an impossibility to manufacture any kind of gears which are perfect, and mount the same in absolute alignment. Furthermore, if this condition of perfection could be established, it could not be maintained in service, on account of the inevitable wear that would take place.

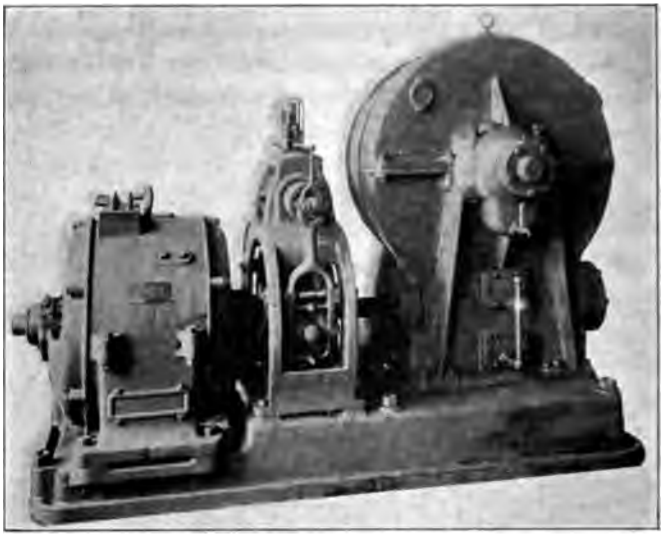


FIG. 19 A GEARED TRACTION ELECTRIC MACHINE USING WORM GEARING

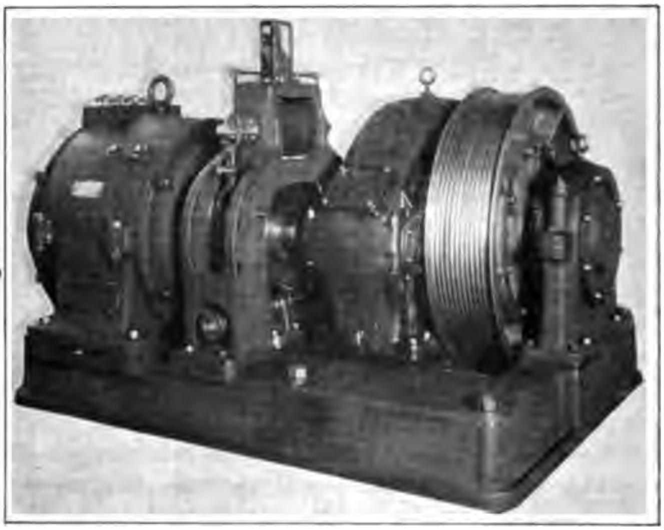


FIG. 20 A GEARED TRACTION ELECTRIC MACHINE WITH HERRING-BONE GEARING

46 Two types of geared machines have been developed, one employing worm gear and the other herring-bone gear reduction (see Figs. 19 and 20). Of these two, the worm-geared is suitable for slow and more moderate speeds, and is used extensively for this purpose. The machine with herring-bone gear reduction is of course not suitable for slow car speeds on account of the difficulty in obtaining sufficient speed reduction. Undoubtedly, it is somewhat more efficient than worm gearing, and has been used with some success in connection with quite high speed elevators. The fact that the herring-bone geared machine has been used for these high speeds should not be understood as indicating that it is equal to the gearless machine, with

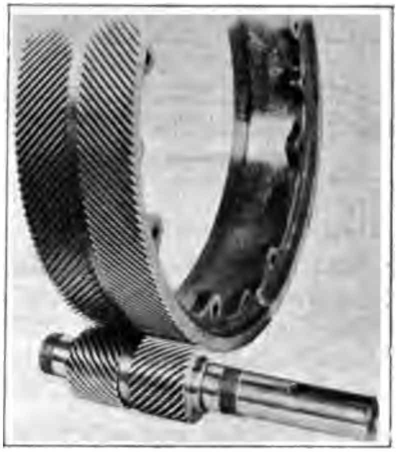


FIG. 21 HERRING-BONE GEAR AND PINION

which it cannot compare as to operating features and power consumption.

47 Of the two kinds of gears mentioned, the worm gear has inherently less tendency to vibrate than the herring-bone gear. On the other hand, the herring-bone gear is generally more efficient than the worm gear.

48 A number of tests of herring-bone gears (Fig. 21) cut by Falk, Fawcus and Otis, have been made, and Fig. 22 shows the results of a series of tests of one particular gear. The difference in efficiency at various speeds is noticeable. This gear has 213 teeth and the pinion 23 teeth, resulting in a gear ratio of 9.27:1. A motor speed of 425 r.p.m gave a car speed of 500 ft. per min. There was quite a difference in the efficiency of the gear at different speeds. The losses included

in figuring the efficiency comprised the bearing losses and losses due to the churning of the oil. The difference in the efficiencies at different speeds is due principally to this churning of the oil, the loss from this source being about 1.5 h.p. at a speed corresponding to a car speed of 500 ft. per min. This appears to be quite a considerable loss, when considering that it is a loss due to the lubricant. If the oil is taken out of the gear case and an amount of oil sprayed on sufficient only to prevent cutting, these efficiencies are somewhat increased, particularly for high speed. Under these conditions, the efficiency is practically the same at different speeds; in other words, the efficiency goes quite a little over 90 per cent.

49 A great number of tests were made also of worm gears with

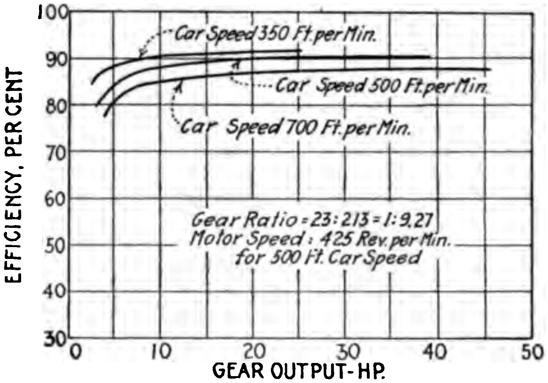


FIG. 22 RESULTS OF EFFICIENCY TEST OF A HERRING-BONE GEAR

from 1 up to 12 threads on the worm, and with a large variation in the number of teeth in the worm gears. With a 12-thread worm and a 108-tooth worm gear, giving a speed reduction of 9:1 or about the same as that of the herring-bone gear, the efficiency was only from 2 to 3 per cent lower, indicating that for high speeds, with comparatively low ratio of reduction, the worm gear is quite efficient. The worm gears with high pitch worms are by no means as inefficient as generally thought. The effect of the oil in this case was practically the same as already mentioned in connection with the herring-bone gear; in other words, at a speed corresponding to 500 ft. per min. car speed, the oil losses were about 1.5 h.p.

50 *Power Consumption.* Considerations of power consumption in the machine may be divided into three parts relating respectively to motor efficiency, gear efficiency and inertia:

a The maximum efficiency of the high speed motor used in connection with the geared machine may occasionally be as high as that of motors used with the gearless, but the efficiency at lighter loads, which is the normal service condition, is lower; hence the high speed motor is at a disadvantage. An equal amount of field regulation may be applied to both high and low speed types of motors.

b High speed motor machines may be considered to have, under the best conditions, a gear loss of about 10 per cent, whereas this loss is entirely eliminated in the gearless machine.

c Although kinetic energy is the most important factor in power consumption, it has in the past been apparently very little considered. The reason for this is due probably to difficulty in readily determining its amount, in a rather complex elevator machine. A method which is very simple and at the same time quite accurate will therefore be described.

51 One of the methods for determining the kinetic energy consists in determining the amount of the inertia of the rotating parts of the elevator by, for instance, the pendulum method, and in ascertaining the mass of the other parts, having straight line motion, by weighing them. This method has a disadvantage in that it cannot be applied to a complete elevator installation. In addition, it is difficult and tedious of execution.

52 A simpler and more accurate method, not affected by the variation of the coefficient of friction and the windage losses at different speeds, may be explained as follows:

53 The kinetic energy of a complete elevator system is the sum of the kinetic energies of each of its constituent parts. In general, these parts run at different speeds, but by reducing all forces and masses to a certain speed, say the speed of the elevator car, the problem can be treated as if involving a single body. If the speed in feet per second of this body equals v_1 when the time equals zero, the kinetic energy ($E = 1/2\ m\ v_1^2$) may be expressed by the equation:

$$E = \int_0^T p v\, dt$$

in which v = the speed at the end of t seconds, p = the retarding force in pounds at the time t, and T = the time in seconds in which the body comes to rest. In this equation pv is the work in foot pounds per second done by the retarding force.

54 If the rate of work is expressed in watts, we have

$$E = 0.7373 \int_{0}^{T} w\,dt$$

in which w represents the retarding watts at the speed v. It will be noted that the quantity under the integral sign is the area comprised between the retarding watt-time curve and the axis of time.

55 To obtain the data for this curve, we may assume the object to be the determination of the kinetic energy of a complete elevator installation at the time when the elevator car has attained the speed v_1. Preferably, the load on the car side of the driving sheave should equal and balance that on the counterweight side. This can be accurately effected, without weighing, by merely taking watt-meter

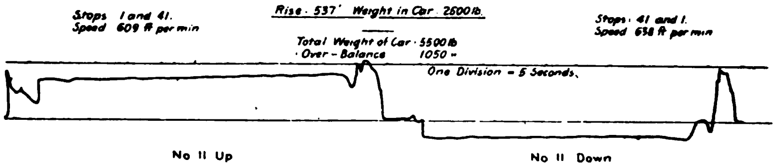


FIG. 23 POWER CONSUMPTION OF A TRACTION ELEVATOR MACHINE IN THE METROPOLITAN LIFE TOWER, NEW YORK

readings for up and down travel at equal speeds, adjusting the load in the car until the up and down readings are equal, indicating that both sides are balanced.

56 The car is then run at the speed v, when the armature current (but not the field current) is interrupted, so that the system may come to rest retarded by the mechanical friction, windage and core losses of the motor. The indicated work in watts due to friction, windage and core losses is then equal to the retarding watts w in the above equation. The speed and time are also observed and a graph of the speed-time curve is drawn.

57 Next, the motor is connected to the line, the elevator run at different speeds, and the watts input observed for each speed. If the I^2R losses represent the losses in the armature winding of a direct current motor and W, the observed watts required to run the elevator at the speed v, evidently then $W - I^2R$ gives the watts necessary to overcome friction, windage and core losses. Consequently, $W - I^2R$ at the speed v is equal to the retarding watts w at that speed.

58 The second series of observations, therefore, provide the data for the graph of the retarding watts-speed curve. Having thus obtained the graphs of the speed-time curve and the retarding watt-speed curve, it is a simple matter to construct the retarding watt-time curve. The quantity W may then be obtained by means of a planimeter and, with the aid of the above equation, the kinetic energy at speed v_1 may readily be calculated.

59 The retarding watt-time curve is so nearly a parabola that only an inappreciable error results from considerating the curve as such. Instead of obtaining the area representing wdt by means of a planimeter, it is sufficient to employ the equation of a parabola, whereby this area is found to be one-third of the product w_1T; w being equal to w_1 when the speed $v = v_1$ which occurs when time $t = o$.

60 When, therefore, extreme accuracy in the determination of the kinetic energy is not required, it is sufficient to obtain merely two observations or readings; one being the time T in seconds in which the elevator comes to rest after the armature current is interrupted at the speed v_1, and the second being w_1 in watts, which is equal to the watt-input W required to run the elevator at the speed v_1 minus the I^2R losses.

61 If it is required to determine the kinetic energy of the hoisting machine only, the car and counterweight may be blocked, and the tests made as before. By removing from the machine part after part, and making tests after each removal, the kinetic energy of each of the various moving parts may be obtained.

62 It will be noted that the method outlined above refers to direct current machines; however, with necessary modifications in regard to the determination of w, the retarding watts, this method may be applied to alternating current machines also.

63 An idea of the relatively high cost of operation of electric elevators utilizing high speed motors with gears may be gained by means of the following example: Assume a weight of 1 lb. revolving at a radius of 9 in. at 800 r.p.m. (which is the speed usual with motors for worm gear machines). The kinetic energy to be imparted to this weight each time it is started from rest and brought to full speed will then be 61.3 ft-lb.

64 If the elevator starts on an average of 4500 times per day and runs three hundred days per year, the cost of the energy absorbed for acceleration, at 4 cents per kw-hr. would amount to $1.25 per lb. per year. If, instead of one lb., the weight were 219 lb. (which is the

effective equivalent weight of a particular herring-bone gear machine), the cost would be $273.00 per year.

65 Of course, part of the kinetic energy required for starting is returned at the time of stopping, but on the other hand the starting efficiency is low, since a large amount of the power is lost in the starting resistance and some in the motor. The loss in the starting resistance and motor is certainly larger than the energy recovered at the stop, and hence the cost per year for accelerating and retarding 219 lb. of weight, under the above conditions, would amount to more than $273.00.

66 In ordinary passenger service the load taken up and distributed to the various floors of the building is practically the same as the load brought down to the starting point, therefore, the electric power consumed is not primarily power required for the transportation of passengers, but is the work required to overcome friction and to supply the kinetic energy for each start, less the amount that is recovered at the stop. The friction losses are by far the smaller of the two. Therefore, a comparison of the kinetic energies of two elevators, for the same service and with the same method of control, gives a good indication of what their relative power consumption and rapidity of acceleration will be in actual service.

67 In the following tabulation are given the kinetic energies of two traction machines, one with herring-bone gear and the other gearless, both built for a car speed of 600 ft. per min., and for the same load. The herring-bone gear machine had a motor running at 505 r.p.m., a pinion with 23 teeth, gear with 213 teeth, and driving sheave 42 in. in diameter. The gearless machine had a motor running at 63.6 r.p.m. and a 36 in. driving sheave.

		Kinetic energy in ft-lb.
Herring-bone gear machine	Motor armature	7,420
	brake pulley	2,920
	driving sheave	1,000
	herring-bone gear	2,060
	Total	13,400
Gearless machine	Armature, brake pulley, driving sheave — Total	2,450

68 It will be noted that the kinetic energy per start of the herring-bone machine would be about 5.5 times that of the 1:1 gearless

machine. Consequently, if the cost of power for accelerating and retarding the 13,400 ft-lb. of the former amounted to \$273.00 per year, the cost for the gearless machine would be but $\frac{\$273}{5.5}$ or \$50.00.

69 It may not be at once apparent why the high speed motor with armature of much small dimensions, although running at high speed, should have such high kinetic energy, compared with that of a slow speed armature of much heavier weight and larger dimensions. A simple example may be given illustrating this fact.

70 Take a motor of certain construction and field strength, and assume that on a 220-volt circuit it will develop 10 h.p. and run at 100 r.p.m. On a 440-volt circuit, this same motor with equal armature current and with shunt field arranged to maintain the same field strength, will develop 20 h.p. and run at 200 r.p.m. That is, the voltage being doubled, the horsepower and speed will be doubled; and, the kinetic energy, varying as the square of the speed, will be quadrupled. If, then, we assume the kinetic energy in the first case to be 500 ft-lb., that in the latter case will be 2000 ft-lb., and the kinetic energy per horsepower of the motor running at 100 r.p.m. will be $\frac{500}{10}$ or 50 ft-lb., and that of the motor running at 200 r.p.m. will be $\frac{2000}{20}$ or 100 ft-lb. In other words, in this example, doubling the speed will double the kinetic energy per horsepower output.

71 As already indicated, the kinetic energy of a complete herringbone gear machine was approximately 5.5 times the kinetic energy of the gearless. This increase is not directly proportional to the speeds of their respective motors because the ratio between the kinetic energies of the driving sheave, brake and gear coupled to the motors, is considerably less than proportionate to the motor speeds. If, however, the armatures only are considered, the fact would be disclosed that in this case the kinetic energy is more than proportional to the speed because, on account of the gear losses, the geared machine requires a motor of greater capacity than does the gearless machine.

72 At this point, it may be interesting to observe the efficiencies and power consumption of some actual installations. The curves in Fig. 23 show recording wattmeter readings taken in the Metropolitan Life Tower of an elevator carrying a load of 2500 lb. up and down. The upper curve indicates the watts taken when lifting the load and the lower curve the watts required for starting, and the watts generated back into the line, when the load is descending.

73 The curve in Fig. 24 shows the efficiency of a complete installation of a gearless traction elevator in the New York Telephone Building, at Walker and Lispenard Sts., New York. It may be noted

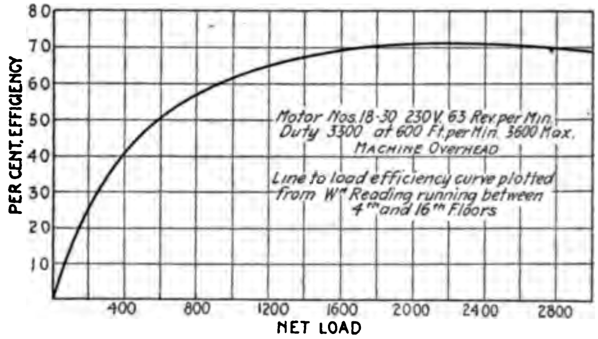


Fig. 24 Efficiency of Complete Gearless Traction Elevator Installation—New York Telephone Building

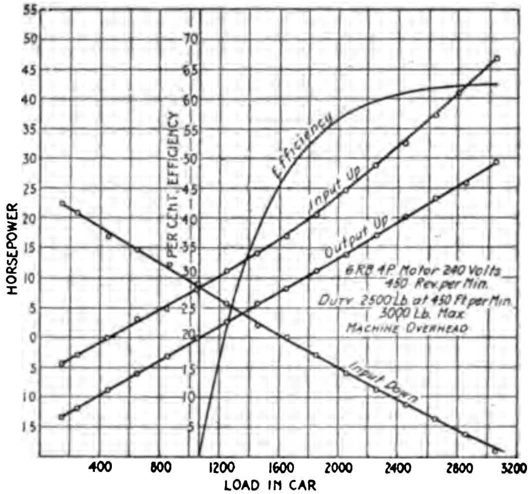


Fig. 25 Efficiency of Complete Herring-bone Gear Traction Elevator Installation—Yale Club Building, New York

that the maximum efficiency from line to load, allowing for all losses, is over 71 per cent. The curve in Fig. 25 shows the efficiency of a complete elevator installation of a herring-bone gear machine in the new Yale Club Building in New York City. It may be noted that the maximum efficiency from line to load is about 62 per cent.

74 The curves in Fig. 26 show the results of recording wattmeter readings taken at the Adams Express Company's Building, New York City. The loads indicated are net loads, that is, loads representing the unbalanced difference between weights on the car side and weights on the counterweight side of the driving sheave; and the ordinate at zero indicates the balanced condition of these weights.

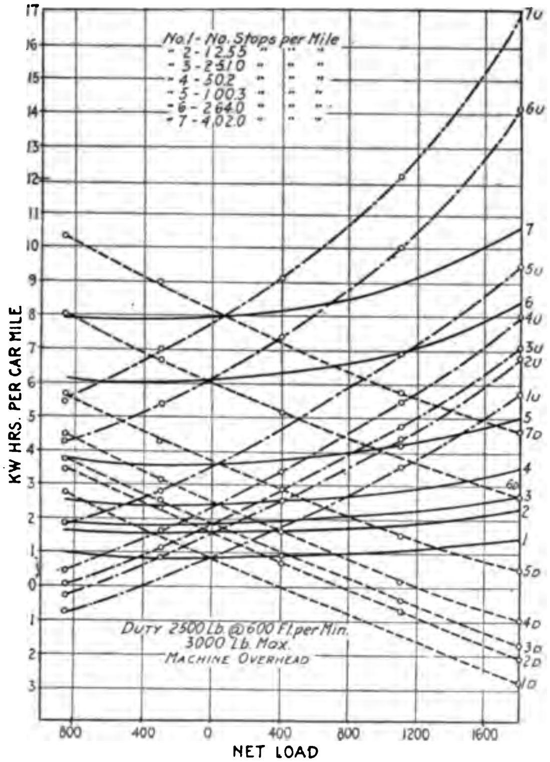


Fig. 26 Power Consumption Test of a Traction Elevator Machine in the Adams Express Co. Building, New York

The broken curves, ascending from left to right, represent power consumption while lifting unbalanced weights. The broken curves, descending from left to right, represent power consumption while lowering unbalanced weights. The full line curves, drawn between the former, represent the average power consumption required for lifting and lowering weights. Curves are shown for various numbers of stops per car mile, ranging from no stops to 402.

75 The average amounts of power consumption, for various loads and numbers of starts, are plotted in curves shown in Figs. 27 and 28. These curves indicate the increase in power consumption occasioned by an increase in number of starts while transporting various loads, Fig. 27 representing express and Fig. 28 local service. By referring to curve 2 in Fig. 27, it may be noted that, with a net or unbalanced load on the car side of the driving sheave of 500 lb., the power consumption would be 0.8 kw. per car mile when no stops whatever are made, and about 3.2 kw. when there are 100 stops. The net load of 500 lb. represents a load in car amounting to 1500 lb. partly balanced by a 1000 lb. counterweight. By comparing the curves one with another, it may be noted that the power consumption increases with the number of starts to a greater extent than with a corresponding increase in net load. Comparison between curves in Figs. 27 and 28 will indicate that the power consumption for equal loads and number of starts, is greater for the express service than for the local service, due to higher speed and greater masses.

76 A number of observations have been made of the actual service conditions encountered in different buildings. Some of the most interesting are those of the Hudson Terminal Building in New York City, where the service conditions are usually severe and varied. The curves in Figs. 29 to 32 record these conditions, indicating the number of cars in service, the power consumption in kilowatt hours, the miles traveled, the number of cars leaving the ground floor, the number of stops, etc., during hours from 8 A. M. to nearly 6 P. M. Both express and local service for certain groups of elevators at the 30 Church St. and the 50 Church St. (both Hudson Terminal) buildings are represented.

77 The curves in Figs. 33 to 35 record the movement of passengers during different hours of the day; the upper curves showing the number of passengers carried up and the lower curves, the number carried down. These curves well illustrate the tremendous inrush and exodus of passengers during short periods of time, morning, noon and evening; as well as the normal traffic between times.

78 The curves in Fig. 35 illustrate the most severe "rush-hour" service encountered. It may be observed that at about 8.30 in the morning, there is a tremendous inrush for about twenty minutes, which tapers off to a slight rush later on; around ten o'clock, or halfway between the rush hours, there is about the same amount of service in both directions. Then just at twelve o'clock, sharp, there is a

tremendous rush outward, which is all over in ten minutes. In other words, all these passengers have had to be taken out of the building in ten minutes, the same number being handled from 12.20 on, during the inrush which, however, lasts for about twenty minutes. Then, during the afternoon, may be noted the continuation of traffic up and

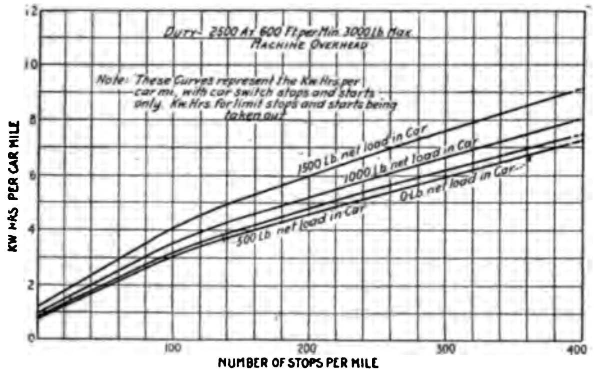


FIG. 27 AVERAGE POWER CONSUMPTION FOR VARIOUS LOADS AND NUMBERS OF STARTS—EXPRESS SERVICE—ADAMS EXPRESS BUILDING

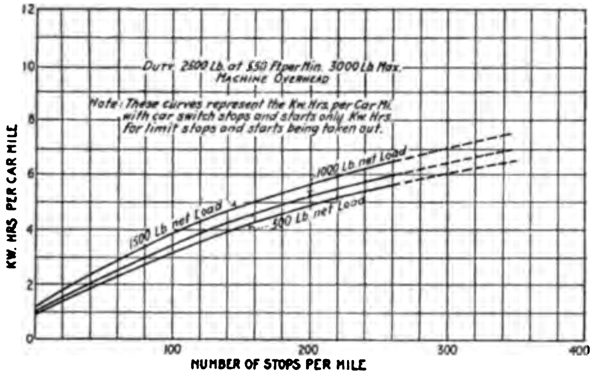


FIG. 28 AVERAGE POWER CONSUMPTION FOR VARIOUS LOADS AND NUMBERS OF STARTS—LOCAL SERVICE—ADAMS EXPRESS BUILDING

down, in about equal amounts. The promptness with which the people leave the building at night may be observed by referring to the curve in Fig. 35.

79 These charts show service conditions, indicating what tremendous variations of traffic are encountered in elevator service. In order to prevent congestion in a building the elevators must be capable of taking care of at least half, or say about two-thirds, of all the

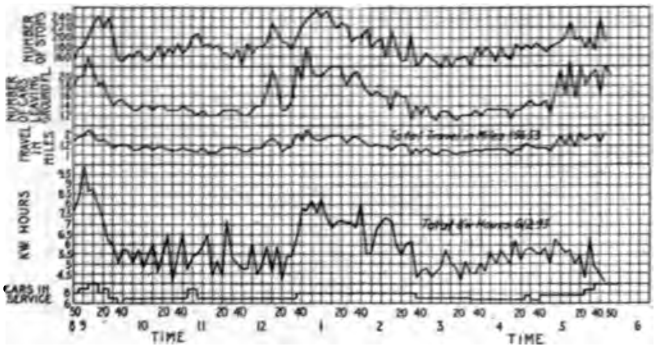


Fig. 29 Express Service

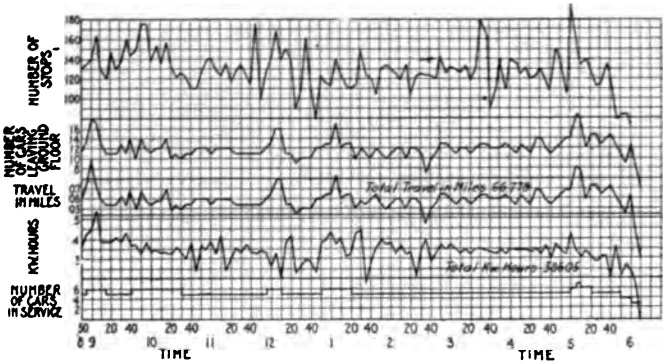


Fig. 30 Local Service

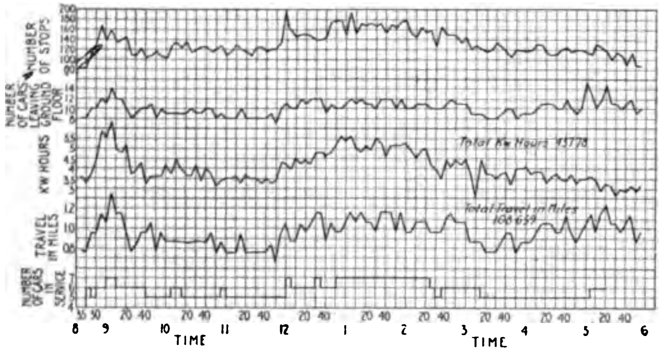


Fig. 31 Express Service

Charts of Elevator Service at the Hudson Terminal Building, New York City

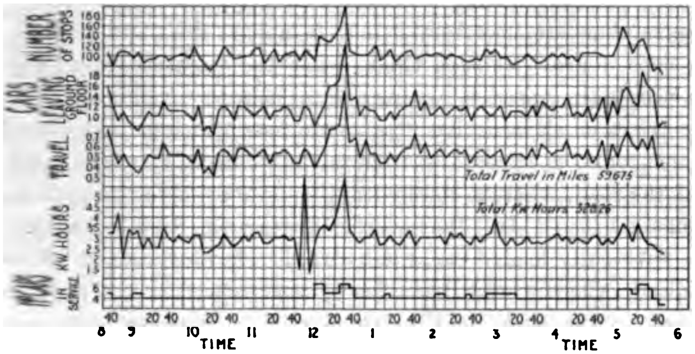


Fig. 32 Local Service

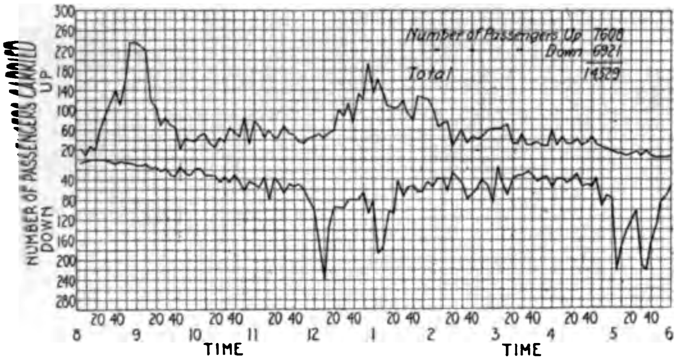


Fig. 33 Express Service

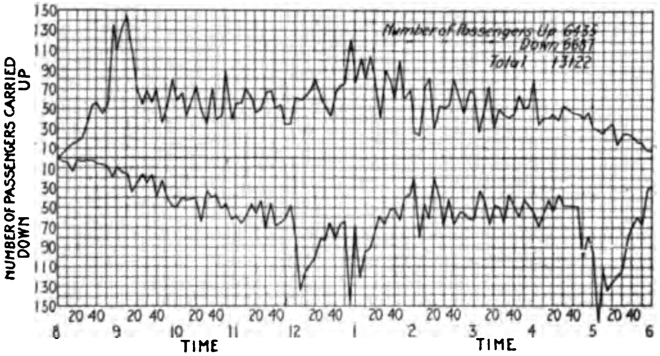


Fig. 34 Local Service

Charts of Elevator Service at the Hudson Terminal Building, New York City

people in a building during practically only ten minutes. The curves indicate further how necessary it is to be able to maintain high speeds during rush hours. At such times, when the elevators have to carry all the people up, speed is an important factor. In this respect, the electric traction elevator has the advantage over the hydraulic elevator, in that it is capable of maintaining constant speed independently of the load, whereas the hydraulic elevator varies its speed with variations in load. In other words, during the time of maximum service, with fully loaded car, the hydraulic elevator runs at its minimum speed, whereas with light loads, when speed is not important, the speed is maximum. Undoubtedly, this indicates one of the principal reasons why the electric has gradually, and now almost entirely, supplanted the hydraulic elevator.

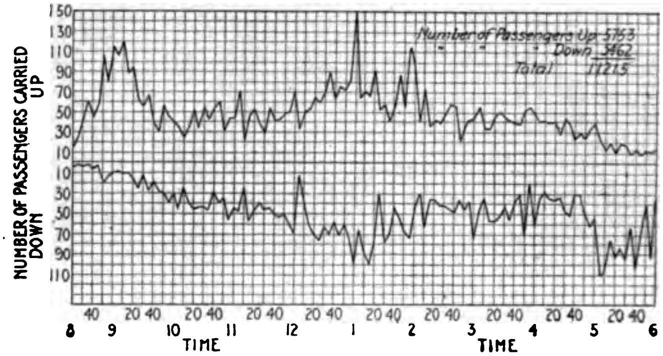


FIG. 35 EXPRESS SERVICE (SEE FIGS. 29 TO 34)

80 *Electro-Mechanical Elevator Car Safety.* In conclusion, mention may be made of various elevator car safeties, with particular reference to a new safety that has lately been put on the market. The essential requirement of an elevator car safety is to stop and hold an elevator car under all conditions of load and speed, whether the hoisting ropes are intact or parted. Fig. 36 illustrates the safety device usually employed until a comparatively short time ago, and which, even at the present time, is still installed. It is the standard so-called wedge clamp safety, which clamps the rails for retarding and holding the elevator car in case of overspeed or free fall.

81 The present practice utilizes the same principle of application whether the safety is actuated under conditions of excessive speed with hoisting ropes intact or during free falling. A very strong retarding force is required to stop a free falling car, but this strong retarding

force, if applied with ropes intact, is excessive and will give an unpleasant shock to the passengers. This is obvious when an actual installation is analyzed.

82 The car, with all attachments, of a modern high rise elevator, weighs about 6500 lb., the maximum load is usually 3000 lb. and the compensating ropes weigh about 2500 lb.; the total load to be stopped by the safety with the ropes parted would therefore be 12,000 lb. To insure the stopping of a car, at least 50 per cent excess force should be

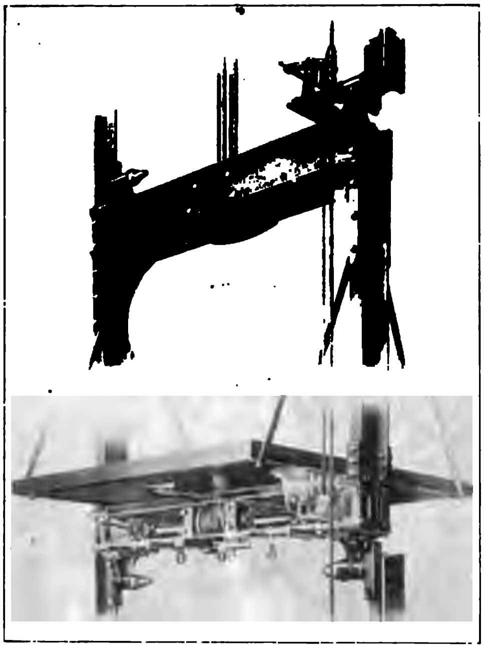


Fig. 36 A Typical Application of the Standard Wedge Clamp Safety

provided, bringing the total retarding force required up to a minimum of 18,000 lb. Were a major portion of this retarding force applied with the ropes intact, the car would be stopped too abruptly. This would be accentuated with a light load, particularly as the machine brake and strong dynamic action of the motor assist the retarding force of the safety.

83 For the above mentioned reasons, it is essential, therefore, that the safety should apply a *strong retarding* force in case of a free falling car with ropes parted, and a *light retarding* force in case of overspeed with ropes intact. The principal safety features which

should be incorporated in an elevator safety for high speed elevat
are as follows:

a The safety should be so arranged that the application o
predetermined and definite *light retarding* force will s

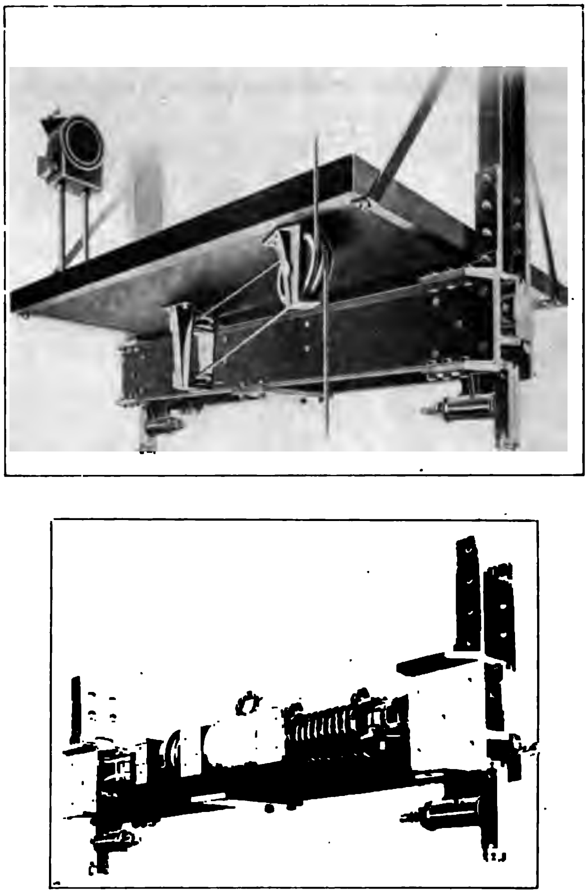


FIG. 37 DETAIL VIEWS OF AN ELECTRO-MECHANICAL WEDGE CLAMP SAFE

the car and net load without shock in case of oversp
with hoisting ropes intact.

b The safety device should be so arranged that the applicati
of a predetermined and definite *strong retarding* fo
will gradually bring the car and maximum load to rest
case of a free falling car.

c The *light retarding* force should be immediately applied, preferably by means of a centrifugal governor, in case the car should attain excessive speed in either direction.

d It should be possible to apply immediately the *light retarding* force from within the car, when desired.

e The *light retarding* force should be applied automatically in case of overrun at the upper or lower terminals, and yet be so arranged as not to interfere with the starting of the car in the opposite direction.

f The *strong retarding* force should start to apply the instant the hoisting ropes part, independently of the speed of car and counterweight.

g A tripping governor should not be necessary to apply the *strong retarding* force, except on safelift machines.

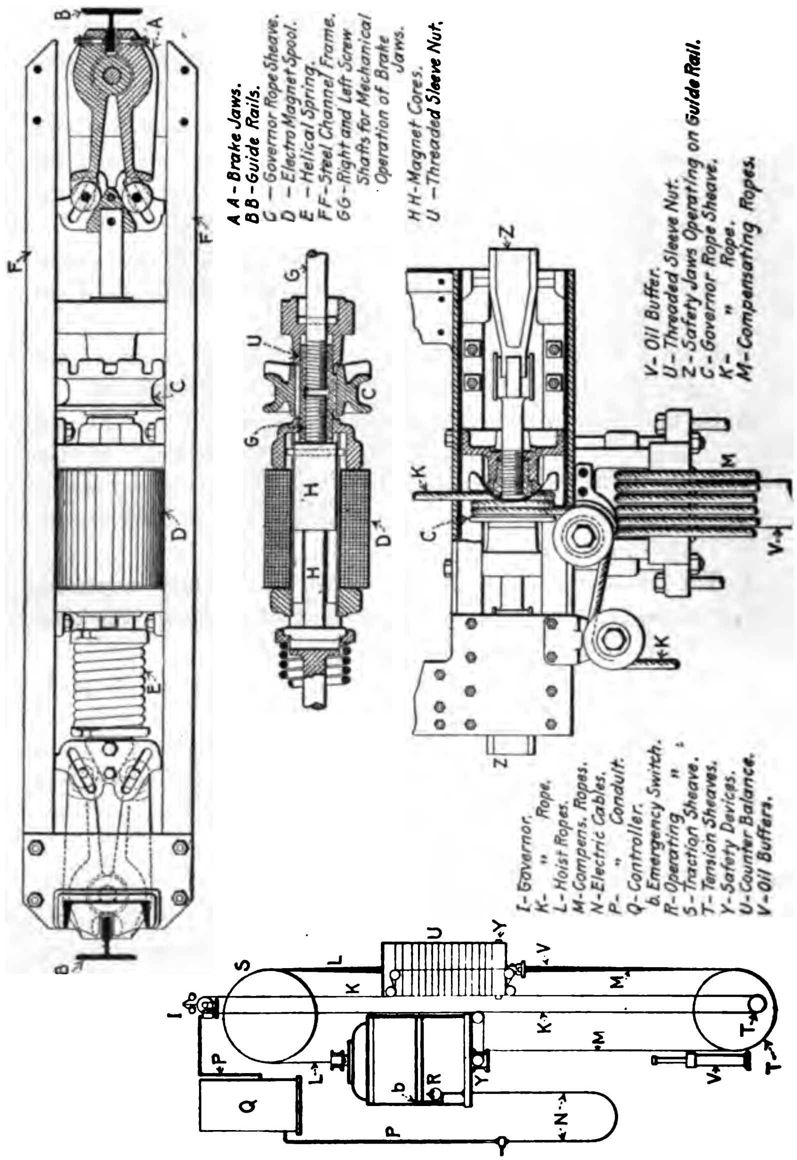
h In the case of safelift machines, a *strong retarding* force should be automatically applied independently of the parting of the hoisting ropes, at a definite speed which should be higher than the speed at which the *light retarding force* is applied.

i The releasing carrier, even when improperly adjusted, should not prevent the application of a *strong retarding* force to the car in case the ropes part.

j The principal actuating parts of the safety should be made to move automatically at frequent intervals, in order to prevent them from clogging up or corroding together; this motion of the actuating parts need be only very small, to give the desired results, but some motion is necessary to secure dependable action of the safety.

84 Figs. 37 and 38 show the general features of an electro-mechanical elevator car safety which embodies all of the above described requirements. The *light retarding* force is obtained by means of a helical steel spring forcing the wedges between the rollers of the safety jaws. When the car is in service this spring is held under compression by means of an electro-magnet. The instant the current in this magnet is interrupted, either by the centrifugal governor, safety switch in car or limit switches in the hatchway operated by the car or counterweight, the *light retarding* force is applied.

85 The *strong retarding* force is obtained by revolving the safety drum, thereby actuating the right and left hand screw, which in turn moves the wedges and forces the jaws against the guide rails. The


A A - Brake Jaws.
B B - Guide Rails.
C - Governor Rope Sheave.
D - Electro Magnet Spool.
E - Helical Spring.
F F - Steel Channel Frame.
G G - Right and Left Screw Shafts for Mechanical Operation of Brake Jaws.
H H - Magnet Cores.
U - Threaded Sleeve Nut.
V - Oil Buffer.
U - Threaded Sleeve Nut.
Z - Safety Jaws Operating on Guide Rail.
C - Governor Rope Sheave.
K - " Rope.
M - Compensating Ropes.
I - Governor.
K - " Rope.
L - Hoist Ropes.
M - Compens. Ropes.
N - Electric Cables.
P - " Conduit.
Q - Controller.
b. Emergency Switch.
R - Operating " "
S - Traction Sheave.
T - Tension Sheaves.
Y - Safety Devices.
U - Counter Balance.
V - Oil Buffers.

strong retarding force is definitely determined by the number of turns around the safety drum, which vary in number according to the maximum load and the amount of the safety rope tension weight.

86 The releasing carrier is placed on the counterweight and will therefore apply the *strong retarding* force of the car safety, even though the releasing carrier fails to release, due to faulty adjustment. The *strong retarding* force only is applied to the counterweight safety.

87 Whenever the current is cut off from the elevator, the electrical parts of the safety are operated, thereby applying the *light retarding* force. The mechanical actuating parts of the safety effecting the *strong retarding* force are continually kept in slight motion when the elevator is in operation, due to the different stretching of the hoist and governor ropes. It is possible to remove all parts without disturbing the car platform, enclosure or car frame, thereby assuring easy access for inspection. In case of safelift elevators, a *strong retarding* force as described in paragraph *h* is applied by means of a parallel jaw double acting governor.

88 An electro-mechanical safety device such as just described has been designed and built by the Otis Elevator Company and subjected to a long series of experimental tests, the results obtained being remarkably successful. Some forty-nine elevators in the new Equitable Building, New York, have been equipped with these safeties and the consulting engineers, the builders and the New York Building Department have thoroughly and exhaustively tested them with entire and complete satisfaction to all concerned.

DISCUSSION

This paper was originally presented at a meeting of the New York Section on April 13, 1914. The complete discussion at this meeting is published in THE JOURNAL, June, 1915. Excerpts from this discussion bearing directly on the statements made by the author are given below.

ANDREW M. COYLE took exception to the statement by the author regarding the relative efficiencies of the direct driven traction machine and the gear driven machine. He said that while no one doubts that for light duty and high speed the direct driven machine is more efficient, there is an immense field in which the geared machine is highly efficient and in which the direct driven machine cannot be used at all. It is, therefore, not altogether fair to make the comparison based on a car speed of 600 ft. per min. The geared machine using herring-bone or helical gears is highly efficient for all duties and speeds from 200 to 400 ft. per min., and this service covers the great bulk of elevator installations.

He also took exception to the statement that helical gears are only slightly more efficient than worm gearing. A worm gear is a special application of the wedge as the load is carried forward upon an inclined plane. The load which can be carried upon a single gear is limited, as the working surfaces have only line contact, and when the load is enough to force the lubricant from this line, abrasion immediately sets in. To overcome this difficulty, the tandem worm gearing has been designed, by which the load is divided between two gears and the line of contact increased.

Worm gears doing heavy or continuous service must be completely submerged in oil, and a certain percentage of the power expended in churning the oil. The power thus expended is constant for any given gear. When the load is light, the power expended represents a considerable percentage of the whole work. On the other hand, when loads are light, the surface lubrication better, and better results are obtained in the thrust bearings.

The herring-bone or helical gear is the most efficient gearing so far developed for elevator or any other service. These gears when properly cut are practically noiseless. The slight tendency to wear is at the points of the teeth, and tends to throw the load toward the pitch line where sliding friction is least.

The efficiency of the worm gear is limited by the fact that the load must slide from five to ten times the distance it is lifted, and the efficiency is entirely dependent upon lubrication. In the case of helical gear, the sliding friction is almost negligible; the gears do not wear and cause backlash as in the case of worm gears, and the lines of contact and the strength of the teeth may be made sufficient to take care of any desired load.

An elevator machine using worm gearing assumes an outline which lends itself readily to installation in buildings, and for this reason, as well as on account of low cost, this type of machine in some cases to be recommended. In point of efficiency, however, the helical gear is entirely preferable.

Robert Johnson called attention to a condition frequently met with, but not touched on at all in the paper, namely, the use alternating-current in elevator operation. The conditions referred to would, he said, hardly apply to an installation of alternating-current machines with regard to reliability or speed, and also as to the application of the brake slowly.

This, he said, was a very important point, and he thought it would be very interesting to know whether alternating current is entirely successful in connection with high speed elevators, particularly in view of the fact that in certain parts of New York City alternating current is the only kind available.

H. F. GURNEY and C. R. CALLAWAY[1] wrote that the author's curves of efficiency of herring-bone gears apparently indicate that the gears have an efficiency of only 88 to 91 per cent—values far below those regularly obtained in good practice, which are found to be from 98 to 98.5 per cent, as determined by the writers' tests. These figures are substantiated by numerous other tests made by competent engineers, showing that, when properly designed and manufactured, gear reductions of this type may confidently be expected to show an efficiency of 98 to 99 per cent.

The author also said that worm gearing was preferable to double helical gearing because it offered greater certainty of smoothness of operation. Emphatically this is not the case, and in any elevator with a machine employing well-made gears of this type, it is impossible to detect their presence, while on the other hand it is a well-known fact that worm gearing very often causes a trembling motion of the car which it is difficult to remedy.

In answer to the implication that gears should not be used if possible to get along without them, a well-known manufacturer of apparatus using herring-bone geared drives has stated: "While there is a general and natural objection to the introduction of gears where they can be avoided, yet here, as in all branches of engineering design, the advantages and disadvantages of one form of construction have to be balanced against those of another before a final judgment can be passed. In other words, the use of gears eliminates certain undesirable features; therefore, if gears can be made that will perform the duty, with low cost for maintenance, high efficiency and requiring little attention, is not their use the logical procedure?"

There is evidently some discrepancy in the figures for accelerating energy in Par. 67, as it is apparent at a glance that the 1:1

[1]Supt., Gurney Elevator Co., New York.

armature with its traction sheave and large brake pulley will require far more accelerating energy than the much lighter gear and sheave, running at the same number of revolutions per minute. These figures actually should be about 2200 ft-lb. for the gear and sheave, and about 3800 ft-lb. for the complete 1:1 machine.

Some of the advantages of the double helical geared machine, as compared with the 1:1 machine, may be summarized as follows:

a Less space occupied in the building.
b Less expensive supports and pent-house construction.
c No traveling crane needed.
d More efficient motor design possible.
e Commutation better.
f Mechanical brake less troublesome.
g Control of car speed secured entirely by field-weakening, against wasteful armature resistances.
h Car runs at a definite speed when running slow, insuring more accurate stops.
i Higher line to load efficiencies.
j Less energy required per start and stop.
k Lower cost of operating on account of smaller energy consumption per car mile.
l Less rope wear.

THE AUTHOR. Up to the present time no alternating-current elevators of the direct connected or gearless traction type have been put upon the market. Those that have been put on the market have been of the geared type, for speeds up to 350 ft. per min. approximately. For 250 to 350 ft. speed, two-speed alternating-current motors are employed, with a speed variation of from 1 to 3 down to 1 to 4. The slowing down is obtained by rearranging the connections of the motor in such a way as to change from a small number of poles, giving the high speed, to a large number of poles, giving the slow speed. During this change of speed, power is actually generated back into the line, in a similar manner as in a direct-current machine when the motor field is strengthened for the purpose of speed reduction.

There was at first considerable difficulty in eliminating an objectionable shock due to reducing the motor speed to one-quarter in one step, but by proper design of motor and controlling apparatus, this was successfully accomplished.

The gearless 1:1 traction machine is intended primarily for high speed, high rise service. For this service, on account of its uncommonly high efficiency (especially with average or less than half load), low power consumption and satisfactory operation, it far excels all other types of elevator machines yet developed. It is not intended to supplant the geared type of machine in its particular field of comparatively low speed and rise. In other words, the interests of clients are best served by having both types available for choice, depending upon a careful consideration of particular requirements. Manifestly, there is a field intervening that of the gearless 1:1 traction and the regular worm geared type of elevator which may be covered by either of them. When, therefore, the load required is heavy, the speed desired is between 250 to 450 ft. per min., and the service is severe—making a reduction in power consumption an important consideration, justifying a reasonable increase in first cost of plant—the use of a gearless 2:1 traction machine is most appropriate. This 2:1 type of machine combines the advantages of high efficiency (with average as well as full loads), comparatively low kinetic energy and consequent low power consumption with requirements for lower speed and corresponding heavier loads.

Exception has been taken to the figures in Par. 67. These were intended to be indicative of two particular machines suitable for the same load and speed. The herring-bone gear was large in diameter and of substantial construction; its weight resulted in the high energy of 2060 ft-lb. This result, as well as all the other figures given, was derived from careful tests made in the manner described. Since it appears of interest, a similar tabulation of the complete installation is:

	Kinetic Energy in Ft-lb. (600 ft. Car Speed)	
	Herring-bone geared Machine	Gearless Machine
Motor armature	7420	2450
Brake pulley	2920	
Driving sheave	1000	
Herring-bone gear	2060	
Secondary and compensating sheaves	1865	1865
Car and counterweight on balanced load condition	15,500	15,500
Ropes	880	880
Total	31,645	20,695

The ratio between these totals is as 6.13 is to 4. As the use of the herring-bone geared or the gearless machine does not entail material differences in the design or layout of car, counterweight or other hatchway units, a comparison of the differences in the machines only was deemed most representative of their relative merits for elevator service and possible effect on power consumption.

No. 1505

TURBINES VS. ENGINES IN UNITS OF SMALL CAPACITIES

By J. S. Barstow,[1] Philadelphia, Pa.
Non-Member

The term "units of small capacities" as used herein is intended to include steam turbines and engines of less than 500 h.p. capacity. The paper will necessarily deal largely with the prime movers of auxiliary apparatus in power plants, since the tendency of the times in all industries, and particularly in central stations, is toward the concentration of power in a few units of large size and uniform capacity as opposed to a multiplicity of small units of different capacities. However, there is a wide field for power application where the steam-operated prime movers are of relatively small size, and where transmitted or central station energy is not able to successfully compete; and it is intended to discuss the type of apparatus best suited in these cases, as well as the type of apparatus which it is advisable to employ for auxiliary units in large plants or central stations.

2 There are certain definite fields where the small turbine is of conceded superiority, and other fields where the engine must hold sway. The desirability of the one as compared with the other is largely determined by the following factors, which govern the adaptability, cost and economy of the equipment to be installed for any given service:

- *a* *Speed conditions and limitations* involving consideration of maximum or minimum permissible speed, and whether the driven apparatus is of the constant or variable speed class
- *b* *Steam pressure and temperature conditions* involving consideration of initial and final pressures, and superheat, if any

[1]818 Pennsylvania Bldg., 15th and Chestnut Sts.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Philadelphia Section.

c *Power capacity of apparatus*

d *Relative space requirements of turbine and engine units* involving consideration of available room, character of power house construction and cost of foundation or other supporting structure

e *Use or application, if any, of the exhaust steam* for feed water heating, steam heating or process purposes

f *Available cooling water supply,* if the turbine or engine is to run condensing, involving also consideration of the temperature of the water and whether it must be artificially cooled and re-circulated

g *Operating conditions* including consideration of attendance, oiling, starting and stopping, vibration, noise, etc.

h *Relative cost of complete installations* including necessary foundations, piping and condenser equipment, if any

3 As to the practicability of the small turbine, it may be said that not until about 20 years ago was any really practicable apparatus of this kind developed, and even up to ten years ago the turbine was looked upon mainly as an experiment. The last few years have witnessed, however, the practical perfection of this type of prime mover in sizes as large as 50,000 h.p., with units of 30,000 h.p. quite common in large central stations. They have also seen quite as much good work done in the perfection of small turbine units as in the development of very large ones, and the turbine in all sizes is quite as well developed today as is the steam engine after more than one hundred years of constant effort and improvement.

4 The writer has no intention of reflecting on the great work done by steam engineers in the development of the reciprocating steam engine. The work of these men, which was the harder task, accelerated the evolution of the turbine, the science of thermodynamics underlying the final development and perfection of both.

5 The present day builders of reciprocating engines are able to report further progress, however, and as a result of their recent efforts there have appeared the rejuvenated poppet valve engine adapted to the use of highly superheated steam; the unaflow or parallel-flow engine, also with poppet type valves, in which cylinder condensation is reduced by causing the steam to travel in one direction only; and the small self-contained power plant, or "locomobile," consisting of a steam engine mounted upon a tubular boiler and operating at high superheat. All these engines are adaptations from European practice,

where, owing to the high cost of fuel and the relatively larger number of technically-trained operaters, they have found high favor.

6 Mention also should be made of the power plant formerly used in the White motor car, which, owing to its high steam pressure, superheat and speed, gave remarkably low steam consumptions. In a series of tests, reported in the Transactions of the Society, vol. 28, pages 598-9, the average steam consumption was 12.7 lb. per delivered hp-hr., operating at 850 r.p.m.; steam at 275 lb.; superheat, 340 deg. fahr.; non-condensing exhaust.

7 Tests have recently been reported by the manufacturers which give the steam consumption of a 115 h.p. Buckeye-mobile, running at 248 r.p.m.; steam at 210 lb.; initial superheat, 171 deg. fahr.; non-condensing, as 13.3 lb. per ihp-hr. This unit produced an ihp-hr. on 1.33 lb. of coal having a calorific value of 14,500 B.t.u. per lb. A 169 h.p. unit running at 200 r.p.m.; steam at 209 lb.; initial superheat, 218 deg. fahr.; vacuum 25.7 in., showed a water rate of 9.2 lb. per ihp-hr. The coal consumption, using fuel with a calorific value of 14,209 B.t.u. per lb., was 1.08 lb. per ihp-hr.

8 While there is the possibility that the use of a shell type of boiler for high pressures will be considered by many engineers as unwise, especially for installation in densely populated centers, it is still too early to predict what the future of this apparatus is to be. However, as indicating what may be accomplished in fuel economy in a well-designed plant of small size, the results reported are interesting.

SPEED LIMITATIONS

9 The question of speed limitations is of first importance in selecting the type of prime mover. Since high peripheral velocities are necessary in order to efficiently utilize the energy of a steam jet in the turbine type of apparatus, the latter shows its lowest water rate when running at a constant high speed. Where the character of the service is such as to require speed variation or reversal in direction, or where the speed is necessarily low, the turbine is unsuited and the engine is much better adapted.

10 In engine installations, the minimum permissible speed has an important bearing on the question of operation. If an engine is run at very high speeds, operating troubles are sure to be numerous, the upkeep excessive and the service unsatisfactory. The lack of driven apparatus designed to run efficiently at speeds consistent with high turbine economy has, in the past, frequently dictated the use of engine prime movers for many kinds of work.

11 Although speed reduction gears are by no means new, having been used with the turbine almost from the beginning of its commercial development, improvements in high speed gearing, as well as in the manufacture of high speed direct connected generators, blowers and pumps, running at 3000 r.p.m. and above, have greatly increased the possibilities for turbine installations. Direct current generators as small as 10 kw. capacity, and 60-cycle alternators of capacities as low as 150 kw., designed for gear drive, are now offered. The manufacturers claim for these machines that the increased efficiency of the higher speed turbine, together with the saving effected in the generator construction by reason of the slower speed permissible in the driven end, justify the expense and complication which the gears introduce.

12 While many may still prefer a direct connection, the increasing popularity of the gear drive, especially for direct current generators, blowers and pumps, would seem to indicate that the gears are here to stay, and that when properly constructed and installed, there is no valid objection to their use.

13 For power station work, where some of the auxiliaries are usually motor driven, the exhaust steam can be entirely condensed in the feedwater heater, and the water rate of the steam driven auxiliaries is not a limiting factor, while reliability, accessibility, low maintenance and labor costs are of more vital importance. Power station designers have always exhibited, therefore, a strong preference for turbo-auxiliary units, and there is now a decided tendency toward geared installations.

14 The selection of turbines for auxiliaries is largely influenced by the high speed at which small engine units are run, which makes it exceedingly difficult to keep them in continuous service, and almost impossible to secure smooth, quiet operation. Such reciprocating units require close attention, and must be shut down, overhauled and adjusted at frequent intervals; the maintenance is high and serious breakdowns are by no means rare. An accident to a circulating or hot well pump, for example, usually necessitates a shutdown of the main generator, with consequent loss of production, and often, in the case of a public utilities plant, loss of prestige and the incurrence of public ill-will. In all central stations, therefore, where the main units are few in number and of large size, high economy as compared with continuous operation becomes relatively unimportant, and turbine driven circulating hot well and boiler feed pumps are almost invariably used.

15 Motor driven exciters are largely used in alternating current stations, a steam driven exciter being provided for starting up and as a reserve unit. Here also, since the steam driven set is idle a large part of the time, high economy is not so important and the saving of floor space and elimination of vibration will often decide the question in favor of the turbine. In fact, in all direct current generator sets of 50 kw. capacity or less, the high speed necessary for the engine generally makes it undesirable.

16 For driving fans of large capacity at low pressures, say less than 1½ in. of water, for induced draft, hot air heating and ventilating systems and the like, engines seem best suited. Fans built for this service ordinarily run at less than 200 r.p.m. and are usually of the paddle-wheel type, which is better suited to these conditions than the multi-blade high speed fan. In induced draft work, load fluctuation may require frequent changes in speed, the engine being under the control of a throttling regulator which is automatically actuated by a change of steam pressure. These conditions are quite unfavorable to turbine economy, and a suitably designed and well-constructed engine will give more satisfactory results.

17 In like manner, where we consider the pumping of large quantities of water against low variable heads (conditions which are encountered in drainage or sewage pumping stations) the turbine must yield place to its rival. These pumps may have single runners of large diameters or may be of the multi-impeller type, but in either case, the speed is below the economical turbine range, even when a gear is used, while the engine, at low speed, has an opportunity to make its best showing, steam economy and operating troubles considered. Where the lift is variable, speed changes are required and the engine is almost always more suitable.

18 As a typical installation may be mentioned the four 76-in. centrifugal pumps for the Plaquemines and Jefferson Drainage District of Louisiana. Here the pump speed varies from 50 r.p.m. for 1 ft. head and 135,000 gal. per min. to 115 r.p.m. for 13 ft. head and 90,000 gal. per min. The best duty was 92,600,000 ft. lb. per 1000 lb. of dry steam, corresponding to 21.4 lb. of steam per water horsepower obtained at 87 r.p.m. for 7 ft. head and 130,000 gal. per min. The prime mover is a compound engine operating at 170 lb. steam and 25.7 in. vacuum.

19 The use of underfeed stokers, operating under heavy forced draft and capable of developing high boiler ratings, has become quite

common as a means of reducing fixed charges and boiler banking losses in railway and lighting plants, and particularly those maintained as standbys to hydro-electric stations. These furnaces often carry in the air duct pressures as high as 6 in. of water, and the high speed multi-blade fan makes the better installation, particularly where one fan serves several boilers, as the blower units frequently become excessively large when run slower than 400 r.p.m. At this speed the engine drive is an uncertain and expensive proposition.

20 Furthermore, as such stokers at best are capable of only from one-quarter to one-third their maximum capacity under natural draft conditions, a blower breakdown under a peak load is a serious matter and the ability of the turbine to stand up under the conditions imposed justly entitles it to the preference which it is accorded.

21 For driving directly connected alternators, a frequency of 25 cycles fixes the maximum speed at 1500 r.p.m., which is too low for the best turbine performance. In 60-cycle apparatus, where a speed of 3600 r.p.m. is possible, the turbine shows to better advantage.

STEAM PRESSURE AND TEMPERATURE CONDITIONS

22 The highly economical steam turbine must necessarily be operated condensing, but, as previously pointed out, there are many cases where high steam economy is not the most important consideration and the non-condensing turbine often finds favor over the steam engine. One of the large heat losses incurred in the steam engine is that due to cylinder condensation and one of the common methods of reducing it is to limit the range of temperature through which the steam is allowed to work in a single cylinder. For this reason simple engines are better adapted for low steam pressures, while compound and triple expansion engines are advisable for high pressure and high temperature ranges, particularly if the load is uniform.

23 The engine as a rule develops mechanical troubles with high superheat, especially where the steam valves have much travel under unbalanced pressures. The consensus of opinion seems to be that for slide or gridiron valve gears, a temperature of 400 deg. fahr. to 425 deg. fahr. should not be exceeded, while the best point for the Corliss type engine will be found below 450 deg. fahr. Above these limits lubrication is unsatisfactory and distortion of the parts is apt to give trouble.

24 In European practice, superheating is much more common than in this country, a superheater being considered quite as indispensable as a feed water heater and the poppet valve engine, which

was first perfected abroad, is accordingly better suited for high superheat conditions than the type of gear commonly used in American engines.

25 A difficulty sometimes encountered with engines using high superheat is the warping of the cylinder, the curvature being caused by the higher temperature which prevails in the metal next to the steam chest. Precaution is taken by some builders to avoid such trouble by leading the steam by two independent pipes, entirely separate from the cylinder barrel, from the throttle directly to the steam valves.

26 For power plant auxiliaries, it would appear that turbines which experience little difficulty from high temperatures will be more and more widely adopted while engines will be less commonly used, especially so as steam pressures and superheats are constantly increasing, it being not unusual for new plants to be designed to carry from 200 to 225 lb. with superheats of 150 deg. fahr. and over. With steam engines running under high vacua, above 27 in., the great volume of steam to be handled increases the size of the engine cylinders, and the size of ports through which the steam must pass, to such an extent as to make the engines very expensive if not of impracticable construction. The cost of an engine which would permit of complete expansion to such a terminal pressure, together with the increase in cylinder condensation, due to the greater range of temperature, would make the high vacuum undesirable.

27 The turbine, on the other hand, can be designed to operate on very low terminal pressures with comparatively slight increase of cost; its action as a heat machine is such that a greater expansion can be utilized and the economy is greatly improved by any increase in vacuum. When run non-condensing, as is well known, the turbine is less economical than the non-condensing engine.

28 In plants where the exhaust is atmospheric and cannot be applied to any useful purpose, the engine best fills the conditions, provided space is available and the speed may be made low enough to ensure smooth, quiet running. Such an application is found in direct current generator sets of 100 kw. capacity and larger, in hotels, office buildings and hospitals, where the exhaust is used for steam heating in winter and must be wasted for several months in the year.

POWER CAPACITY OF APPARATUS

29 The lower cost of large turbine units and the greater reliability of this kind of apparatus in regular service, coupled with the

smaller space taken up by turbines as compared with engines, has practically put the engine out of the running as far as large power plants are concerned. Where 60-cycle apparatus is installed and condensing units are used, the engine has no field beyond the 500 kw. mark, while with direct connected 25-cycle apparatus, the engine must stop beyond the 1000 kw. limit, and with the perfection of high speed reduction gears, it is doubtful if 25-cycle engine driven generators can compete with turbine apparatus of even 500 kw. capacity. The reduction gear is also rapidly driving the engine from the direct current field in units of all sizes, above, say, 200 kw. capacity. At the same time, elimination of operating troubles by the use of direct connected turbines for exciter purposes is fast causing the turbine to supplant the engine for this service.

30 In the case of non-condensing units where moderate speeds are required, the engine must continue to hold the field, though special conditions may make the non-condensing turbine a factor to be considered. In this connection, one installation might be mentioned where a belted turbine of 750 h.p. capacity, running at 1500 r.p.m., is used for driving the constant speed shafting of a paper mill, it being contended that the greater uniformity in rotative speed secured by the turbine results in fewer breaks and a more satisfactory product. In this case, the exhaust steam is, of course, utilized in the dryers of the machine, and the variable speed power is supplied by direct current motors.

RELATIVE SPACE REQUIREMENTS

31 Owing to the freedom from reciprocating motion, the foundations required for turbines are of small size and light weight, there being little vibration to be absorbed when the alignment and balancing are well done. The small sizes can be safely operated on floors of usual construction, designed for the ordinary floor loads. There is no difficulty experienced with the transmission of vibration to the structural members of the building or to the piping system.

32 The small space required for the installation of a turbine gives it an advantage in water works plants, operating against moderately high heads. The vertical triple expansion engine, which was formerly used almost exclusively for such work, requires a strong massive substructure to absorb the shock and distribute the weight, and a deep pit to accommodate the water end. Where foundation or other construction difficulties are encountered in this work, the cost may easily climb to a high figure, and in a case under the author's

personal observation the additional cost of building, incidental to the use of the vertical triple engine, would have more than paid for a turbine driven centrifugal pump, while the fixed charges on the pump alone were more than four times the cost of the fuel required to run it to full capacity ten hours per day.

33 The turbine driven pump is not capable of showing on test the high duty of the vertical triple engine—which is one of the most economical steam engines—but the great difference in the first cost of installation often makes the turbo set decidedly preferable, especially when the saving in building is considered. A geared turbine unit to pump 100 million gallons per day against a 56-ft. head was recently installed in Ross Station, Pittsburgh; the pump speed was 350 r.p.m. and that of the turbine 3600 r.p.m. On the duty trial, with steam at 151 lb. and vacuum 28.38 in., the pump showed a performance, including power consumed by the auxiliaries, of 120.5 million ft. lb. per 1000 lb. of dry steam, corresponding to 16.44 lb. of steam per whp-hr. Six similar sets, ranging in capacity from 6½ to 30 million gallons per day capacity, are now in process of construction. The displacement of the reciprocating engine from a field where its superiority was formerly unquestioned, shows the substantial progress which has been made in the development of the steam turbine and the centrifugal pump.

34 For boiler feed pumps of more than 250 gal. per min. capacity, the turbine is often used, and on account of its small size, usually results in a neater and more compact layout. Where regulation by throttling is unnecessary, and the pumps run at or near capacity, the economy as compared with the direct acting type is good and can be better maintained. Valve renewal and packing troubles are avoided. The overload capacity of the centrifugal type, however, is small and the delivery of the pump must be proportioned to meet the maximum demand, not the average boiler horsepower requirements. In the smaller sizes, the cost of turbine units is high; where the load fluctuates widely and the speed must vary, the economy is poor and it is better to install reciprocating pumps.

35 In the modern plant containing large turbo-generator units, space limitations in the basement arrangements are an important consideration. With the high vacua carried, large volumes of water must be handled and the turbine drive for circulating, condensation and air removal pumps is in many cases the proper selection. Such condenser sets have a compact arrangement, especially when a single

turbine is used to drive all the pumps, which greatly relieves the crowded condition that would otherwise obtain. As previously stated, they are also preferable as being more reliable.

36 The turbo-compressor supplying air to blast furnaces under pressures ranging from 20 lb. to 30 lb. has almost entirely supplanted the compound reciprocating blowing engine. One large concern formerly in this work has abandoned the construction of blowing engines and is now building turbine apparatus exclusively. For this service, the turbine may be run from 2500 to 4000 r.p.m., and there is a great saving of weight and space, as an engine of this type is six or eight times as heavy as the centrifugal blower, and consequently costs much more. It can be installed comfortably where the blowing engine would be out of the question, and in new installations the relative cost of building and foundation for the two types has a direct and important bearing.

UTILIZATION OF EXHAUST STEAM

37 The advantage of an oil-free exhaust is in many plants of considerable value, and especially so in manufacturing processes where steam is used, as there are many such opportunities for the utilization of low pressure steam if the oil has been eliminated. For the blocking of hats and in the treatment of other felt and textile products, absolutely clean steam is necessary. As heretofore mentioned, paper manufacturers have used turbines for driving the constant speed mechanism of paper machines in order to secure more uniform angular velocity, and to avoid among other things trouble caused by oil accumulation in the drying rolls. The danger of oil deposits in high pressure steam boilers is well known to all.

38 In chemical processes where steam is used for precipitation, as in the precipitation of magnesia, a small fraction of a grain of oil per gallon will often retard the process or cause the precipitate to be of an entirely different character from that obtained with oil-free steam. The separation of the oil in exhaust steam is never absolutely complete, and fatty constituents are especially apt to pass the separator.

AVAILABLE COOLING WATER SUPPLY

39 Where the available cooling water supply is limited and must be artificially cooled and recirculated, the cost of the cooling apparatus and the power required must be considered. The conditions will, perhaps, be best illustrated by an example: Assuming a turbine to

run at 28 in. vacuum, and a temperature rise of the cooling water to within 10 deg. of that due to the vacuum, the circulation of 52 units by weight of cooling water for each unit of steam condensed will be necessary. In the case of the engine, which will operate at, say, 26 in. vacuum, other conditions remaining the same, there will be from 25 to 27 lb. of cooling water to be handled for each pound of steam condensed.

40 With a cooling pond returning water at 90 deg. to produce 27 in. vacuum in a turbine plant, the pumps must circulate 70 units of cooling water per lb. of steam, as against, say, 30 units required to produce 25 in. vacuum for the reciprocating engine.

OPERATING ADVANTAGES

41 From the operating point of view, the turbine possesses a great advantage in the simplicity of its construction, a factor which tends toward increased reliability and lower cost of maintenance. It can usually be more quickly started and loaded and, in operation, usually requires very much less attention than an engine unit of corresponding capacity. The lubricating arrangements are few in number and of simple design.

SUMMARY

42 Summarizing the foregoing, the fields of usefulness of the turbine and engine may be briefly stated to be:

APPLICABILITY OF TURBINES

1 *Direct connected units, operating condensing.* 60-cycle generators in all sizes, also 25-cycle generators above 1000 kw. capacity. (This paper is, however, not intended to deal with units of this size.)

Direct current generators in sizes up to 1000 kw. capacity, including exciter units of all sizes.

Centrifugal pumping machinery operating under substantially constant head and quantity conditions, and at moderately high head, say from 100 ft. up, depending upon the size of the unit.

Fans and blowers for delivering air at pressures from 1½ in. water column to 30 lb. per sq. in.

2 *Direct connected units, operating non-condensing* for all the above purposes, in those cases wherein steam economy is not the prime factor or where the exhaust steam can be

completely utilized, and, in the latter case, particularly where oil-free exhaust steam is desirable or essential.

3 *Geared units*, operating either condensing or non-condensing for all the above mentioned applications, and in addition, many others which would otherwise fall in the category of the steam engine, on account of the relatively slow speed of the apparatus to be driven.

APPLICABILITY OF ENGINES

1 *Non-condensing units, direct connected or belted* and used for driving:

Electric generators of all classes excepting exciter sets of small capacity, unless belted from the main engine.

Centrifugal pumping machinery, operating under variable head and quality conditions and at relatively low heads, say up to 100 ft., depending on the capacity of the unit.

Pumps and compressors for delivering water or gases in relatively small quantities and at relatively high pressures —in the case of pumps at pressures above 100 lb. per sq. in. and in the case of compressors at pressures from 1 lb. per sq. in. and above.

Fans and blowers (including induced draft fans) for handling air in variable quantities and at relatively low pressures, say not over 5 in. water column.

Line shafts of mills, where the driven apparatus is closely grouped and the load factor is good.

All apparatus requiring reversal in direction of rotation, as in hoisting engines and engines for traction purposes.

2 *Condensing units direct connected or belted*, for all the above purposes, particularly where the condensing water supply is limited, and where the water must be re-cooled and recirculated.

DISCUSSION

This paper was originally presented at a meeting of the Philadelphia Section on April 12, 1915. Subsequent to this presentation, contributed discussions of this subject were received from C. R. Waller,[1] Sanford A. Moss[2] and Paul A. Bancel.[3] Excerpts from

[1]Jour. Am. Soc. M. E., February, 1916.
[2]Jour. Am. Soc. M. E., February, 1916.
[3]Jour. Am. Soc. M. E., April, 1916.

these discussions bearing directly on the statements in the paper follow.

C. R. Waller. The data regarding economy of turbine driven pumps given in this paper do not represent the highest performances that have been obtained with a turbine driven centrifugal pumping unit, up to the present time.

The geared turbine pumping unit installed at the Ross Pumping Station, Pittsburgh, was the first pumping machine of this description ever built. Since that time other units have been constructed where better results have been obtained. I refer to the official test made by the City of Cleveland of the 30 million gallon centrifugal pumping unit installed at the Kirtland pumping station. This unit showed a duty of 128,400,000 ft-lb. per million B.t.u. or 152,020,000 ft-lb. per 1000 lb. of steam. The unit in question during test showed a delivery of 30.3 million gallons per day against a total head of 236.3 ft.; the steam supplied was 153.58 lb. per sq. in. gage, with 102.6 deg. fahr. superheat, and the turbine exhausting into a vacuum of 28.25 in. of mercury.

Mr. Barstow, discussing boiler feed pumps, gives the impression that turbine driven centrifugal boiler feed pumps of less than 250 gal. per min. are not practicable. I would like to add to .his data that these pumps are built and used in quite large numbers for capacities ranging from 50 to 200 gal. per min.

Sanford A. Moss. As the author has stated, the general advancement of the steam turbine has been exceedingly rapid. It is not surprising, therefore, that there is only a small region from which the engine has not yet been driven. Small turbines have already begun to invade this field, however.

Usually, as pointed out by Mr. Barstow, the water rate of a small turbine has been unimportant. By proper selection of the number of stages, number of bucket rows per stage, and wheel diameter for a given horsepower, speed and steam supply, almost any desired water rate can be obtained.

Single stage machines with two or three rotating bucket rows have been very popular. Now, however, more efficient turbines are available with a greater number of moving bucket rows.

A balance must always be made between increased first cost and increased operating cost.

One advantage of turbines over reciprocating engines, which the author has not mentioned, is that the turbine maintains it original water rate, while many types of engine fall off appreciably This was very clearly shown in a series of tests published by F. W Dean in THE JOURNAL, June, 1908.

PAUL A. BANCEL. In the last paragraph the author states the engines are particularly applicable for condensing units where the condensing water supply is limited and where the water must be recooled and recirculated.

A compound condensing engine or a single cylinder uniflow engine has high efficiency—that is, utilizes a large share of the available energy between the terminable pressures, provided the exhaust pressure is not too low. A vacuum of 26 in. is about as high as it pays to go with an engine. Now where water must be recooled by a spray pond or a tower, very high vacuums are not attainable because of the physical and commercial limits to the temperature to which the water can be cooled, to say nothing of the cost of pumping the water if a large quantity is circulated. Thus the reciprocating engine has a field of peculiar applicability in plants of this kind.

The steam turbine is at a disadvantage under these conditions of moderate vacuum. In a turbine the efficiency of conversion of energy of the steam into mechanical energy is greater the lower the existing pressure. High vacuum is doubly important to the steam turbine—to furnish more energy and to furnish energy which can be abstracted most efficiently.

The turbine is cheaper than the engine, requires less space and foundations, is cheaper to operate and can be built in larger units than the engine, and therefore it is pertinent to inquire how far it is possible and profitable to go in obtaining high vacuum, so as to reduce the steam consumption to that of the engine.

No. 1506

THE FLOW OF AIR THROUGH THIN-PLATE ORIFICES

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MEASUREMENT OF NATURAL GAS IN LARGE QUANTITIES

That the problem of measuring accurately large quantities of natural gas has not yet been solved will be conceded by most engineers familiar with the situation. This question is not an easy one. It is only in the last few years that anything like the proper amount of attention has been given the subject. The realization of the rapidly diminishing supply of the commodity, however, has given an impetus to accurate measurement as an end towards conservation. The increasing market price of natural gas has also been a factor.

2 During the past five years, as the desire for greater accuracy in measurement became more general, engineers began to look into the problem, investigating and improving upon the types of meters already in use, and applying other principles to new types of meters. Considerable experimental work in the field of gas measurement has been and is now being done by all the larger pipe-line companies, also by some manufacturers, and unquestionably great strides have been made in this period as a result.

3 Among the more important types of meters now used in measuring natural gas in large quantities, the following might be mentioned: Proportional meters, pitot tubes, orifice meters, venturi meters, rotary meters (an application of the anemometer principle) and electric (or "calorimetric") meters. The last few years have seen a great increase in the number of pitot tube and orifice meter installations, due to the development some years ago of satisfactory differential pressure gages for recording small differences in pressure between two high static pressures. Before these

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Awarded the Junior Prize, 1915.

gages were put on the market, it was necessary to have two employees at each station where an orifice or tube was installed, to record quarter or half hourly readings of the differential pressure, as indicated on an ordinary water column. Except in the very large measuring stations, this meant a prohibitive operating cost.

4 This paper will describe in some detail the method used by a large pipe-line company in the so-called Mid-Continent Field in calibrating its orifice meter discs.

DESCRIPTION OF AN ORIFICE METER

5 An orifice meter consists of a calibrated disc in a pipe-line, with pressure line connections running to two indicating or recording gages—one gage for measuring the static pressure of the flowing gas, the second to measure the differential drop in pressure across the orifice disc. The disc is usually made of iron or steel plate, and is inserted in a recessed flange union. It is necessary to take the disc out of the line at times, either for examination or to be replaced by a different sized disc; and inserting the disc in a flange makes the most convenient installation.

6 The discs tested by the method to be described are machined out of quarter-inch boiler plate. The edge of the orifice proper is flat for $\frac{1}{32}$ in. and bevelled at 45 deg. for the remainder of the thickness of the plate. The disc is inserted in the flange union, so that the bevel faces the outlet side.

7 The ordinary practice in orifice meter installations is to have the gage line connections right at the flange, that is, the inlet and the outlet pressures are taken within an inch or two of the orifice disc, through holes drilled into the companion flanges. In this particular, the meters of the type tested by the author show a departure from the common practice. In the meters tested, the high pressure connection was two and a half times the diameter of the pipe-line ahead of the orifice disc, and the low pressure connection was eight times the diameter of the pipe-line behind the disc. This means that in an orifice meter installation on a 10-in. line, for example, the high pressure connection is 25 in. in front of the disc, and the low pressure connection 80 in. behind it, regardless of the size of the orifice in the line.

8 It was found by experiments made at Charlottenburg some eight years ago that, for any flow through an orifice disc not giving an excessive drop in pressure, pressure connections at just the distances mentioned above would give a smaller pressure drop across

the disc than would connections placed at any nearer position t the disc. There can hardly be any doubt but that the inserting c an orifice disc in a pipe-line would cause eddies, and while there : no evidence to show that the presence of eddies would affect th accuracy of any measurement through the disc, it was thought be to eliminate this source of possible uncertainty.

DERIVATION OF ORIFICE METER FORMULAE FOR FLOW OF AIR

9 The fundamental formula for flow through an orifice is

$$V = C_v \sqrt{2gH} \ldots\ldots\ldots\ldots [1$$

where V = velocity of flow through the orifice, ft. per sec.

C_v = so-called "velocity coefficient," varying with the size an shape of the disc. This constant is also known sometim as the "efficiency," though this term is misleading.

g = acceleration due to gravity, ft. per sec., per sec.

H = drop in pressure through the orifice disc, expressed i feet of head of the fluid flowing, at temperature an pressure conditions of flow.

10 In this fundamental formula, the differential drop across th orifice is given in terms of feet head of fluid. The differential pre sure gages used in commercial meter installations are nearly a graduated to read in inches of water drop in pressure. It is nece sary, therefore, to derive from the fundamental formula an expressio in which the drop is in terms of inches of water. This can readil be done, as follows:

11 Assuming air as the flowing fluid, the fundamental formul [1] can be written

$$Q_1 = \frac{\pi d^2 900}{4 \times 144} \cdot C_v \sqrt{2gH} \ldots\ldots\ldots\ldots [2$$

where Q_1 = volume of fluid (air in this case) flowing per 15 min., i cubic feet at pressure and temperature P_1 and T_1, r spectively (the conditions at inlet of orifice)

d = diameter of orifice, in.

C_v, g and H as in the fundamental formula [1].

12 To reduce the value Q_1, which expresses volume at ten perature and pressure conditions of flow, to Q_o, the volume at th standard conditions of temperature and pressure (call T_o and I the standard conditions), it is only necessary to apply the perfe gas law. This is done by multiplying the right-hand side of equ tion [2] by P_1T_o/P_oT_1.

13 The drop in head, H, now expressed in feet of fluid at P_1 and T_1, must be reduced to inches of water, as explained above. Kent gives 1 ft. of air at 32 deg. fahr. as equal to 0.015534 in. head of water at 62 deg. From this, one foot head of air at P_1 and T_1 is equal to $(0.015534\ P_1\ 492) \div (14.7 \times T_1)$ inches head of water at 62 deg. Formula [2] can now be written:

$$Q_o = \frac{900\,\pi\, d^2\, P_1\, T_o}{4 \times 144\ P_o\, T_1} C_v \sqrt{\frac{2\,g\,h\,.\,14.7\ T_1}{0.015534 \times 492 \times P_1}} \quad \text{[3]}$$

and then simplified to

$$Q_o = 54.65\ d^2 \frac{T_o}{P_o} C_v \sqrt{\frac{h\,P_1}{T_1}} \quad \text{[4]}$$

14 Formula [4] is the general formula for calculating the flow of air through an orifice disc. A further simplification of this formula is practicable, however, for commercial purposes. As the temperature of the flowing gas is not usually measured, an average value is assumed. This is taken in Oklahoma as 60 deg. fahr. The pressure and temperature standards are definite, being usually fixed by contract. All these values, together with d, the diameter of the orifice, can be assembled into one constant. A reduction to the unit of 1000 cu. ft. is also made. This gives us:

$$Q_o \div 1000 = C_A \sqrt{h\,P_1} \quad \text{[5]}$$

where C_A is the so-called "air constant," found experimentally.

ORIFICE METER FORMULAE FOR GAS

15 The orifice meter formulae for flowing gas are derived by the same steps as those for flowing air. In the reduction of H (the differential of the fundamental formula expressed in feet head of the fluid) to h (the differential in inches of water) the density of the gas must be considered. If the specific gravity of the gas be taken as G, where air equals unity, the general formula for the flow of gas through an orifice meter becomes

$$Q_o = 54.65\ d^2 \frac{T_o}{P_o} C_v \sqrt{\frac{h\,P_1}{G\,T_1}} \quad \text{[6]}$$

This formula corresponds with formula [4] for flowing air.

16 The simplified commercial formula for a gas flow becomes

$$Q_o = \frac{C_A}{\sqrt{G}} \sqrt{h\,P_1} \text{ or } C_G \sqrt{h\,P_1} \quad \text{[7]}$$

where C_G is the so-called "gas coefficient," the meaning of which will be explained. This formula corresponds with formula [5] for flowing air.

17 Formula [7] is the formula actually used in commercial measurement. The values of P_1 and h are shown by the recording pressure and differential pressure gages, and C_G is mathematically derived from the constant of the disc as found by experiment. From the two readings and the gas coefficient, the delivery through the meter can be calculated.

RELATIONSHIP BETWEEN THE CONSTANTS, C_V, C_A AND C_G

18 As a rule, the theoretical velocity coefficient, C_V, is not used in calculating deliveries through an orifice. Its usefulness lies principally in the mathematical analysis of the formulae and for purposes of comparing experimental data of tests made under widely differing conditions.

19 C_A, the air constant, and C_G, the gas coefficient, are the quantities that are used commercially. The relation between these two, as can be seen by comparing formulae [5] and [7], is expressed by the equation

$$C_G = \frac{C_A}{\sqrt{G}} \qquad [8]$$

20 C_A, the air constant, is the value that is experimentally found, and does not vary for any disc, unless the assumed standards are changed. The gas coefficient, on the other hand, being a function of the gravity of the flowing gas, will vary, and the gas coefficients of identical discs would be different if the discs were passing gases of different gravities. Orifice disc calibration tests are therefore usually figured for C_A, the air constant, and this is the value that is recorded. Whenever a disc is put in line at a measuring station, the gravity of the gas to be measured is found by a test, and the proper gas coefficient calculated.

21 The relation between C_V and C_A is found by equating the right-hand sides of formulae [4] and [5], and can be expressed as

$$C_V = 11.55 \frac{C_A}{d^2} \qquad [9]$$

GENERAL OUTLINE OF THE JOPLIN TESTS

22 The tests on orifice meter discs to be described in this paper were carried out at Joplin, Mo. The discs were calibrated against the displacement of air from an old artificial gas holder at that place.

The holder was a two-lift holder, water sealed and of 250,000 cu. ft. nominal capacity. Roughly speaking, its dimensions were 90 ft. in diameter by 40 ft. total height. The lower lift only was used in the tests; this lift has a capacity of 110,000 cu. ft. The reason for using only the lower lift was the change in pressure of the air in the holder, as one lift seated on the bottom.

23 Of the several original outlets from the holder, all but one were securely blanked. The remaining 12-in. outlet was led into a long building, and connected to a straight run of some 40 ft. of pipe, near the center of which was the orifice flange. The air passing out of the holder went through the orifice disc, and discharged into the atmosphere perhaps 20 ft. beyond. A motor driven blower was used to fill the holder with air previous to each test.

24 *Leakage Tests on Holder.* The first tests made were to determine the rate of leakage from the holder. In order to obtain a fair average, a number of such tests were run at the start, with the holder at varying heights. Leakage tests were also run at intervals throughout the whole work, to make sure that the leakage figure first obtained had not materially changed.

25 The first leakage tests (run during August, 1913) were unsatisfactory on account of the large difference between temperature conditions at the start and finish of test. To avoid this difficulty, tests of 24 hours duration, starting at about midnight, were made, and better results obtained. The average of three long leakage tests showed 103 cu. ft. leakage per hr. The correction used in all the Joplin tests was taken as 100 cu. ft. per hr. The result of later leakage tests showed practically the same leakage as the above average, the highest value in any 24-hr. test being 115 cu. ft. per hr.

26 *Changes of Volume in Holder with Temperature Variation.* During the leakage tests, it was noticed that the rise and fall of the holder with temperature changes was a greater factor than had been anticipated. A 4-ft. rise from midnight to noon was not uncommon during the hot weather. It was necessary, therefore, to ascertain very accurately the proper correction to apply for temperature changes taking place during a test.

27 Table 1 shows observations and calculated results made a test run for this purpose. The holder was filled to about th[illegible] quarters capacity and allowed to stand, hourly readings being ta[illegible] of all quantities involved. The so-called "top" temperature is reading found by lowering a thermometer 2 ft. or so into the ho[illegible] through a bolt hole on top.

28 From the data the net change in volume due to temperature variation for each hourly period was calculated. It was found that the changes in volume as observed were always greater than a

TABLE 1 READINGS AND CALCULATED RESULTS OF FIRST 24-HOUR TEST ON GAS HOLDER FOR INVESTIGATING VARIATION OF VOLUME OF AIR IN HOLDER WITH TEMPERATURE CHANGE

Time	Temperatures		Change of Volume of Air in Holder, Corrected for Level of Water Seal and for Leakage. In. Height		Calculated "Combined" Temp.	Calculated Theoretical Change of Volume from Beginning of Test	Same Values Corrected for Calculated Lag of 8¼ In. at Start
	Atmos.	"Top"	During Previous Hour	From Beginning of Test			
6 p.m.	96	103	....	....	98½	....	— 8¼
7 p.m.	93	97	—8 9/16	— 8 9/16	94½	— 6⅛	—14⅜
8 p.m.	90	91	—8 13/16	—17⅜	90½	—11⅞	—20⅛
9 p.m.	88	88	—4⅛	—21½	88	—15⅛	—23⅜
10 p.m.	87½	86	—2⅝	—24⅛	87	—16⅜	—24⅝
11 p.m.	86	86	—1⅛	—25¼	86	—17⅝	— 25⅞
12 night	85	85	—1 7/16	—26 11/16	85	—18⅞	—27⅛
1 a.m.	84	84	—1¼	—27 15/16	84	—20⅛	—28⅜
2 a.m.	82	83	—1⅛	—29 1/16	82½	—21¾	—30
3 a.m.	81	82	—1⅛	—30 3/16	81½	—23	—31¼
4 a.m.	80	81	—1¼	—31 7/16	80½	—24¼	—32½
5 a.m.	79½	80	—1⅜	—32 13/16	79½	—25	—33¼
6 a.m.	78½	80	— ¼	—33 1/16	79	—25¾	—34
7 a.m.	80	86	5⅛	—27 15/16	82	—21⅞	—30⅛
8 a.m.	85½	100	7⅞	—20 1/16	90½	—10¾	—19
9 a.m.	87½	108	7¼	—12 13/16	94½	— 5	—13¼
10 a.m.	92	118	7 5/16	— 5½	101	4⅞	— 3⅜
11 a.m.	95	122	7⅛	1⅝	104	9⅛	⅞
12 noon	96	131	5⅛	6¾	108	16⅜	8⅛
1 p.m.	98	132	2½	9¼	109½	19⅛	10⅞
2 p.m.	100½	131½	1	10¼	111	21⅞	13⅝
3 p.m.	100½	129	2 3/16	12 7/16	110	20	11¾
4 p.m.	98½	126	—2 15/16	9½	107½	15⅜	7⅛
5 p.m.	97	117	—2¾	6¾	104	9½	1¼
6 p.m.	94½	109	—4 9/16	2 3/16	99½	2	— 6¼

Height of top of holder, at start, above water in seal, 421 in.
Date of test, August 8-9, 1913.

calculation based on the ratio of absolute temperatures alone would give. After some little study and debating, it was decided that this was due to the presence of aqueous vapor in the holder. This point has always appeared especially interesting, and therefore des further analysis here.

29 That saturated water vapor was present in the holder evident. By Dalton's law, it is correct to assume that the pre inside the holder is made up of two distinct quantities (*a*) the te of the saturated aqueous vapor, (*b*) the pressure of the air in holder. The sum of these two component pressures, expresse absolute, will be the barometer reading plus the reading of a U-connected up to the holder pressure. Therefore, for a cons barometer, the total pressure of the holder will not change.

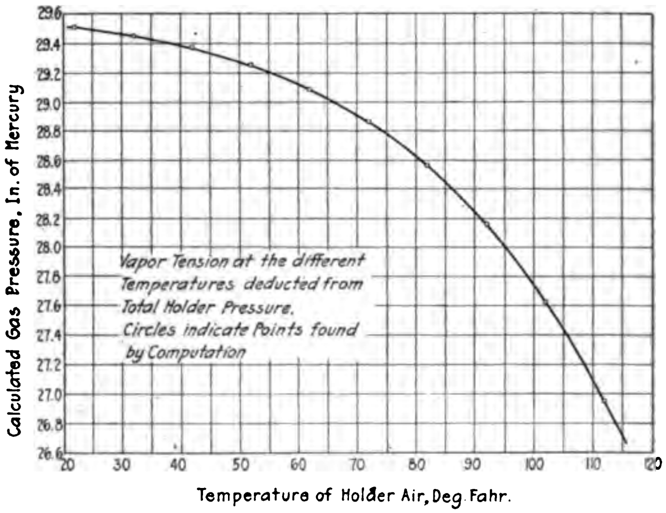


FIG. 1 CALCULATED "GAS PRESSURE" IN HOLDER WITH VARYING TEMPERATURE

however, the temperature should rise while the barometer rem constant, the tension of the saturated vapor will increase, and second component of the total pressure, the pressure of the (which will be called the "gas pressure" in this connection) mus correspondingly decreased. With varying barometer readings, change in value of each component pressure will be different. H ever, a very close approximation can be had by basing all co tions on the average barometric reading at Joplin, viz., 29.3 in.

30 For any temperature, the second component of the t holder pressure—the gas pressure—can be found by subtrac from the total holder pressure the vapor tension for the temperat The total pressure is the assumed barometric reading, 29.3 in.,

the observed pressure of the holder, 4.60 in. of water. The vapor tension for varying temperatures can be found in any handbook. Fig. 1 shows the variation of the gas pressure in the holder for temperature changes. The circles indicate points found by computation, and through these the curve is drawn.

31 Charles' law states that the ratio $p v/t$ is constant for a perfect gas. Under such small changes of pressure and temperature, air can be assumed a perfect gas. The value of the pressure corresponding with the temperature being known from Fig. 1, the volume can be found by calling the volume at any assumed standard

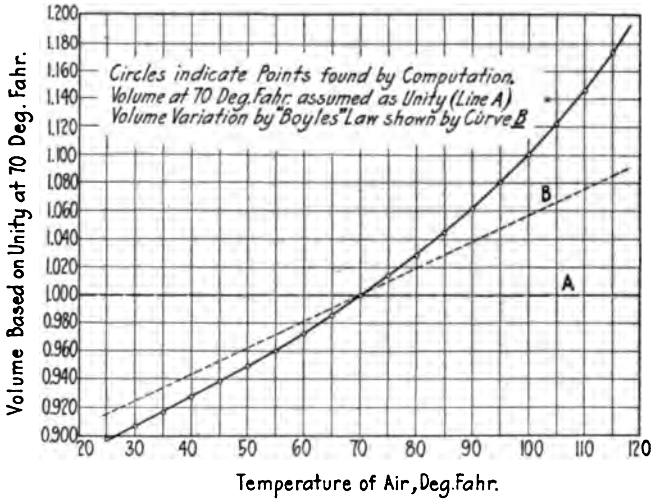


FIG. 2 CALCULATED VARIATION IN VOLUME OF AIR ENCLOSED IN HOLDER OVER WATER, WITH CHANGE OF TEMPERATURE

temperature unity. In these calculations, the volume at 70 deg. fahr. is taken as unity. The full curve of Fig. 2 is the final result. Dotted in, for purposes of comparison, is a curve showing the Boyles' law variation for constant pressure. It can be seen at a glance that the volume variation under temperature changes, as observed at Joplin, is considerably more than it would be in the absence of the water vapor.

32 To go back to the test made on the holder to study the correction to be applied for varying temperatures, taking account of the effect of the vapor tension, as just explained, it was found in the study of the volume variation with temperature that the observed rise and fall of the holder during the test corresponded with a tem-

perature change equal to the sum of two-thirds the atmospheric temperature, plus one-third the "top" temperature. This simply means, of course, that the average temperature of the holder air is that combination of the two observed temperatures.

33 Fig. 3 shows a test of the correctness of this average temperature of the holder air. The dotted curve shows the computed theoretical rise and fall of the holder, using the temperature correc-

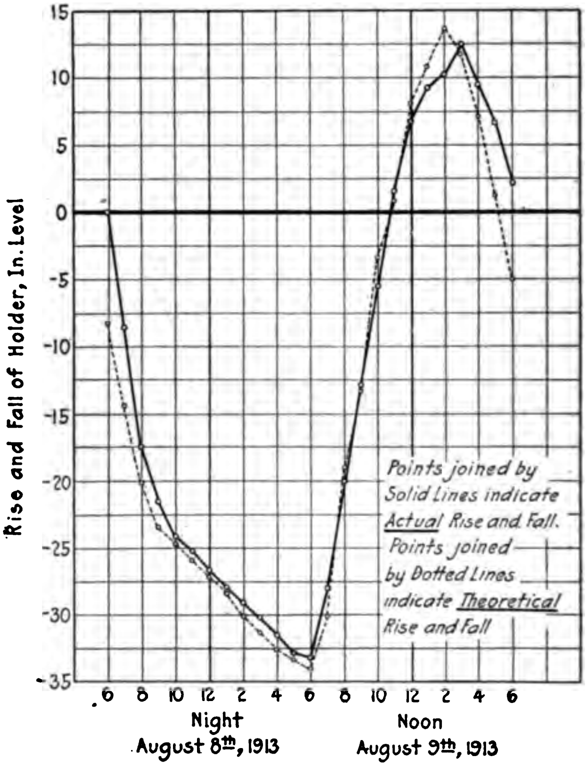
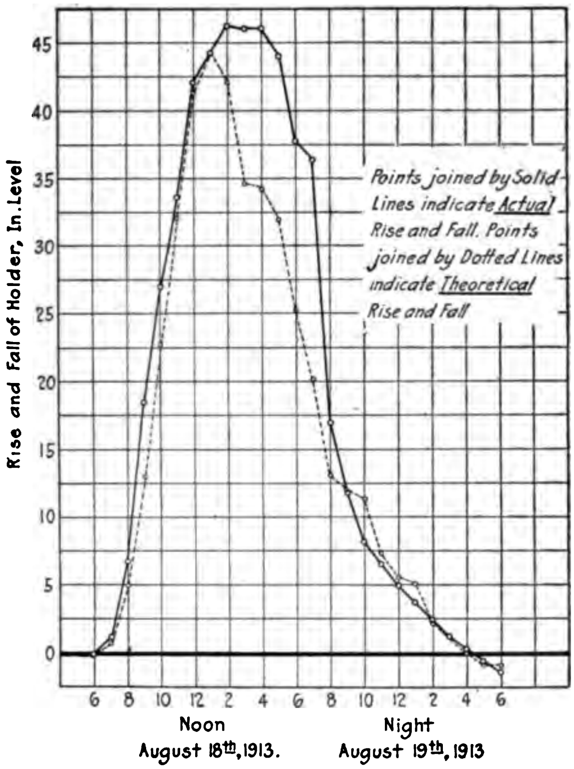


FIG. 3 ACTUAL AND THEORETICAL RISE AND FALL OF HOLDER, UNDER TEMPERATURE CHANGES. FIRST 24-HR. TEST

tion just described, and based on the so-called "combined" temperature. The heavy curve shows the actual rise and fall observed. These two curves follow each other fairly well, except for a lag during the middle of the day. They bear the closest relation during the period between 9 P.M. and 6 A.M., when the effect of the sun shining down on the black holder top is not present. During the afternoon it is evident that the "top" temperature has a greater proportionate ect on the average temperature of the air in the holder than the mbined" temperature allows.

34 Fig. 4 shows curves of a similar nature for a second run made check the results of the first run. The actual and theoretical rise fall of the holder, plotted on the same scale over each other, n show very good agreement, particularly during the night.



G. 4 Actual and Theoretical Rise and Fall of Holder, under Temperature Changes. Second 24-hr Test

lag during the hottest part of the day is again shown, in somet better shape, due to the fact that this second test was started A.M., while the first started at 6 P.M.

DESCRIPTION OF PROCEDURE DURING ORIFICE TESTS

35 Table 2 shows a sample page of the observations and calted results of one orifice test. These tests were made with a ng start, that is, the air was started discharging through the

meter orifice several minutes before the first tank level reading was taken. The final tank level reading was taken under similar conditions. Half-hourly readings of atmospheric temperature, "top" temperature, tank level, water level in seal, differential drop in inches of water across orifice disc, and the temperature of the discharged air were taken. To facilitate calculation, the tank level reading was taken by a gage stick, marked off in cubic feet displacement. The exact diameter of the lower lift was found, and from this it was computed that a displacement of 100 cu. ft. corresponded

TABLE 2 READINGS AND CALCULATED RESULTS OF A SAMPLE HOLDER TEST ON ORIFICE METER DISC

Time	Temperatures			Gage Stick Reading	U-Tube In. Water	Calculated (Apparent) Displacement in ½ hr.	"Combined" Temp.	Correction for Temp. Changes	Calculations
	Atmos.	"Top"	Orifice						
8.15	66	66	66	252,675	4.39	....	66	....	
8.45	67½	....	65	244,050	4.39	8,625	66	0	
9.15	67	66	65	236,350	4.38	7,700	66⅔	330	Apparent total. 108,975
									Seal water correction....... —140
9.45	65	....	64½	228,125	4.37	8,225	66	—320	
10.15	66	64½	64	219,725	4.36	8,400	65½	—230	Leakage corr... —650
10.45	66	....	64	210,950	4.36	8,775	65½	0	Temperature corr......... —960
11.15	65½	63½	63½	202,325	4.35	8,625	65	—210	
11.45	65½	....	63½	193,700	4.35	8,625	65	0	Net total...... 107,225
12.15	65½	63	63½	185,625	4.34	8,075	64½	—200	Avg. flow...... 8,248
12.45	65	....	63½	177,175	4.32	8,450	64½	0	Calculated C_A.. .520
1.15	65	63	63½	168,850	4.30	8,325	64½	0	Calculated C_V.. 67.3%
1.45	64½	....	63	160,425	4.28	8,425	64	—170	
2.15	64	62½	63	152,070	4.26	8,350	63½	—160	
2.45	64	....	63	143,700	4.24	8,375	63½	0	
			64		4.34	108,975	65	—960	

Test No. 227, Nov. 12, 1913. Orifice Disc No. 8301. Barometer 29.35 in.

with a holder drop of 0.1835 in. This 100-ft. graduation was the smallest on the gage stick. The space could readily be divided visually into quarters, so the content of the holder at any moment could be read to the nearest 25 cu. ft. Differences between two consecutive half-hourly tank level readings would therefore give the uncorrected (or apparent) quantity of air passing out in that period. Preliminary corrections have to be applied to this value as follows: (1) holder leakage, at the rate of 100 cu. ft. per hr., (2) variation of the water level in seal, due to leakage out, evaporation, or pumping in of fresh water, (3) temperature change during test.

TABLE 3 SUMMARY OF HOLDER TESTS ON 8-IN. ORIFICE METERS

Test No.	No. of Meter Disc	Date of Test and Duration in Hr.	Avg. Corr. Rate. Cu. ft./30m.	U-Tube Reading, In. Water	Barometer, In. Hg.	Observed Temp. Flow	Observed Temp. "Combined"	Calc. C_A
201	8401	Sept. 25 1½	15,503	4.07	29.3	58	52	1.035
202	8501	26 2	25,361	3.31	29.3	62	56	1.868
203	8301	26 3	8,178	4.415	29.3	58	54	0.522
204	8351	27 3	11,671	4.26	29.3	62	60	0.752
205	8451	27 3	20,223	3.82	29.3	62	60	1.375
206	8451	28 2	20,589	3.79	29.2	64	61	1.409
207	8351	28 3	11,668	4.33	29.2	61½	60	0.747
208	8401	28 2½	15,796	4.11	29.2	61	58	1.041
209	8351	30 2½	11,850	4.31	29.3	66	62	0.759
210	8451	30 2	20,421	3.83	29.3	64	59	1.394
211	8501	Oct. 2 1½	26,647	3.36	29.3	70	67	1.914
212	8501	2 1½	26,187	3.40	29.3	67½	63	1.883
213	8401	2 2½	15,839	4.14	29.3	65	60½	1.038
214	8301	3 2½	6,124	2.355	29.3	70	69	0.527
215	8301	4 5½	8,496	4.41	29.2	70	70½	0.532
216	8201	6 4½	3,579	4.52	29.2	68	65	0.2234
217	8201	7 4½	2,575	2.37	29.3	69	65	0.2217
218	8452	Nov. 3 2	18,989	3.37	29.4	58	55	1.387
219	8452	3 2½	18,772	3.42	29.4	57	54	1.358
220	8452	4 2	18,970	3.44	29.5	57	52	1.372
221	8452	4 2	18,791	3.44	29.5	55	49	1.361
222	8502	5 2	23,099	2.81	29.45	57	52	1.841
223	8502	5 1½	23,215	2.82	29.45	56	50	1.853
224	8502	5 1½	23,375	2.80	29.45	55	50	1.879
225	8201	6 5	3,641	4.52	29.3	60	59	0.2274
226	8301	11 3½	8,166	4.33	29.35	59½	58½	0.521
227	8301	12 6½	8,248	4.34	29.35	64	65	0.520
228	8502	13 2	23,485	2.80	29.4	63	64½	1.852
229	8201	14 5	3,566	4.53	29.45	64½	64½	0.2248
230	8201	15 4	3,560	4.585	29.3	54	49	0.2242
231	8352	Dec. 5 3	11,257	4.10	29.4	59	56	0.742
232	8601	6 1½	31,267	1.70	29.2	55½	46	3.273
233	8352	5 4	11,241	4.18	29.1	55½	48	0.746
234	8601	7 1½	30,478	1.65	29.5	51	33	3.293
235	8352	6 3½	11,319	4.09	29.2	55	46	0.762
236	8601	7 1½	30,463	1.65	29.5	51	31	3.306
237	8353	5 4½	11,161	4.08	29.4	59½	56	0.738
238	8502	7 1½	22,758	2.90	29.5	50	30	1.866
239	8601	30 1½	36,548	2.355	29.3	47	32½	3.305

TABLE 3 SUMMARY OF HOLDER TESTS ON 8-IN. ORIFICE METERS—CONCLUDED

Test No.	No. of Meter Disc	Date of Test and Duration in Hr.	Avg. Corr. Rate Cu. ft./30m.	U-Tube Reading, In. Water	Barometer, In. Hg.	Observed Temp. Flow	Observed Temp. "Combined"	Calc. C_A
240	8551	30 1½	31,023	2.99	29.3	48	33	2.487
241	8551	31 1½	30,942	3.00	29.2	50½	37	2.467
242	8571	31 1½	33,982	2.665	29.2	49	37	2.871
243	8571	Jan. 1 1½	29,408	1.965	29.0	53½	46½	2.865
244	8601	1 1½	31,093	1.64	29.0	53	45	3.325
245	8551	1 1½	27,337	2.30	29.0	52	40½	2.485
246	8352	2 3½	10,969	4.13	29.3	48½	30	0.752
247	8352	2 3½	11,148	4.31	29.3	48	27½	0.752
248	8551	3 1½	26,775	2.285	29.3	48½	31½	2.468
249	8601	3 1½	30,503	1.615	29.3	49½	32	3.344
250	8571	3 1½	28,495	1.96	29.3	48½	28½	2.853
251	8521	Feb. 16 1½	28,108	3.13	29.4	49	42	2.163
252	8251	18 3	5,684	4.62	29.1	47½	35½	0.3653
253	8251	19 3½	5,536	4.51	29.3	49	35	0.3599
254	8521	19 1½	28,075	3.22	29.3	47½	35	2.162
255	8151	20 9	1,877	4.63	29.4	52½	31	0.1216
256	8151	21 5½	2,144	4.63	29.7	56	49	0.1338
257	8151	23 9	1,802	4.61	29.6	46	12½	0.1196
258	8305	Mar. 4 3	8,101	4.49	29.4	52½	36	0.5275
259	8506	6 1½	23,425	3.00	29.5	49½	35½	1.865
260	8506	9 1½	23,617	2.96	29.4	52½	44	1.867
261	8473	25 1½	22,375	3.30	29.35	58½	64½	1.615
262	8474	25 1½	21,985	3.28	29.35	59	64½	1.595
263	8473	25 1½	22,162	3.28	29.4	59	62½	1.615
264	8474	25 1½	22,045	3.27	29.4	59	63	1.607
265	8251	26 5	5,655	4.51	29.4	59	59	0.3532
266	8151	27 5½	2,048	4.59	29.35	63	59	0.1274
267	8251	30 6	5,780	4.52	29.4	62	63	0.3603
268	8151	April 1 6½	2,161	4.64	29.5	62	56½	0.1339
269	8251	2 5	5,590	4.50	29.5	61	59	0.3479
270	8171	6 5	2,742	4.62	29.3	59	55½	0.1696
271	8171	7 6½	2,564	4.61	29.55	54	36½	0.1644
272	8171	8 8	2,553	4.62	29.65	54	29	0.1658
273	8171	9 6	2,513	4.61	29.5	55	34½	0.1622
274	8151	18 6½	2,000	4.61	29.4	59	52	0.1253
275	8151	19 6	1,866	4.60	29.45	59	44	0.1188
276	8151	20 6½	1,998	4.59	29.4	64	60	0.1238

TABLE 4 SUMMARY OF HOLDER TESTS ON 10-IN. ORIFICE METERS

Test No.	No. of Meter Disc	Date of Test and Duration in Hr.	Avg. Corr. Rate. Cu. ft./30 m.	U-Tube Reading, In. Water	Barometer, In. Hg.	Observed Temp.		Calc. C_A
						Flow	"Combined"	
401	10401	Dec. 20 3	13,978	4.345	29.4	48	32	0.9286
402	10501	20 2	23,156	3.94	29.5	47	32½	1.611
403	10501	20 2	23,138	3.94	29.4	48½	31	1.620
404	10801	26 1	53,855	1.38	29.4	45	29½	6,384
405	10751	21 1	49,513	1.89	29.5	48	33½	4.983
406	10801	21 1	53,975	1.38	29.3	50	35	6.370
407	10801	22 ½	55,595	1.43	29.2	48	36	6.429
408	10751	22 1	50,575	1.95	29.2	49	35	5.025
409	10501	23 2	23,209	3.95	29.2	49	32½	1.626
410	10401	23 3½	14,189	4.38	29.2	48	33	0.9406
411	10551	22 1½	29,042	3.89	29.2	48	35½	2.035
412	10551	26 1½	28,225	3.68	29.4	45	26½	2.055
413	10751	27 1	48,850	1.90	29.4	48	36½	4.976
414	10801	27 ¾	55,220	1.39	29.4	49	37	6.450
415	10501	27 2	21,080	3.28	29.4	49	35½	1.605
416	10701	28 1	45,038	2.41	29.5	46½	35½	3.986
417	10401	Jan. 5 3	13,990	4.31	29.4	43½	28½	0.9355
418	10801	5 1	53,363	1.37	29.5	47	28½	6.368
419	10501	6 2	23,076	3.89	29.3	49½	39	1.604
420	10651	6 1½	39,034	2.87	29.3	50	40	3.159
421	10551	7 1½	28,593	3.65	29.1	53½	49½	2.025
422	10601	7 1½	34,335	3.27	29.0	54	48½	2.581
423	10751	7 1	50,320	1.89	29.0	55½	47½	4.997
424	10551	8 1½	28,457	3.63	29.0	52½	46	2.035
425	10601	8 1½	34,028	3.24	29.0	49	37½	2.613
426	10601	8 1½	34,105	3.25	29.0	51½	43	2.592
431	10771	17 1	50,952	1.61	29.3	51½	44	5.474
432	10351	17 3½	10,890	4.39	29.3	53	44½	0.7067
433	10601	18 1½	34,250	3.31	29.0	52	44	2.578
434	10601	20 1½	33,983	3.27	29.2	52	42½	2.571
435	10451	20 2½	18,172	4.16	29.2	50	38½	1.224
436	10651	20 1	38,922	2.91	29.2	49	33	3.174
437	10351	22 3½	11,028	4.44	29.0	57	57	0.7003
438	10651	22 1¼	40,104	2.92	29.0	56½	57	3.147
439	10551	22 1½	14,412	3.68	29.0	57	56½	2.015
440	10551	23 1½	14,713	3.69	29.0	55	53½	2.063
441	10751	23 1	25,288	1.91	29.0	53	46	5.001
442	10701	23 1	45,510	2.39	29.0	52	40½	4.061
443	10701	24 1	44,938	2.35	29.2	47½	34	4.076

With these corrections applied, the correct quantity of air]
through the orifice is obtained. This quantity, however, is ex;
at holder pressure, and at some definite temperature, varyin;
day to day, i.e., the "combined" temperature. A reduction is
sary in order that the quantity of air passing through the
may be expressed in cubic feet at standard conditions of pressu

TABLE 4 SUMMARY OF HOLDER TESTS ON 10-IN. ORIFICE METERS—C

Test No.	No. of Meter Disc	Date of Test and Duration in Hr.	Avg. Corr. Rate. Cu. ft./30 m.	U-Tube Reading, In. Water	Barometer, In. Hg.	Observed Temp.	
						Flow	"Combined"
444	10801	24 1	53,895	1.36	29.2	46½	32½
449	10351	28 3½	11,101	4.44	29.0	60	62
452	10451	31 2½	18,329	4.23	29.4	49½	36
454	10501	Feb. 1 2	23,345	3.93	29.3	50	43
·456	10801	2 1	53,800	1.37	29.3	50	38
457	10351	2 3	10,805	4.37	29.3	51	40
458	10801	3 1	53,863	1.38	29.4	48	33
459	10701	3 1	44,560	2.40	29.4	47	30
460	10351	4 4	10,751	4.46	29.2	47	31½
461	10771	4 1	51,580	1.64	29.2	44	31½
462	10771	6 1	51,187	1.65	29.4	39	13½
463	10551	6 1½	27,538	3.76	29.4	39	7½
465	10621	11 1¼	36,934	3.12	29.3	53	48½
466	10621	11 1¼	36,742	3.11	29.3	51½	47½
467	10621	11 1¼	36,818	3.10	29.3	51½	46
477	10301	April 21 6	4,059	4.49	29.4	64	67
478	10301	22 6	4,056	4.47	29.5	65	67
479	10252	23 7	2,735	4.52	29.4	64	66½
480	10301	25 6½	3,863	4.37	29.65	60	47
481	10252	26 7	2,707	4.53	29.3	67	67
482	10252	27 7	2,745	4.57	29.25	63	62
483	10301	28 6	4,005	4.42	29.4	62½	56
484	10301	29 6	3,990	4.48	29.6	62	53
485	10301	30 6	3,931	4.49	29.6	61	53
486	10252	May 1 6	2,701	4.53	29.4	62	55½

temperature, (29.4 in. of mercury or 14.41 lb. per sq. in., and (
fahr.).

36 This corrected value of the quantity of air discharg
duced to standard conditions of temperature and pressur
responds with the value Q_0 in formula [4]. In this formula,
the quantities but C_v are known, the latter can be calculated.

relation between C_V and C_A, as shown by equation [9], gives a means of obtaining this latter value. As stated earlier in the paper, the 15-min. air constant was the value calculated in all the Joplin tests.

SUMMARY OF RESULTS OF TESTS

37 About one hundred and sixty tests on 8 and 10-in. orifice meter discs were run at Joplin during 1913-1914. A summary of the results of these tests is included in Tables 3 and 4. A note on the system used in numbering the discs will make the summary self-explanatory. The first one or two digits indicate the size of the pipe line in which the disc is inserted; the next two digits, the size of the orifice; the remaining digits, the serial number of the disc. For example, 8473 represents an 8-in. meter disc, 4¾-in. orifice; 104211 is a 10-in. meter disc, 4¼-in. orifice, etc. It was found necessary to discard perhaps half a dozen tests, on account of their disagreeing widely from the averages of the remainder. In two or three of these discarded tests, a shower or a fall of snow during the test furnishes a possible explanation; in other tests, no explanation was found.

38 It is worthy of special note that in these tests the standard used is an actual measurable volume, and not a standardized pitot tube or other indirect method of measurement. The advantage of being able to calibrate directly against displacement is a most important feature of these holder tests. A second feature of the tests is the ability to automatically secure a practically constant flow, without regulation of any kind.

39 Another point deserving mention is the fact that duplicate discs of sizes already tested at Joplin require no calibration of any kind. It is merely necessary to micrometer the orifice and to correct mathematically for any small deviation from the nominal diameter. For example, if a new 8-in. by 4-in. orifice disc micrometers 4.004 in. in diameter, and the result of the Joplin tests on the master 8-in. by 4-in. orifice be 1.034, the value of C_A for the slightly oversized disc would be $1.034\ (4.004 \div 4.000)^2$ or 1.036.

40 The possibility of securing constants for duplicate discs without actual calibration is a great advantage, as will be realized. It means that sufficient time and effort can be spent in calibrating the master discs to secure the highest possible accuracy, without having the cost of an individual disc excessively high. That the method of calculating constants for new discs as described above is correct has been very well shown by careful checks made at Joplin.

COMPARISON OF RESULTS WITH CHARLOTTENBURG TESTS

41 The only published report[1] of tests made on orifice discs similar to those tested at Joplin gives the values of the velocity

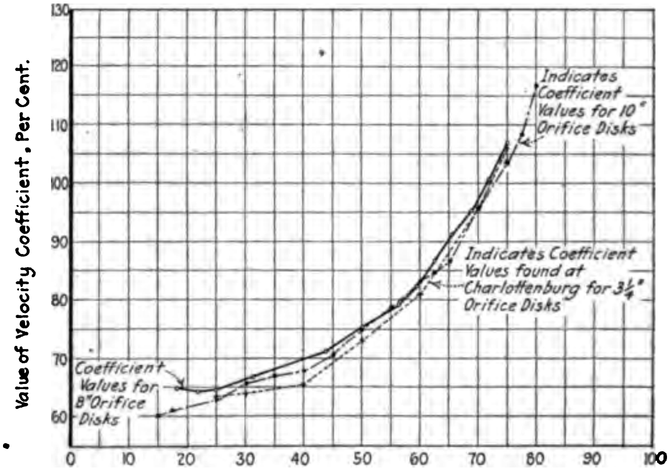


Fig. 5 Comparison of Velocity Coefficients of Orifice Meter Discs, as Found in Joplin Tests for 8-in. and 10-in. Pipe Lines, with Values found at Charlottenburg

SUMMARY OF VALUES PLOTTED

Joplin 8-in.		Joplin 10-in.		Charlottenburg 3¼-in.	
d/D	*c*	*d/D*	*C*	*d/D*	*C*
0.1875	64.9	0.15	60.1	0.25	63.5
0.219	64.2	0.175	61.1	0.30	64.0
0.25	64.7	0.25	62.8	0.40	65.5
0.3125	66.9	0.30	65.6	0.50	73.0
0.375	67.9	0.35	66.9	0.60	81.0
0.4375	71.1	0.40	67.9	0.70	96.0
0.50	75.3	0.45	70.4	0.75	106.0
0.5625	79.0	0.50	74.9		
0.594	82.2	0.55	78.7		
0.625	86.6	0.60	83.2		
0.655	91.3	0.625	84.6		
0.6875	95.3	0.65	86.6		
0.7187	100.7	0.70	95.9		
0.75	107.0	0.75	103.5		
		0.80	116.6		

coefficient, C_v, (as used in the fundamental formula) for 3¼-in. pipe found in tests made at Charlottenburg, Germany. Fig. 5 shows

[1]Zeit. des Ver. d. Ing., Feb. 23, 1908.

graphically the values found at Charlottenburg compared with the Joplin results for 8 and 10-in. pipe. The Joplin curves agree fairly well with the Charlottenburg values, especially if the difference in the pipe sizes is taken into account.

CONCLUSION

42 Nearly fifty orifice meters have been installed and are now in operation, their deliveries of high pressure gas being calculated from the air constants found in the Joplin tests. Experiments which are now in the course of completion, show conclusively that these constants (with the proper correction applied for the gravity of the gas being measured) are accurate to a very satisfactory commercial degree, and it is confidently expected that with some little further study and experimenting, particularly along the lines of the mechanical details, the orifice meter will take its place among the most reliable of the various methods of measuring natural gas in large quantities.

DISCUSSION

P. F. Walker (written). The experiments made at Joplin go far in establishing the use of the orifice meter as a reliable instrument, at least when employed with conditions essentially similar to those which obtained at that time.

It is well known that for a gas flowing through orifices and nozzles the conditions of flow may be treated by two distinct methods: *first,* considering the relationships existing during the expansion which takes place with the drop in pressure, and *second,* disregarding expansion and treating the gas as a non-expansive fluid. The second method, adopted in the tests described, depends for its absolute accuracy upon the amount of expansion or drop in pressure in proportion to the static pressure in the line on the upstream side of the orifice. The application of constants derived under one set of conditions must, therefore, be considered with care when conditions are different in marked degree. Because of these facts the corroborative tests referred to in the last paragraph of the paper were undertaken, and the writer has been associated in the work of analyzing the data from these tests.

The question which every engineer will ask, probably, is with reference to this matter of application of factors derived at substantially atmospheric pressure with a drop in pressure of but 3 to 5 in. of water in the orifice, to pipe line conditions where pressures may run close to 300 lb. and the rates of flow may vary through very wide limits, and so cause the differential pressure drop to vary correspondingly.

In the experiments referred to in the last paragraph, a short section (one mile or more) of unused pipe line was cut out and used as a reservoir. From this the natural gas was allowed to flow through orifices arranged much as in the tests with air, the only real difference being that the amount of gas flowing had to be calculated from observations on the drop in pressure in the reservoir. Observations could be, and were, made for a complete range of static pressures up to the values existing in the pipe lines at the time. A corresponding variation in the rates of flow extended the investigation over the desired range of values of the differential pressure across the orifices.

As first calculated, and this is as far as the work has progressed up to date, it was assumed that Boyle's and Gay Lussac's laws for perfect gases held for the gas in use. This affects three elements, as follows:

a the ratio P_1T_0/P_0T_1 which is introduced to convert the measured volume at the measured pressure P_1 to an equivalent volume at the standard pressure and temperature. Since the temperatures varied so slightly in this work, as is also true for regular operating conditions, no sensible error results from considering only the possible variation of volume ratio from the inverse pressure ratio at a constant temperature.

b the pressure which occurs in the denominator of the radical expression, in conjunction with 14.7 in the numerator, to express the change in density of the gas from standard to existing pressure.

c the volume flowing, Q_0, which is determined by the use of the pressure ratio, as in the first item.

The sum total of these elements is that the left hand side of the equation of flow is affected by the pressure ratio in the unit power and the right hand side is affected in the one-half power.

Dealing with all values as for perfect gases, the value of the coefficient designated in the paper as the "air-coefficient" was calculated for each observation, the value for air being found by reducing backwards by means of the density ratio in order to compare alts secured with gas of different densities. In analyzing this ss of data, three main variables had to be considered: static pres-e, differential drop in pressure or the dependent quantity hP_1 der the radical, and the ratio of orifice diameter to pipe line ameter. By reducing all to equivalent values of the coefficient r one standard area of orifice, the element of variable size of ifice was eliminated. A total of 1244 values was divided into 12 oups as follows:

Those for pressures over 100 lb., with $(hP_1)^{\frac{1}{2}}$ over 40
Those for pressures over 100 lb., with $(hP_1)^{\frac{1}{2}}$ from 21 to 40
Those for pressures over 100 lb., with $(hP_1)^{\frac{1}{2}}$ from 11 to 21
Those for pressures over 100 lb., with $(hP_1)^{\frac{1}{2}}$ below 11
A similar set of four for pressures from 50 to 100 lb.
A similar set of four for pressures below 50 lb.

For each of the twelve groups, values of the "air-coefficient" ere plotted against ratio of diameters of orifice and pipe, and an eraging curve drawn. From these average curves values of the efficient were then taken and plotted against values of $(hP_1)^{\frac{1}{2}}$ r each standard orifice, each orifice thus represented fixing a aracteristic curve for its particular ratio of diameter to diameter pipe.

These final curves establish certain laws, or tendencies. For ifices less than 40 per cent of pipe diameter the coefficient is essen-ally constant. For larger orifices there is a tendency toward lower lues for the greater rates of flow or larger values of $(hP_1)^{\frac{1}{2}}$. In neral the values tend to become less for the lower static pressures, is tendency being more marked with the orifices which are large proportion to the pipe. In general, too, the coefficients have lues slightly below the corresponding value determined for air in e Joplin tests which form the basis of the paper.

The work is not yet finished, but laboratory experiments are nder way to determine the actual *pressure-volume* relationship, the variations from Boyle's law, for the natural gas of the Mid-ntinental field. When these are completed the results will be calculated. In a surprising degree, considering the many vari-les between quantities and relationships for air near atmospheric essure and a gas far from the ideal gas on which the laws are sed and handled at wide variations in pressure and rate of flow, e values of the coefficients as found in the tests with the gas agree th the values found for air. The variations discovered involve so small a percentage of error, and are themselves subject to correction in such degree, that the air coefficients determined by the author remain standard for their purpose to date.

E. D. LELAND[1] (written). The author has presented a valuable record of results obtained by a reliable method of calibrating orifice discs in connection with the flow of air at low pressures, and the painstaking work tends to increase our confidence in the accuracy of this form of gas measuring device.

It would be an interesting check upon these results and afford valuable additional information, if a series of tests was made in which the flow of gas was reversed, using the same discs and the same holder, but a range of pressures materially higher, of course, making suitable provision for restoring the loss of heat due to expansion. Higher pressures are mentioned because accurate measurement of the flow of gas at comparatively high pressures is an important matter with large gas companies and has been receiving their careful attention.

TABLE 5 COMPARISON OF ACCURACY OF METERS

Cu. Ft. by Venturi	Cu. Ft. by Orifice	Cu. Ft. by Pitot	Cu. Ft. by Proportional Meter
Six-hour Test of Natural Gas at Average Pressure of 180 Lb.			
3,885,000	3,908,000	3,846,000	3,958,000
Eight-hour Test of Natural Gas at Average Pressure of 124 Lb.			
5,448,000	5,416,000	5,351,000	5,393,000
Volumes given are at 10 oz. and 62 deg. fahr.			

. The author's conclusion that the orifice meter will take its place among the most reliable of the various methods of measuring natural gas in large quantities is in harmony with results obtained by F. W. Schell[2] from recent tests near Blacksville, W. Va. In August, 1915, Mr. Schell conducted a long series of tests with special reference to the accuracy of the venturi meter, but including various other measuring devices, and the results which are given in Table 5,

[1]Supt. Compressing Stations, Philadelphia Co., Pittsburgh, Pa.
[2]Mechanical Engineer, Philadelphia Co.

include the performance of the orifice meter at a pressure of al 130 lb.

In these tests a 12-in. by 6-in. orifice meter, a 12-in. Wylie portional meter, and 5-in. and 3-in. calibrated pitot tubes were nected in series with the 12-in. by 6-in. standard venturi meter in at that point. Before using, the proportional meter was tested means of a standard test flowmeter and found to be accurate an good condition.

The close agreement of these various devices was very sa factory and indicated that the gas was being measured with a gree of accuracy well within the requirements of the Pennsylva State Public Utilities Commission.

H. B. Bernard (written). I wish to ask the author whe he has any trouble due to erosion of the orifice on account of small edge employed. The more common form of orifice is of 1/8 stock, with a flat edge for the entire thickness of the plate. Som this type have been in use since 1911, and to date have shown appreciable increase in diameter.

No mention is made in the paper of what size holes are use making connections. T. R. Weymouth, Mem. Am. Soc. M. E., found by numerous investigations that a saddle used over a 1/8 drilled hole gives the best results.

I cannot agree with the author if he believes distant connect more efficient than flange connections. With the latter, the co cient of flow—ratio of diameters curve—is much flatter and m mizes an error in not having the exact internal diameter of the p Undoubtedly eddying occurs at the orifice, but I know of no ins lations, not made near a compressor, where the accuracy of meas ment is impaired. Of course, any meter, if installed on the suc or discharge line close to the compressor, will invariably read h even though deadeners be used in the gage line. I doubt if connections employed by Mr. Hickstein would eliminate this.

In a test by W. T. Young, at the Buffalo Test Station in J 1914, an effort was made to determine the recovery of differen in an orifice meter. A Foxboro type flat edge orifice was u Flange connections and connections as described by the author, ing a maximum and a minimum differential respectively, were c pared. The results showed a recovery close to 17 per cent for in. differential at the flange, under 36.72 lb. per sq. in. abso pressure on the low side, and increasing at the rate of 0.4 per cent per 10 in. increase in differential under constant pressure. It was also shown that there was a slight drop between the inlet points, the percentage decreasing as the differential increased.

I have some data from tests performed by Mr. Young at the Buffalo Test Station in June, 1914, on Mr. Hickstein's plates, in which distant connections were used, and the results vary considerably from those obtained by the author. The tests were conducted at practically constant pressure, ranging from 30 lb. gage on some

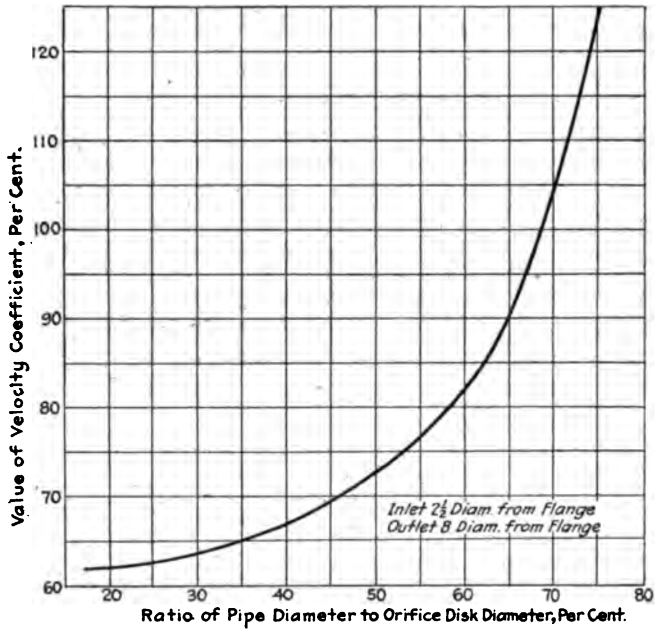


FIG. 6 VELOCITY COEFFICIENTS OF ORIFICE METER DISCS FROM TESTS BY W .T. YOUNG

plates to 50 lb. on others, with increasing differential from zero to 17 in. of water as maximum on the largest plate, and from zero to 27 in. on the smallest plate. The efficiency curve, Fig. 6, had a value of 0.62 at 0.20 ratio of diameters; the curve crossed the author's curve at 0.63 ratio of diameters, and at 0.75 ratio it had a value of 1.234. The gas was measured by pitot tubes and flange connection orifices, which had been calibrated previously against gas holders by natural gas.

In the comparison of the author's with the Charlottenburg t
I cannot understand why the difference in pipe sizes should be ta
into account, inasmuch as the Charlottenburg curve falls out
the author's curve for the 10-in. line for the greater portion o:
distance.

Considerable work on the subject of flow measurement by
orifice method has been done by T. R. Weymouth and by the]
boro Company, and it seems that in no great length of time
orifice meter will be adopted by artificial gas companies as sta
meters.

G. T. VOORHEES said that some years ago he had occasion to n
a number of observations on the flow of gases and liquids thrc
very tiny orifices and at very high pressure. As he remembered,
orifices were made in a ⅛-in. plate and some of them were dr
as small as 1/100 in. The pipe used was about a ⅜-in. pip
that the relation of the orifice to the pipe was large. He found
the coefficient of this discharge was between 60 and 80 per cent

He asked whether the coefficient of discharge should decr
with the size of the orifice.

J. T. WILKIN said that in the manufacture of rotary blo
and gas exhausters there comes up quite often the questio
efficiency tests necessitating the accurate measurement of the air
gases. The paper shows that this measurement is a very deli
thing to make correctly owing to temperatures, water vapors
other conditions.

He recited an instance in which results of tests on a large
holder had been influenced very much by the sun shining on
holder.

He hoped there would be developed some standardized me
of making tests on the efficiency of machines that would be ac
able. He thought the paper was of interest in the developmen
such a method.

CARL C. THOMAS said that the difficulty of making accu
holder tests is very great. When the gas comes up through
piping and is collected in a large vessel the changes in volume d
humidity, and the changes in humidity resulting from change
temperature, necessitate very considerable corrections. He had
quite a good deal to do with holder tests and he believed that

only accurate way to conduct these tests was to wait until the gas inside the holder had attained as nearly as possible a constant temperature. This condition does not occur in the daytime except on some very cloudy days, and then the results are not so good as at night. The best results he had seen were obtained between 11 p. m. and 4 a. m. He would have some doubt about computing the contents of the holder by using some function of the temperatures at the top and bottom of the holder.

With regard to the change of coefficients with different sized orifices, that depends upon the amount of gas or the velocity of gas passing through. It is fairly well established that the coefficient does change as some function of the velocity of the gas flowing through the orifice, but accurate experimental data are not yet available showing over what range orifice measurements are reliable. The condition of the orifice has a great deal to do with the matter. He has been interested in results obtained on natural gas lines connected with the City of Los Angeles, where questions had arisen which were evidently dependent upon the maintenance of orifices in good condition. The velocity flow is very high,—sand is carried along with gas,—and other influences, such as corrosion, deposit, etc., affect the readings.

The bevel of the orifice has a very distinct effect upon the quantity of gas which will flow through it, so that some standard has to be arrived at and maintained in order to insure continuous reliability.

S. A. Reeve said that he had made considerable use of a holder for commercial tests of gas flow and had found that entirely satisfactory conditions could be obtained by using a covered holder, and by keeping a constant flow of water circulating over the bell of the holder from the tank. The air was not pumped in through the water but through a dry pipe so that the humidity did not vary considerably, and the temperature in the building, through the action of the water circulation, seemed to have been kept constant. The holder was about 30,000 cu. ft. capacity.

Prof. A. M. Green, Jr. asked whether with a small differential change in pressure it is not necessary to consider the pressure on both sides of the orifice or some function of that pressure.

THE AUTHOR. As Mr. Bernard states, one type of meter disc
nade of $\frac{1}{8}$-in. stock, with a flat edge for the entire thickness of
plate. Another type, on the other hand, is about $1\frac{1}{2}$ in. thick,
th a flat-edge orifice. What is probably the oldest type in use in
e natural gas industry is about $\frac{5}{8}$ in. thick, with the orifice edge
milar to those tested by the author, i.e., flat for a very short dis-
ance, and then beveled. These last-mentioned discs are case-
ardened before being put in the line. As was brought out in the
iscussion, it is unfortunate that no one type of orifice has been
tandardized.

Mr. Bernard asks whether the size of the orifice increases after
he disc is put in the line. This question is well answered by a
nicrometer measurement, made several months ago, of the diameter
f an 8-in. by 5-in. disc through which possibly fifteen billion cubic
eet (twenty million a day for about twenty months) of natural
as at between 250 and 400 lb. line pressure had been passed. No
ppreciable increase in diameter was found, and the author feels
hat there is little ground for the apparently common misappre-
ension that the orifice diameter increases when a disc is put in use.
ne disc has come to the author's attention with a part of the edge
possibly one quarter the size of a finger nail) chipped out; this
as a case-hardened disc, and the presumption is that the damage
as caused by a rock traveling with the gas at high velocity. It is
nportant, however, to examine the disc at a meter station occasion-
lly, as the effective area of the orifice may be altered by some object
odging in front of it.

The connections used were ordinary $\frac{1}{4}$-in. nipples screwed into
he pipe, care being taken that the nipple did not extend further
han the inside of the pipe. This method of connecting is very
imple, and proved as satisfactory as more complicated methods.
possible explanation of Mr. Bernard's criticism of the fact that
he Charlottenburg curve falls outside (below) the 10-in. Joplin
urve may be found in the error he mentions, involved in not using
he exact size of pipe when d/D was calculated.

A long discussion of the distant connections vs. flange connec-
ions question is hardly in place here. The advantage of having
flat curve "to minimize an error in not having the exact internal
iameter of the pipe" does not enter, unless coefficients found for
ne size of pipe-line are arbitrarily used on some other size of line.

As Professor Walker mentioned, corroborative tests were made, before the Joplin constants for air at atmospheric pressure and with low differentials were used for high pressure gas at varying differentials. Very probably more will be said about these tests in a paper by F. P. Fisher, Mem. Am. Soc. M. E., at the coming Spring Meeting of the Society. The results of Professor Walker's study of these tests are most interesting. His handling of the large number of variables that entered into the tests was a revelation to the engineers associated with him in the work.

Mr. Schell's success, as reported by Mr. Leland, in getting four types of meters to run so closely together, is unusual. It is an indication of the improved methods and ideals now coming into vogue in gas measurement. The statement that it is impossible to get two gas meters in series to read alike is still frequently heard, but there are some engineers nowadays who know better.

The author was interested in Professor Thomas' comment on his experiences with holder tests. In the Joplin tests, it is worthy of note here, the air did not pass through the water, either in entering or in leaving the holder, as several of the contributors to the discussion appear to have in mind. The fact that Professor Thomas also found it advisable to conduct the tests at night is significant. The author, when he described his method of arriving at an approximate temperature of the air inside the holder, did not mean to establish a hard and fast rule that this temperature for *any* holder could be found as a function of two thermometer readings. All he wanted to show was that for the particular holder in question, and especially during the hours at which tests were made, such a function of two other temperatures would give the value required.

The question of the variation of disc coefficient with varying pressures and differentials is mentioned by Professor Thomas. Professor Walker's contribution cites some tests made to study these changes. Apparently, for constant pressure, a variation of the differential between commercial limits (say 5 to 100 in. of water) has no appreciable effect. The effect of varying the pressure between the limits found in pipe lines has a slight effect (not more than one per cent, between 0 and 250 lb.). Sometimes this is taken care of by the use of a "sliding" coefficient for the orifice disc.

Professor Greene asks if it is not necessary to consider the pressure on both sides of the orifice, or some function of that pressure. This question is often asked by engineers who expect to see, as a factor in the equation, a square root of the difference of two squares of pressures. However, for all purposes where the differential reading is not greatly in excess of the values found in practice, the $\sqrt{hP_1}$ is practically equal to Constant times $\sqrt{P_1^2 - P_2^2}$, where $h =$ differential drop in pressure, $P_1 =$ inlet pressure (absolute) and $P_2 =$ outlet pressure (absolute), and the equation involving the factor, $\sqrt{hP_1}$, is in a more convenient form.

In some orifice meter installations, the "static" pressure reading is taken on the outlet side of the meter. This seems more or less a matter of preference, with the restriction, of course, that the same method of connecting must be used as when the disc was originally calibrated.

Mr. Voorhees inquires whether coefficient of discharge should decrease with diameter of orifice. Such is the case not only with discs of the type tested at Joplin, but with orifice meter discs of every type with which the author is familiar. It is not true, however, of an orifice at the end of a pipe-line, or inserted at the end of a drum, and discharging into the atmosphere. With this type of orifice, an increase of the orifice diameter causes a decrease in the coefficient.

The discussion indicates clearly the increased interest in gas measurement by orifices. That the number of commercial orifice meter installations now in use for measuring natural gas is far in excess of 1000 is believed to be a conservative estimate. Since the paper was written, about a year ago, at least one additional type of orifice meter has been placed on the market, and there is good reason to expect that several more will follow.

Probably the most important mechanical development of the past year has been the perfecting of a recording gage especially suited to orifice meter use. This is a compound pressure and differential-pressure gage, i.e., two records are made on the same chart by two pens using different colored inks, and marking over the same scale. This means that just half as many charts have to be handled each day, and also makes possible simplified pipe connections. A "safety" or blow-over device as an integral part of the gage—one that really prevents damage to the gage on excessive differential readings—is a most welcome innovation. This gage is of the mercury float type—the principle towards which the author has always been partial. Altogether, the installation of some sixty or seventy of these gages has gone a long way towards solving his gage troubles of the last three years.

No. 1507

ELASTICITY AND STRENGTH OF STONE-WARE AND PORCELAIN

By James E. Boyd, Columbus, O.
Member of the Society

This investigation was undertaken at the suggestion of Ralph). Mershon,[1] Mem. Am. Soc. M. E., who expressed the belief that tact knowledge of the form of the stress-strain diagrams of clay roducts in tension and compression would make possible the design f insulators of greater mechanical strength and more definite factor f safety.

TEST PIECES

2 The test pieces for most of the experiments were obtained om the manufacturers by Dean Edward Orton, Jr. Of these ieces the porcelain ones were secured from the General Electric Co. rough Lawrence E. Barringer, and the stoneware pieces from the easbey Stoneware Works through F. Whitaker. Pieces used in me of the later tests were made in the laboratory of the Department Ceramics of the Ohio State University under the direction of rof. A. S. Watts.

3 Fig. 1 shows the shape and size of the test pieces furnished by le General Electric Co. and the Keasbey Stoneware Works. The orcelain pieces from the former were all of one composition, while le stoneware pieces from the latter were of 5 different types, desig-ated by the manufacturer as W. M., 132, 2x4, 53, and 3 respectively. he sets from both companies were made with great care, the diam-er of any one rod seldom differing from the mean as much as 01 in. throughout its entire length and most pieces being straight within $\frac{1}{16}$ in.

[1] Member of Research Committee and Chairman of Sub-Committee on Ma-rials of Electrical Engineering.

Presented at the Annual Meeting, December, 1915, of The American ociety of Mechanical Engineers. Contributed by the Research Committee.

4 Fig. 2 shows the form of the test pieces made at the Ohio State University. These were rectangular in section, and with heads tapering gradually to the stem to avoid abrupt change of section. The pieces were made by forcing the clay through a rectangular die, and then carefully cutting to the desired form in the hope of eliminating internal stresses.

5 In Fig. 9 is seen one of the General Electric porcelain rods in a horizontal position, a stoneware rod of quality W. M. vertically at the left and one of the form of Fig. 2 slightly inclined from the vertical near the center.

EXTENSOMETERS

6 All measurements of deformation were made by means of a lever extensometer, shown diagrammatically in Fig. 3. A pair of

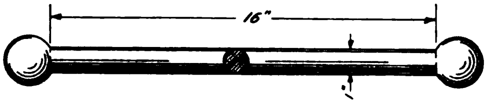


Fig. 1 Round Test Piece

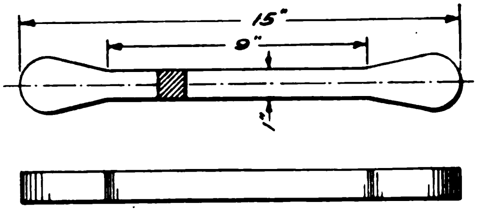


Fig. 2 Rectangular Test Piece

cone pivots act as the fulcrum of the lever and the shorter arm ends with a hardened steel knife edge. The arms are approximately 0.25 in. and 12.5 in. in length respectively. Each instrument is calibrated by using it to measure the pitch of a micrometer screw, and the longer arm is adjusted to make the magnification 50.

7 Fig. 4 shows the clamp which supports two extensometer levers for tension or compression tests, together with the details of one lever. Fig. 5 shows the apparatus as mounted for a tension test. Instead of the scale shown in Fig. 3, a Brown and Sharpe micrometer is used to measure the movement of the longer arm. This arm ends with a black sphere, the contact of which with the polished plane surface of the micrometer may easily be determined to within 0.001 in.

The micrometers are supported by a pair of wooden bars iped to the test piece about 2 in. above the clamp supporting the rs. No correction has been made for the deformation of this ;th, which involves an error of about $\frac{1}{3}$ of 1 per cent when the ed length is 12 in. This error is in one direction in the tension s and opposite in the compression tests.

9 To avoid error when the load is eccentric, care was taken to e the two levers at equal distances from the test piece, and also ave the knife edges and the axis of the test piece in the same ical plane.

10 Before an increment of load was added, the micrometer ws were first turned down an amount sufficient to prevent tact when the load was applied. The levers were raised and ered before taking a reading to eliminate error due to possible ion at the pivots. With proper adjustment, it rarely happened

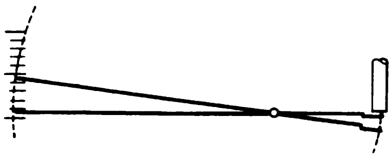


FIG. 3 LEVER EXTENSOMETER FOR MEASURING DEFORMATION

t the sum of the readings of the two micrometers varied as much wo divisions at successive settings under the same load. Frently there was no difference, or, at most, a variation of one sion, corresponding to a deformation of 0.00001 in. in the gaged gth.

11 Contact between clamps and test pieces was made through solder. The contact pieces, C, Fig. 4, are $\frac{3}{8}$ in. wide and the ler practically covers the entire width. Although the solder was ed a little at the middle, it spread out under pressure, causing e uncertainty as to the exact points of contact and the real value he gaged length. It has been assumed that the gaged length nded from center to center of the contact clamps. There is a sible (though not probable) error here of $\frac{3}{8}$ in., which, in a gaged gth of 12 in., might cause an error of 3 per cent in the value of the lulus of elasticity.

12 The rods connecting the upper and lower clamps in Fig. 5 of steel and cause a variation of the zero point when there is appreciable fluctuation of temperature, accounting for some

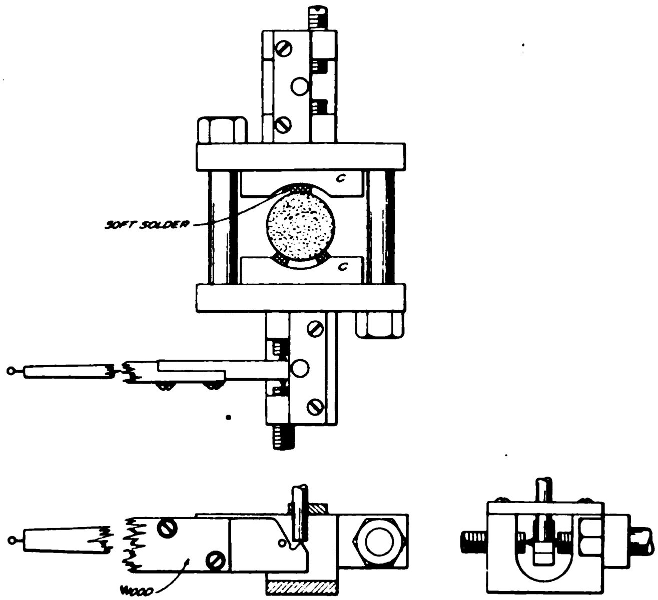


FIG. 4 CLAMP FOR SUPPORTING EXTENSOMETER LEVERS

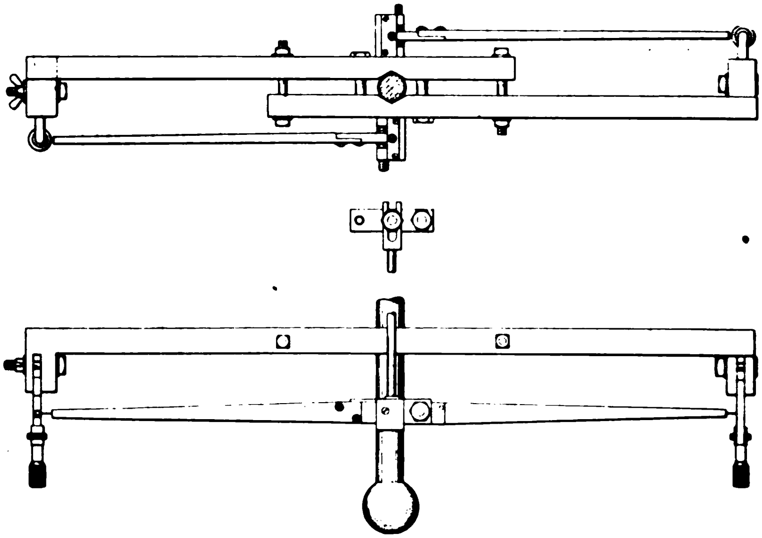


FIG. 5 APPARATUS ARRANGED FOR TENSION TESTS

irregularities of the curves, such as that of the porcelain tension test of Fig. 10.

13 Fig. 6 shows the corresponding rod for compression tests. The levers are placed near the top of the test piece. The knife edge of a lever is placed in contact with the lower end of the short cylinder A. This arrangement causes the knife edge to rise as the load is applied and enables the apparatus to remain on the test piece till final failure. As through most of its length this rod is of wood, little difficulty from change of temperature is experienced with it.

GRIPS FOR TENSION TESTS

14 The first grip tried consisted of a 2-in. pipe coupling with a reducer bushing at the end. The bushing was cut in halves so that it could be fitted around the test piece below the head and then screwed into place. One of these grips is shown in Fig. 9 at the

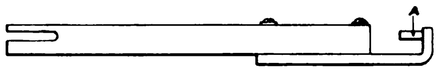


FIG. 6 DISTANCE ROD USED IN COMPRESSION TESTS

lower end of the vertical rod at the left. Soft material, usually leather, was placed between the bushing and the head of the test piece. A rod with a spherical head was passed through a second bushing (not shown) at the other end of the coupling, giving the effect of a ball and socket joint.

15 Upon using this grip in tension tests, the extensometer readings revealed considerable eccentricity of loading due to lack of perfect symmetry in the heads of the test pieces and want of uniformity in the soft material against which the heads pressed. This eccentricity might have been eliminated by supporting each grip on pairs of adjustable knife edges at right angles, but on account of the inconvenience and expense this arrangement was not adopted.

16 Fig. 7 shows another form of grip used for most of the tension tests of the cylindrical pieces. This consists of two steel plates held together by three bolts, one of which is omitted from the drawing. A conical depression in each plate holds some soft material—lead, leather, or rubber—which, in turn, grips the head of the test piece. The upper bolt passes through the pulling rod B, forming a hinge connection. The rod B may be moved along the bolt to adjust slightly and a further adjustment may be made by loosening the

bolts and changing the alignment of the test piece and grip. With this arrangement it is practicable to so reduce the eccentricity that the elongations as shown by the two micrometers become nearly

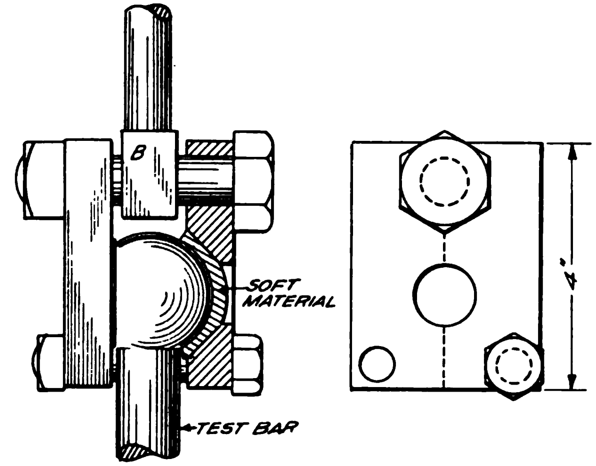


FIG. 7 GRIP USED IN TENSION TESTS OF ROUND RODS

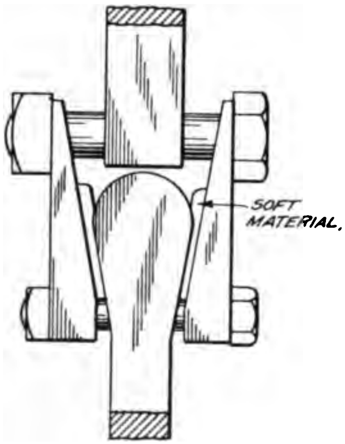


FIG. 8 GRIP USED IN TENSION TESTS OF RECTANGULAR RODS

equal. There still remains, of course, the possibility of eccentricity at right angles to the plane of the instruments, which may only be determined by means of a third lever.

7 With this grip it was found that practically all the pieces broke at the head. (Four of these shown in Fig. 9 illustrate the acteristic failure.) As it was thought that this failure might be ed by too abrupt change of section, the form of test piece shown ig. 2 was developed. The clamp for this piece is shown in Fig. 8 n Fg. 9. Sheet lead was first tried as the soft material to ibute the stress. With a bearing area of about 4 sq. in. the lead d, principally by shear, when the pull was 2900 lb. Leather d at 2400 lb. It is evident that the slope of the head of this

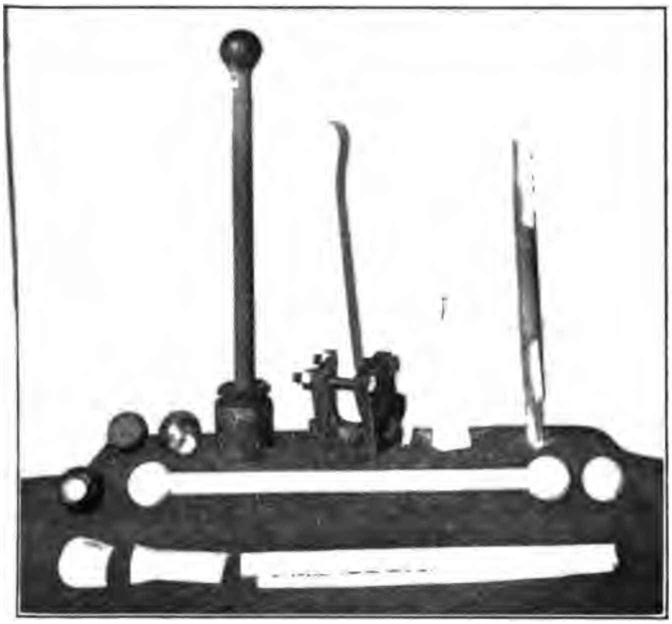


FIG. 9 TEST PIECES, SHOWING GRIPS AND TYPICAL FRACTURES

n of test piece is too small for these materials, unless the ratio he bearing area to the cross section of the stem is considerably eased. Copper and aluminum were found too hard, causing porcelain to chip locally.

18 These test pieces all failed at the head, but at some distance ve the stem. Fig. 9 shows the head and part of the stem of one e with a characteristic failure. Sometimes the fracture was a le below the lower edge of the bearing area, sometimes a little ve.

19 Two explanations are suggested for this method of failure. One is that the stress is concentrated in the outer fibers of the head and is really of greater intensity here than in the smaller section of the stem, where it has become uniformly distributed. As stress is transmitted from the outer fibers inward in the form of shear, the shearing strength and modulus become factors. The other explanation is that the tensile stress developed as a result of the transverse compressive stress makes the total stress at the section of failure much greater than at the stem. Assuming that the unit compressive stress is 10,000 lb. per sq. in. and that Poisson's ratio is $\frac{1}{4}$,

TABLE 1 TENSION TEST OF STONEWARE 53 I

Area, 0.77 sq. in. Feb. 7 1913.
Gaged length, 12.57 in.

Load in Lb.		Micrometer Reading			Elongation in 0.00001 In.		
Total	Per Sq. In.	Left	Right	Sum	From 10 lb.	From No Load	Unit
8	10	503	384	887	0	2	....
154	200	526	404	930	43	45	3.57
308	400	558	415	973	86	88	7.00
462	600	567	448	1015	128	130	10.34
616	800	588	467	1055	168	170	13.52
770	1000	609	489	1098	211	213	16.96
924	1200	635	501	1136	249	251	19.97
1001	1300	648	509	1157	270	272	21.64
1078	1400	665	521	1186	299	301	23.95
1155*	1500	....	....	....	....	....	....

* Broke before reading could be taken.

the compression alone develops a tensile stress of 2500 lb. When this is added to a direct tensile stress of perhaps 2000 lb. per sq. in., we get a stress in the head of 4500 lb. per sq. in., which easily accounts for the failure. It is probable that both of the above factors are present and that, combined with more or less eccentricity of loading, they each contribute to the failure.

RESULTS OF TENSION TESTS

20 Table 1 gives the results of one tension test. In this test several sets of readings were made with loads up to 800 lb. per sq. in., in order to adjust the grips to reduce eccentricity. In these preliminary runs several readings were taken at each load. As these showed little variation, a single reading only was taken at each load for the last set which is given in the table.

21 The third column of micrometer readings is the sum of the dings of the two instruments. The micrometers read in 0.001 in., d the lever magnification is 50, so that differences between suc-ssive readings of either micrometer represent 0.00002 in., and fferences between figures in the column of sums are in 0.00001 in.

22 The initial load was 10 lb. per sq. in. The elongation for a ad increment of 200 lb. is about 0.0004 or 0.00002 for 10 lb The a zero reading in the column of sums may be taken as 885 instead 887.

23 The results of this test are shown graphically by the lower rve of Fig. 10. The diagram is practically straight up to 1300 lb.

TABLE 2 TENSION TESTS OF STONEWARE AND PORCELAIN

Lb. per Sq. In.	Unit Elongation in 0.00001 In.			
	Porcelain I	Stoneware		
		132 III	2x4 I	53 I
200	2.06	2.24	3.28	3.57
400	3.96	4.72	6.32	7.00
600	6.10	7.25	9.20	10.34
800	8.24	9.20	12.40	13.52
1000	9.90	11.60	15.68	16.96
1100	.....	12.48	17.36	.....
1200	11.97	14.00	18.40	19.97
1300	.....	15.08	Broke	21.64
1400	14.34	16.32	below	23.95
1500	.....	17.25	1400 lb.	Broke
1600	15.93	18.25		
1700	.....	19.57		
1800	18.07	20.73		
1900	.....	21.85		
2000*	19.97	Broke below 2000 lb.		

*For porcelain above 2000 lb. per sq. in. see Table 3.

r sq. in. The broken line from 1400 to 1500 lb. per sq. in. is in-ided to show that it carried the load represented by the ordinate t that no deformation reading was secured above 1400 lb. per in. The modulus of elasticity, calculated from the single reading 1000 lb. per sq. in., is

$$E = \frac{1000}{0.000170} = 5,900,000$$

24 Besides stoneware 53 I, Fig. 10 represents 3 other tension sts, porcelain I, stoneware 132 III and stoneware 2x4 I. In each se the particular tension test represented is the one showing the ghest ultimate strength, as may be seen from Table 6.

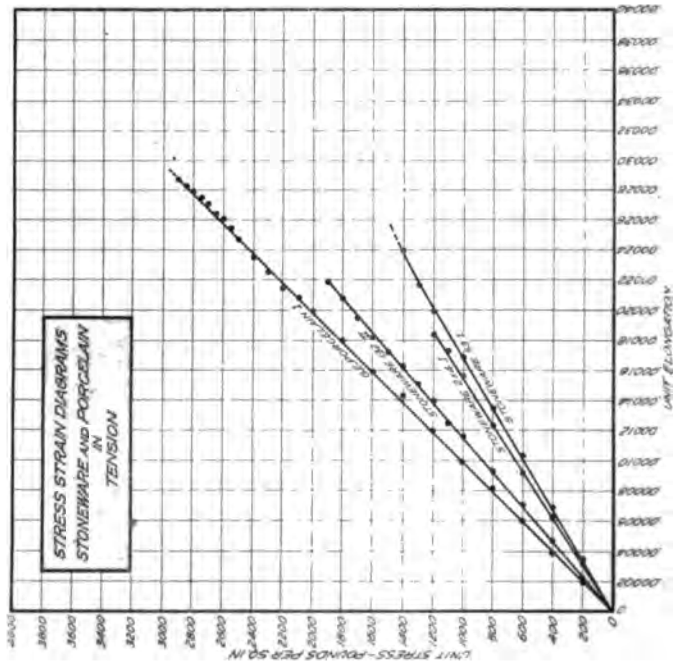


FIG. 10 STRESS-STRAIN DIAGRAMS, PORCELAIN AND STONEWARE

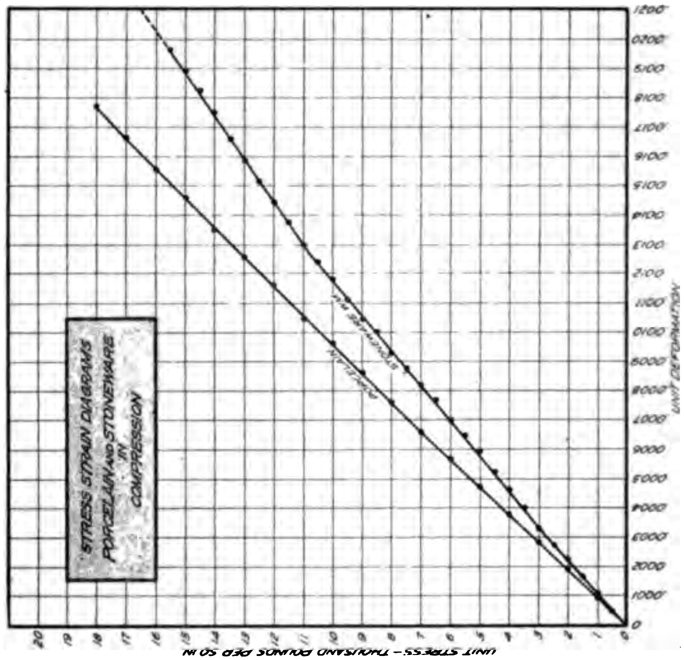


FIG. 11 STRESS-STRAIN DIAGRAMS, PORCELAIN AND STONEWARE

25 It will be noticed that each of these diagrams, except stoneware 53 I, is practically a straight line giving no definite indication of a true elastic limit. The double curve in the porcelain diagram is probably due to temperature changes, as there was considerable draft on the apparatus at times during this run.

26 Table 2 gives the data from which all the curves of this figure were plotted, except those of porcelain above 2000 lb. per sq. in.

27 Table 3 gives the results of tension tests of 4 rods of General Electric porcelain. In the case of rod I readings were taken at

TABLE 3 TENSION TESTS OF PORCELAIN

Lb. per Sq. In.	Unit Elongation in 0.00001 In.					Modulus of Elasticity
	I	II	III	IV	Average	
200	2.06	1.83	1.90	1.96	1.94	10,310,000
400	3.96	3.74	3.96	4.16	3.95	10,130,000
600	6.10	5.48	5.86	6.19	5.91	10,150,000
800	8.24	7.24	7.69	8.47	7.91	10,110,000
1000	9.90	9.07	9.91	10.35	9.81	10,190,000
1200	11.97	10.75	11.81	12.23	11.69	10,260,000
1400	14.34	12.89	13.79	14.43	13.86	10,100,000
1600	15.93	14.57	16.24	16.39	15.78	10,140,000
1800	18.07	16.72	17.75	18.74	17.82	10,100,000
2000	19.97	18.39	19.89	20.71	19.74	10,130,000
2100	20.76	Broke	.....	.....		
2200	21.39		21.31	Broke		
2300	22.58		.....	*		
2400	23.53		23.30			
2500	24.64		24.72			
2600	26.16		25.44			
2700	27.10		26.70			
2800	27.97		27.89			
2900	28.76		28.68			
2950	Broke		.....			
3000			Broke			

*Broke at extensometer clamp at point where solder was crushed so that brass was in contact with porcelain.

50 lb. intervals above 2500 lb. per sq. in. These are shown in Fig. 10, but only half of them are given in the table. This table gives the average unit elongation of the 4 pieces and the modulus of elasticity calculated from this average. It will be seen from the table that rod III is practically the same as rod I. The diagram for rod II is shown in a small scale in Fig. 14. Its curve seems to bend slightly.

28 Two tension tests of Ohio State bars of the form shown in

Fig. 2 gave practically straight line diagrams up to 2700 and 2900 lb. per sq. in. respectively, with a modulus of elasticity of about 9,000,-000. The lower modulus is probably due to a lower temperature of burning.

29 Fig. 15 gives tension diagrams for two rods of stoneware W.M., and Fig. 16 for three rods of stoneware 3. All these diagrams are curved slightly, indicating that the true elastic limit has been exceeded.

COMPRESSION TESTS

30 For compression tests rods were cut off to a length of 16 in.,

TABLE 4 COMPRESSION TESTS OF PORCELAIN I AND STONEWARE W.M.II

Lb. per Sq. In.	Unit Compression in 0.00001 In.		Modulus of Elasticity	
	Porcelain	Stoneware	Porcelain	Stoneware
1000	9.4	11.2	10,640,000	8,930,000
2000	18.9	22.2	10,580,000	9,010,000
3000	28.1	33.3	10,680,000	9,010,000
4000	37.7	46.4	10,610,000	8,620,000
5000	46.8	59.1	10,680,000	8,460,000
6000	56.6	69.6	10,600,000	8,620,000
7000	65.7	81.7	10,650,000	8,570,000
8000	75.7	92.7	10,570,000	8,630,000
9000	85.9	104.0	10,480,000	8,650,000
10000	95.8	118.2	10,430,000	8,460,000
11000	104.8	129.8	10,500,000	8,470,000
12000	116.0	144.7	10,340,000	8,220,000
13000	125.7	158.2	10,340,000	8,220,000
14000	135.3	174.0	10,350,000	8,050,000
15000	145.6	189.1	10,300,000	7,930,000
15500	.....	196.5		
16000	155.4	Levers		
17000	166.4	removed		
18000	177.3	Failed at 21500		
19000	Failed	lb. per sq. in.		

and 12 in. was used as the gage length. Sheet lead was placed between the ends of the test piece and the plates of the testing machine. A preliminary test was made on a rod of stoneware 3. This rod, 8 in. in length, failed suddenly with a sharp noise under a load of 15,000 lb. per sq. in. It flew into many small fragments with the principal planes of fracture parallel to its length.

31 Measurement of elongation was made on two pieces, both of which failed by splitting lengthwise. Both rods are shown in Fig. 9.

The porcelain rod, in a horizontal position in the front of the picture has a single piece split off for almost its entire length. The split off piece extends from the top of the rod to the position of the lower extensometer clamp. The other rod of stoneware W.M. shown standing at the right of Fig. 9 failed at the ends. Table 4 gives part of the results of the compression tests of these two pieces. Fig. 11 gives the diagram for Table 4. The porcelain curve varies little from a straight line. There is a slight change of slope at 7000 lb. per sq. in. which may indicate a true elastic limit. The stoneware curve bends slightly at 3000 lb. and shows considerable change of slope at 11,000 lb. per sq. in.

BENDING TESTS

32 Fig. 12 shows the arrangement of the apparatus for bending tests. The test bar is supported at two points B and B' and loaded equally at two symmetrical points A and A'. The extensometer lever

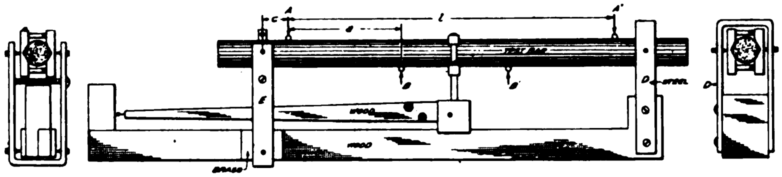


Fig. 12 Apparatus Arranged for Bending Tests

is carried on a wooden beam, so mounted that it practically connects two points on the neutral surface of the test bar at equal distances c outside the load points A and A'. The connection between test bar and beam is made through two steel stirrups, of which the right one, D, is rigidly fastened to the beam and the left one, E, is joined through a pair of cone pivots. Brass clamps gripping the test bar are connected to the stirrups through cone pivots. This arrangement keeps the wooden beam parallel to the line joining the upper pivots and at a constant distance therefrom. At a point midway between the supports a third clamp is attached to the test bar and connected to the extensometer lever by a short steel rod. As the middle of the test bar rises when the load is applied, the test may be carried to failure of the piece without danger to the apparatus. The movement of the longer arm of the lever is measured by a micrometer attached to the wooden beam.

33 With this apparatus, the error due to temperature change is

very small, and the curves obtained are more satisfactory than those of the tension tests.

34 In most of the tests the distance l between the loads was 12.6 in., and the distance a from A to B and from A' to B' was 4.2 in., so that the deflection of the middle of the bar above the line joining A and A' is the same as that of a beam supported at the ends and loaded at the third points. This deflection is

$$\frac{23\,P\,l^3}{1296\,E\,I}$$

where P is the total load, and I the moment of inertia of the cross-section. The additional deflection at a distance c from A is the product of the length c multiplied by the slope of the tangent to the bar at A. With a beam loaded at the third points, the slope at the supports is

$$\frac{P\,l^2}{18\,E\,I}$$

TABLE 5 BENDING TEST OF STONEWARE 2x4 I

Diameter, 0.945 in. to 0.955 in. Dec. 5, 1913
Average diameter 0.949 in.
l = 12.6 in.; a = 4.2 in.; c = 1.2 in.

Load in Lb., P	Deflection in In., y	Unit Stress in Outer Fibers	Unit Deformation in Outer Fibers 0.00001 In.	Modulus of Elasticity
12	0.00208	300	4.5	6,670,000
22	0.00382	551	8.3	6,660,000
32	0.00558	801	12.1	6,650,000
42	0.00728	1051	15.7	6,680,000
52	0.00904	1302	19.5	6,660,000
62	0.01080	1552	23.4	6,640,000
72	0.01256	1802	27.2	6,630,000
82	0.01442	2052	31.2	6,580,000
92	0.01618	2303	35.0	6,580,000
102	0.0180[illegible]	2553	39.0	6,540,000

The length c as used was 1.2 in., so that the total deflection y is given by the equation

$$\frac{E\,I\,y}{P} = \frac{23\,l^3}{1296} + \frac{1.2\,l^2}{18} = \frac{376.2\,l^2}{1296}$$

$$E = \frac{46.08\,P}{I\,y}$$

35 Table 5 gives the results of one test of a stoneware rod. The stress in the outer fibers has been calculated in the usual way

d the modulus of elasticity by means of the formula above. In der to draw curves which may be compared with the tension and mpression diagrams, the unit deformation in the outer fibers for at portion of the bar between the supports is given in the fourth lumn.

36 Fig. 13 gives the diagram for Table 5, together with those of wo other kinds of stoneware and one of porcelain—the same materials hich are shown in tension in Fig. 10. It will be noted that the iagram of porcelain is straight while those of stoneware 132 and 53 end a little. The diagram of stoneware 2x4 seems to be straight hen drawn to this scale, but the figures of Table 5 show that there a slight curvature. Since only the outer fibers in the middle hird of the test bar reach the calculated stress, it is evident that e diagram from the bending test should be nearer a straight line an the corresponding tension curve.

COMPARISON OF RESULTS

37 Fig. 14 gives compression, bending, and tension diagrams for eneral Electric porcelain. It will be seen that the compression irve for porcelain I has the same slope as the bending curve of orcelain II, while the tension curve for II has a little higher and e tension curve for I a little lower slope. Considering the ariation in the tension tests as shown by Table 3, together with e fact that the curve for bending depends upon tension and comression jointly, it is probable that the modulus of elasticity in nsion differs little, if any, from the modulus in compression.

38 If the bending curve for this porcelain be, as it seems, a raight line with the same slope as the compression curve, it folws that the ultimate tensile stress is the same as the modulus of pture, or 7800 lb. per sq. in. If the bending curve deviates a tle from a straight line, but not enough to show without greater finement of measurement, there may be considerable deviation in e upper portion of the tension curve so that the real stress at rupire may be somewhat below 7800 lb. per sq. in. In any case, it is ident that the tension tests did not develop the full tensile strength.

39 Fig. 15 shows tension, bending, and compression for stoneare W. M. Here, again, the compression and bending diagrams e parallel while one tension curve has a higher and the other a wer slope. There is a decided bend, however, in both tension and nding diagrams, and the tensile stress developed is nearly $\frac{2}{3}$ of the odulus of rupture.

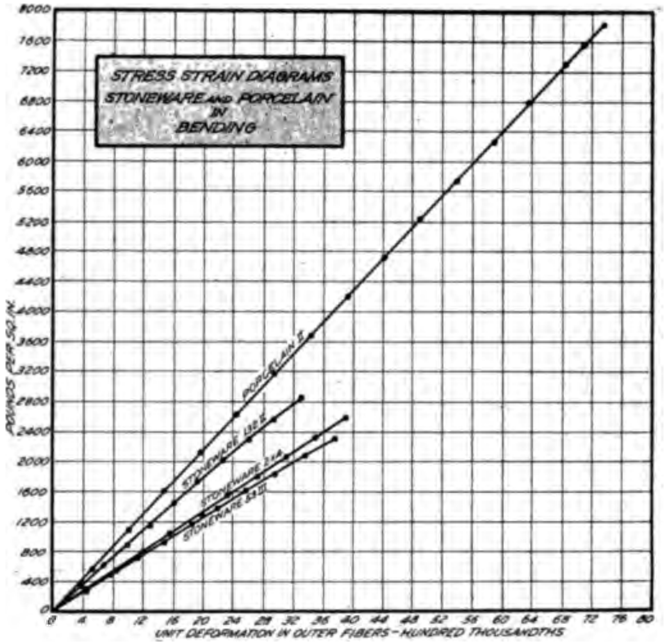


Fig. 13 Stress-Strain Diagrams, Bending Tests

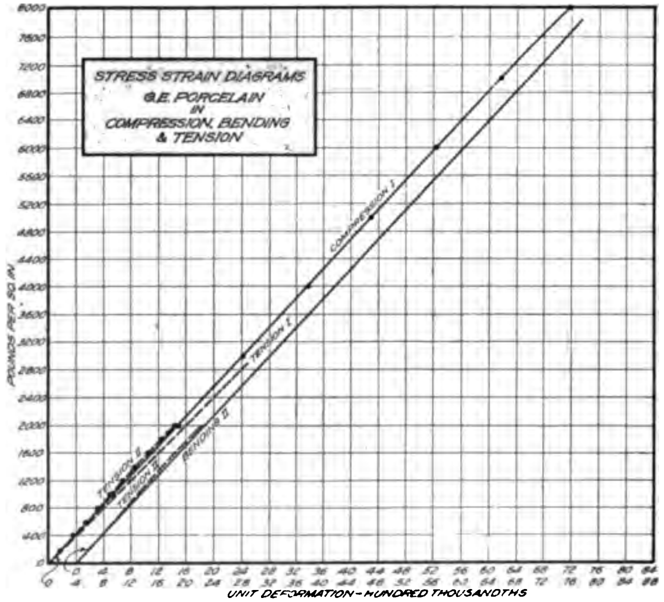


Fig. 14 Stress-Strain Diagrams for Porcelain

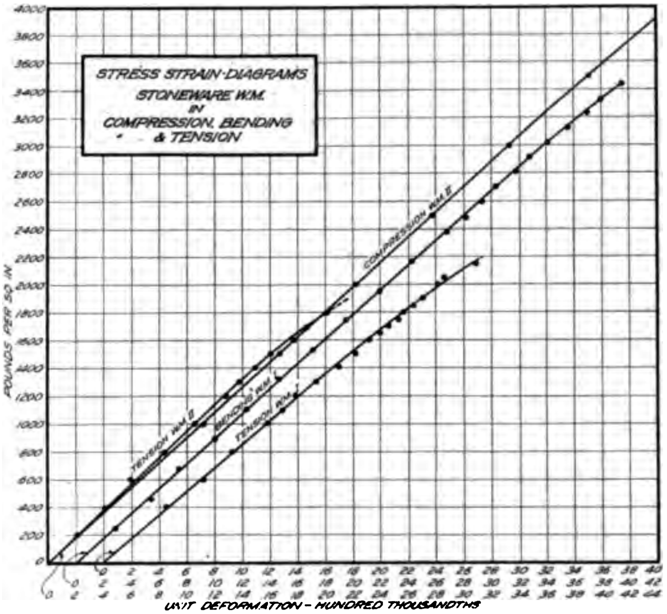


FIG. 15 STRESS-STRAIN DIAGRAMS FOR STONEWARE

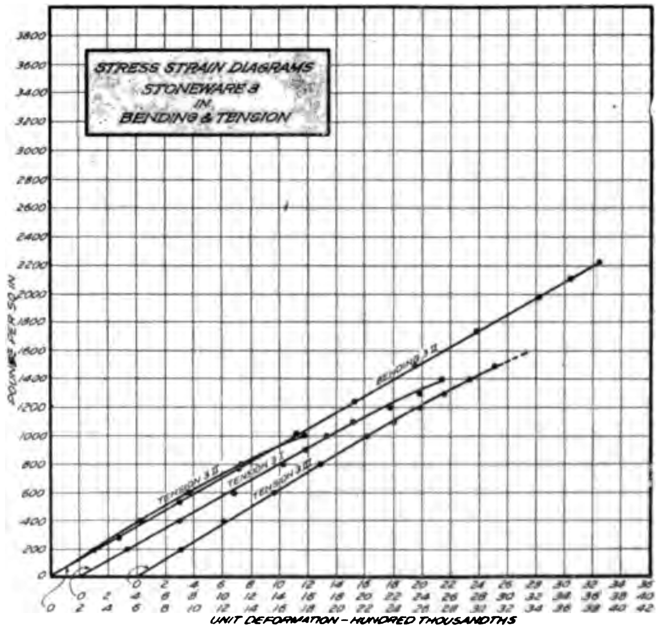


FIG. 16 STRESS-STRAIN DIAGRAMS FOR STONEWARE

40 Fig. 16 gives tension and bending diagrams for stoneware This material behaved like stoneware W. M. except that the diagra from the bending test shows no certain deviation from a straig line.

TABLE 6 SUMMARY OF TESTS OF STONEWARE AND PORCELAIN

Test Bar		Kind of Load	Unit Deformation at 1000 Lb. per Sq. In.	Modulus of Elasticity	Last Unit Deformation Read	Unit Stress	
						At Last Reading	At Failure
Porcelain							
G. E.	I	tens.	0.000099	10,100,000	0.000288	2900	2950
G. E.	II	tens.	91	11,000,000	184	2000	2100
G. E.	III	tens.	99	10,100,000	187	2900	3000
G. E.	IV	tens.	103	9,700,000	207	2000	2200
G. E.	I	comp.	0.000094	10,600,000	0.001773	18000	19000
G. E.	II	bend.	92	10,800,000	736	7822	7822
G. E.	III	bend.	97	10,300,000	646	6448	6448
Ohio S. U.	III	tens.	0.000116	8,600,000	0.000330	2600	2700
Ohio S. U.	IV	tens.	121	8,300,000	271	2200	2400
Ohio S. U.	IX	tens.	108	9,200,000	329	2800	2900
Stoneware							
53	I	tens.	0.000170	5,900,000	0.000240	1400	1500
53	II	tens.	166	6,000,000	231	1400	1500
53	I	bend.	160	6,250,000	441	2666	2666
53	II	bend.	158	6,300,000	376	2291	2291 +
2x4	I	tens.	0.000157	6,400,000	0.000184	1200	1200 +
2x4	II	tens.	158	6,300,000	178	1100	1200
2x4	III	tens.	155	6,450,000	155	1000	1100
2x4	I	bend.	150	6,700,000	390	2553	2553
W. M.	I	tens.	0.000118	8,500,000	0.000269	2150	2200
W. M.	II	tens.	105	9,500,000	201	1800	1900
W. M.	III	tens.	109	9,200,000	191	1800	1900
W. M.	I	bend.	112	8,900,000	395	3444	3444
W. M.	II	comp.	112	8,900,000	1965	15500	21500
3	I	tens.	0.000173	5,800,000	0.000244	1400	1400 +
3	II	tens.	177	5,650,000	177	1000	1100
3	III	tens.	161	6,200,000	250	1500	1600
3	II	bend.	169	5,900,000	384	2231	2231
132	I	tens.	0.000113	8,900,000	0.000181	1600	1650
132	II	tens.	127	7,900,000	152	1200	1250
132	III	tens.	116	8,600,000	219	1900	1900
132	II	bend.	112	8,900,000	328	2848	2848

41 Table 6 gives a summary of all the tests in which a complet series of readings were taken. The third column of this table give the unit deformation at the unit stress of 1000 lb. per sq. in. Wher no reading was taken at this load, the deformation has been com

ted by interpolation from the two nearest readings. The fourth lumn gives the modulus of elasticity calculated from the stress of 00 lb. and the deformation of the third column. The fifth and th columns give the last unit deformation read and the correspond- g unit stress. The modulus of elasticity calculated from these gures, or from the difference between them and the corresponding formation and stresses at 1000 lb., will show whether the curve the material deviates from a straight line.

CONCLUSIONS

42 From the results of these tests, the following conclusions ay be drawn:

a The modulus of elasticity of stoneware and porcelain is practically the same in tension and compression. Its value may be obtained conveniently by a bending test.

b The modulus of elasticity of porcelain is about 10,000,000. The modulus of stoneware ranges from 6,000,000 to 9,000,000, depending on the material.

c The compressive strength of porcelain and high grade stoneware in a column 16 in. long and 1 in. in diameter is about 20.000 lb. per sq. in. The stress-strain diagram is practically straight up to 7000 lb. per sq. in.

d The tensile strength of porcelain is above 3000 lb. per sq. in. The diagram is a straight line up to this stress. The tensile strength of stoneware ranges from above 1100 to above 2200 lb. per sq. in. The stoneware of the greater modulus has the greater strength.

43 These tests failed to develop the full tensile strength of e material. Judging from the bending tests and the form of the agrams, it is probable that the real tensile strength is about twice great as the figures here given.

ACKNOWLEDGMENT

44 In addition to the firms and individuals mentioned in the troductory paragraphs, the thanks of the author are due to Pro- ssors William T. Magruder, William A. Knight, Robert Meiklejohn, nd Edwin F. Coddington.

DISCUSSION

Ralph D. Mershon (written). In suggesting to Professor Boyd e investigation he has so excellently carried out on the mechanical properties of ceramic materials, I had in mind more than the bearing of the subject upon the matter of insulation. For a long time it has seemed to me there were uses for ceramic materials where they are not now used, presumably because of the lack of information relative to their mechanical properties. One important possibility is that of the employment of clay pipes in many places where iron or steel ones are now used, especially underground. Of course such pipes are regularly used for sewers and drains, and I believe that to some small extent they have been employed for low pressure gas mains. But their use seems to have been confined to cases where the internal pressure was little or nothing. It would seem that their durability might be availed of in uses involving considerable internal pressures, if more were known of the characteristics and the possible uniformity of the material composing them.

So far as I know, this work of the author is the first systematic investigation ever made of the mechanical properties of ceramic materials. It is to be hoped that it is a forerunner of further work along this line, both by Professor Boyd and others, and that we may have, before long, the results of investigations as to the influence of various mixtures, methods of working, and degrees of firing, upon the mechanical properties of ceramic materials, and upon the uniformity with which products composed of such materials can be produced.

R. C. Carpenter (written). The investigation described by Professor Boyd is the first that I have been able to learn about which gives information of the strength of stoneware and porcelain under conditions of use. The information given by the paper is of great value in all the arts where porcelain is employed either as a structural member or for the purpose of acting as an insulator under conditions where it must sustain a considerable load.

Porcelain is a difficult material to test in the ordinary standard testing machine because of its brittleness, so that the methods described in the paper will be found useful for all future tests of this material made in any laboratory.

The Society is to be congratulated for the success of this investigation, which is one of the results of the activity of its Research Committee. The field covered by the investigation gives information of the strength when subjected to tension, bending or compresion strain, and for that reason is complete to a remarkable degree, specially when the difficulties which had to be overcome in making he tests are considered.

L. E. BARRINGER[1] (written). The mechanical strength of porcelain is satisfactory in some respects and decidedly unsatisfactory in thers. The most consistent property is the resistance to crushing nd the most erratic and unsatisfactory is the toughness, or resistnce to the sudden shock of impact or vibration.

A few years ago I stated that in our experience with vitrified orcelain used for electrical insulation we had found an average ompressive strength of 20,000 lb. per sq. in., and a tensile strength f from 900 to 1800 lb. per sq. in. The tension tests were made pon briquettes of the standard shape used in cement testing, with he use of special clips and cushioning materials, and the compresion tests on 2-in. cubes. Professor Boyd's paper indicates a comressive strength for porcelain of the same value; I think 20,000). per sq. in. is a safe basis for the calculations of designing ngineers.

The comparatively high and uniform compressive strength for orcelain, as compared to its other mechanical strengths, has led ngineers to take advantage of this property as much as possible, nd certain types of line insulators are designed to bring the strains pon the porcelain as nearly to compressive as possible.

In several instances porcelain, because of its high insulating alue and weather resistance, has been found quite the best material or insulation in electric railway equipment. In this particular eld, however, porcelain can only safely be used in compression, nd would meet with early failure if subjected to tensile, transverse, r impact strains.

In the matter of determining tensile strength the conditions are nore complex than in developing compressive strength. The author hows it is quite necessary to devise special clips and cushioning naterials to develop the true tensile strength, and in his tabulation f conclusions states that even his very carefully conducted tests ailed to develop the real tensile strength of either porcelain or toneware.

Inasmuch as the paper states only 3000 lb. per sq. in. could be eveloped, and this with the use of special clips, cushioning material

[1] Engr. of Insulations, Genl. Elec. Co., Schenectady, N. Y.

and manipulating, although the true strength was probably 7000 lb. per sq. in., it is suggested that a working figure of 1500 lb. per sq. in. be taken for tensile strength.

One point in connection with the mechanical strength of porcelain is the effect of temperature changes. In the kilns a marked difference of strength will be found, depending upon the rate of cooling. In the finished product temperature changes even within the range of 20 to 90 deg cent. have an influence upon the mechanical strength. In general, sudden temperature fluctuations cause lowering of mechanical strength; and temperature changes in the finished porcelain, even though brought about at a comparatively slow rate, will have a harmful effect within certain ranges. Porcelain baked at 90 deg. cent. exhibited a decided weakness upon cooling.

The effect of temperature changes is more pronounced as the rate of fluctuation increases, and has the greatest effect upon the toughness. The tensile and transverse strengths are next most seriously affected, and the compressive strength least of all.

F. M. Farmer presented a written discussion. He stated that this constructive paper represents a considerable amount of very careful and obviously reliable work which will be particularly appreciated by all electrical engineers who have had to do with porcelain insulators. There is a remarkable dearth of reliable information of this character on porcelain.

The paper is, however, of interest for another reason, namely, it emphasizes the small amount of research work which is being done on standard engineering materials. The importance of this general class of research work is frequently overlooked. The ultimate commercial engineering value of such data is often, and perhaps usually, much greater than that obtained from researches which have terminated in more startling results. Many of our common engineering materials are constantly changing, due to improvements in the methods of manufacture, different methods of treatment, etc., and consequently their properties should be redetermined occasionally.

Porcelain, for example, has been improved very greatly in the last fifteen years and yet, despite the enormous increase in its use for purposes where knowledge of the physical properties is important, no reliable data are available on such an important property ensile strength. In this paper it is concluded that the tensile ngth of porcelain is above 3000 lb. per sq. in., whereas the usual ces of information give the value as 650 to 2200 lb. per sq. in. se latter figures were undoubtedly obtained many years ago and not apply to the modern porcelain.

It will be conceded that no general conclusions as to the strength orcelain can be drawn from the small number of tests described. ile four or five specimens of a given sample of porcelain may be cient, it is necessary to test not only samples of as many different makes as possible, but samples of different thickness, and ples which have been subjected to various temperatures and ods in firing, etc.

Nothing is stated as to the kind of porcelain tested. Presumably what is known as electrical porcelain, but further information

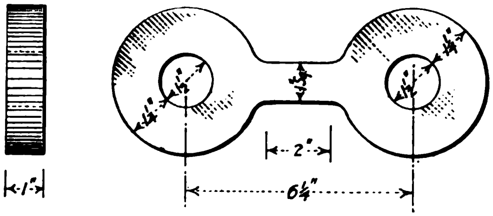


FIG. 17 FORM OF TEST PIECE FOR MAXIMUM STRENGTH

ld have been desirable, such as the kind of clays used, methods ring, temperature of the furnace and time of firing.

A form of grip and of test specimen which would develop the imum strength of the porcelain would be desirable. While the lusions in regard to the tensile strength drawn from the bend- and compression tests are probably correct, data obtained by ct tests are more convincing. It would seem off-hand that the angular test piece with longer bearing surfaces at the ends ld have developed the maximum strength of the porcelain.

T. D. LYNCH[1] (written). We have done most of our testing of celain in the final piece as made for service. The sections of pieces are more or less irregular and for that reason difficult educe to a square inch section. We have made a few tests, how-, on samples of porcelain with holes in either end for the pur- of attaching clevises, as shown in Fig. 17. Such samples showed

Research Engr., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

approximately 1500 lb. per sq. in. tensile strength, and about 15,000 lb. per sq. in. compressive strength, or ten times the tensile value. These tests are not at all conclusive. Our full-size tests on the finished piece indicate that the strength of porcelain is probably very much influenced by internal stresses, and it would be interesting to know of any study made by the author along this line.

An investigation of the internal stresses in brasses and bronzes was conducted by P. D. Merica and R. W. Woodward and reported before the American Institute of Metals in 1915. Perhaps some similar test could be made on porcelain to determine if possible a heat treatment that will eliminate internal stresses or at least reduce them to a minimum.

JOHN F. ANCONA thought there was a possibility that the peculiar failure of the pieces in the tensile tests might be attributed to some condition of internal stress.

ELLIOTT H. WHITLOCK said that in connection with the effect of temperature of cooling on the homogeneity of the sample he had found that with a certain material the temperature rise curve of the firing had a great deal to do with the final strength. If the temperature rose to a certain point and was then allowed to drop and then again brought up to a higher point, the material was very much weaker than if the same temperature had been reached by a continuous rise. He thought that possibly the rate of firing might have as much to do with the tensile strength of porcelain as the rate of cooling.

PERCY H. THOMAS[1] (written). The author found that fair and reliable results were difficult on account of the great tendency in the test machine for the stress to be applied unevenly to the porcelain section, and also on account of the extreme local stresses at the point of application. While great care was taken in this work to eliminate such effects, other forms of specimens and other methods of holding the test pieces might be much better.

Porcelain is used in practical service in certain cases in such a way as to be called upon to support very material stresses, and does so very reliably. But in practice the shape of the porcelain pieces and the method of support are such as to avoid the difficulties found in the tests described. For example, the high tension sus-

[1] 2 Rector St., New York.

ıion type porcelain insulator serves to give mechanical support line conductor, at the same time insulating it. The porcelain , as far as we are here concerned, is a cup in which an iron pin ıstened by filling the cup with Portland cement. The cup is on the outside by a metal cap into which it enters and to which so is secured by Portland cement. When the inner pin is pulled in the testing machine, the metal cap being made fast to the ting jaw of the machine, the porcelain cup is pulled apart, leaving its bottom portion in the cap, and the rim following the pin. characteristic fracture is that of tension stress. A porcelain 1 in. internal diameter and with a wall ¾ in. thick may resist 'ess as high as 13,000 lb. in some instances, which is equivalent 500 to 3000 lb. per sq. in., assuming an even distribution. But ıis stress must be greater nearer the surface of the wall than on interior, the true unit stress must be much higher. This cor-onds fairly well with the test values of the paper. Furthermore, ite of the very serious stresses to which these insulators are ected, there are no signs of brittleness or unreliability in service his account.

'he conclusion would be that, when properly used, porcelain is ry strong and reliable material, and by no means deserves the tation it has been given by its behavior in tests of porcelain and narrow tubes.

OHN A. BRASHEAR. For nearly twenty-five years I have ght of using or attempting to use a porcelain disc of a special m instead of glass for a reflecting telescope. Silvered glass ors were introduced some forty years ago; that is, silvered on front instead of the back surface, and I have thought in the of large glasses, the disc could be made of porcelain, which l be given an approximate curvature before the enamel was sited upon it, and if this enamel could be put on thick enough, ground, fined and polished, and then the surface silvered, it d make a very excellent substitute for the glass mirror. I do know whether any studies of the coefficient of the expansion contraction of porcelain have been made, but in the glass disc an important factor, so that in the case of large telescopes it very great trouble. In the case of the 72-in. reflector which re now making for the Canadian Government, the disc alone, h is 12 in. thick, weighs 2¼ tons; and the 100-in. which is being made for the Mt. Wilson Observatory, weighs in the neighborhood of 4 tons. If this weight could be reduced in the porcelain disc, and the material be made homogeneous, with a coefficient less than glass, it would be a great advantage to the instrument itself.

THE AUTHOR. In regard to internal stresses, mentioned by Mr. Lynch, it was first thought that the failure of the round rods at the head was due to this cause. Accordingly the rectangular pieces were made with a gentle change of slope, and considerable care was used to so handle the clay as to leave all parts of the piece in the same condition. The nature of the failure of these pieces seemed to indicate, not internal stress, but the combination of the direct tension with the tension resulting from the transverse compression of the grips. As suggested by Mr. Farmer, a grip with larger bearing surface should make it possible to develop the full tensile strength. The head of the test piece should enlarge somewhat more rapidly than shown in Fig. 2.

Transverse compression not only weakens a rod subjected to longitudinal tension but also strengthens it for longitudinal compression. Bands around porcelain rods near the ends seem to have the same effect as in hooped concrete columns.

Internal stress is probably an important factor in insulators where there are rather abrupt changes of section. In metals the amount of internal stress may be determined by measuring the dimensions of a small portion, then cutting it free from the rest of the body and measuring again. The difficulty of cutting makes this impracticable with porcelain. A form of test bar with an enlargement or contraction near the middle will enable one to separate the effects of compression of the grips from those of internal stress and unequal distribution of stress. To differentiate completely between these last two factors may not be possible, but series of tests varying one at a time the two elements of heat treatment and form of piece should give valuable results. Whatever internal stress is found may be due either to the heat treatment or the mechanical treatment of the wet clay, and another set of experiments will be necessary to settle all these questions. For testing these pieces, bending tests with a constant moment for a considerable length will be found easier to apply than direct tension tests.

No. 1508

FOUNDATIONS

By Chas. T. Main, Boston, Mass.

Member of the Society

fter the location of a plant has been decided upon, and the elected, a sufficient number of borings should be made or test sunk to determine the character of the soil and its bearing ity. Such exploration will reveal whether the site is suitable hether the cost of foundations would be excessive. If the site und suitable, the knowledge of the underlying soil or rock is sary for the proper and economical design of the foundations.

The borings are made for the purpose of determining at what firm soil is to be reached, if at all; the thickness of any stratum m soil; the character of the underlying material; the level of round water; whether piles will be required and the probable h of the same.

It is of great importance to support all structures on a stratum il below silt or peat. If the structure is to be a heavy or an rtant one and it is found necessary to use piles, some of the gs should be carried to bed rock, if possible, and dry samples e soil should be taken every few feet in depth. The samples d be examined as soon as taken, as the moisture in them evapo- and their character changes rapidly. If uniform conditions ound a few widely scattered borings will be sufficient, but e the conditions vary a greater number should be made. A iption of the processes and values of borings is given later.

If it is found necessary to drive piles, test piles should be n and careful records kept. Piles should be tested by loading marked settlement takes place and careful readings should be before and after each increment of load. If possible, loads d be allowed to rest at least 24 hr. after each increment, t the final load which should remain on at least 48 hr. unless

esented at the Annual Meeting, December 1915, of The American ty of Mechanical Engineers.

a failure of pile or testing platform prevents. All test piles should be pulled, whether load-tested or not, to determine their condition and suitability for the work.

5 From the accumulated data obtained from borings, test pits, test piles and from pile loading tests, it will be possible to select working loads for the piles suited to the building to be supported. In general the working pile values should have a factor of safety of not less than 2½ based on a load producing ⅜ in. total settlement by test. However, it may be desirable to select a working load based on allowable settlements such as $\frac{1}{16}$ in. to ⅛ in. The values of the factor of safety and working or ultimate settlement are all to be fixed to suit the class of structure to be supported.

6 Buildings which are to contain moving machinery or delicate instruments would naturally require piles with fairly large factors of safety, while in cheap one-story structures for storage purposes the safety factors could be much lower. Where piles are not load-tested, the values given by the Engineering News formulæ can safely be used. These formulæ are:

For a pile driven with a drop hammer

$$P=\frac{2\,W\,H}{S+1}$$

For a pile driven with a steam hammer

$$P=\frac{2\,W\,H}{S+0.1}$$

in which

P = safe load in lb.
W = weight of hammer in lb.
H = fall of hammer in ft.
S = penetration or sinking in in. under the last blow

7 Test pits should be sunk for determining the level of the ground water and for making a study of the soil for a reasonable depth more accurately than can be done with borings. Both the maximum and minimum levels of the ground water should be determined, the maximum for obtaining the hydrostatic head on waterproof basements and the minimum for finding the safe level for cutting off wood piles or to determine if it will be better or cheaper to use concrete piles.

8 When the structures are of any considerable magnitude and piling is unnecessary, tests should be made to determine the bearing

ıcity of the soil to insure the maximum economy in design. ing ascertained the condition of the soil and decided that piles not necessary, and having also decided upon the maximum sure to which the soil is to be subjected, the widths of founda-s can be determined from the estimated loads.

Solid ledge forms the securest support for foundations, pro-ng it goes all round the building. Part earth and part ledge are to cause unequal settlements. The ledge, if uneven, should be led off and, if on a slope, cut to form steps to give an approxi-ely horizontal surface. In going from rock to earth, the footing ses of the foundation should be spread out to a greater width on soil next to the ledge, gradually narrowing into the regular width earth foundations. If there should be any unequal settlement, ill be spread over a greater length of the superstructure and ably save cracks in the walls which might otherwise occur at junction of earth and rock.

0 Hard gravel or hard pan is quite as desirable as ledge. vel and sand are also good when kept dry. A stratum of 6 to 8 f hard-compacted and well-cemented material, even if underlaid softer material, is usually safe. With dry sand, this stratum ld be double the thickness.

1 Clayey soils are somewhat treacherous. Upon exposure ie air they dry and crack, and exposed to rain they become semi-or expand. With this soil it is best to open only a small portion ie trench at a time and quickly fill in behind.

2 With buildings used for industrial purposes, there is usually e or less vibration caused by machinery in motion, and the s carried by the foundations should be less per square foot of ing material, or less per pile, than in buildings which are not ect to such vibration. On soft clay or running sand confined, pressure should not exceed ½ to 1 ton per sq. ft.; medium blue , whether or not mixed with fine sand, 1 to 2 tons; hard clay, ns; compact sand and gravel, 2 to 3 tons; hard pan, 5 to 6 tons sq. ft. Under favorable conditions of soil and use of building, e loads may be exceeded.

3 In any building with a uniform firmness of earth under it, area of foundations for walls, towers, piers, and other portions ne building should be in proportion to the pressure. The loads he outside walls may be a little lighter than those on the piers. ase the soil should vary considerably in one portion from another, areas supporting equal weights should be changed to correspond

with the soil. In this way unequal settlements are avoided and the most economical structures of approximate uniform strength obtained.

14 Where there is unreliable soil, piling must be resorted to. If firm earth cannot be reached by the bottom of the piles, the supporting force is friction alone. If there should be an underlying firm stratum, even at such a depth that excavation would be too expensive and sometimes impossible, but still within the reach of piles, then piling can be used with good success.

15 Where wood piles are properly driven, that is, not broomed or broken, the bearing capacity of the piles, when driven into sand, usually shows under the test loads a slight increase over that indicated by the Engineering News formula; but those which have brought up hard and have been crippled will show a much less capacity under the test load than was indicated by the formula.

16 From tests recently made on wood piles embedded in medium and stiff clay on the site of the new buildings for the Massachusetts Institute of Technology, it was found that for about each 5 ft. of embedment of the pile in clay, 1 ton value could be added to that given by the Engineering News formula for drop hammers.

17 Spruce or similar soft wood piles without iron points, driven with drop hammer, should be used cautiously where there is a hard crust of gravel or sand overlying clay. Oak piles without iron points will usually penetrate the hard ground and come practically to a standstill without material injury. Any wood pile which is to depend largely upon point bearing for its value should be of oak or southern pine.

18 When piles are driven through a hard fill and an intervening layer of peat, silt or mud, to the hard sub-soil it will be necessary to reduce the value given by the formula as it contains the values of the hard fill above the intervening peat. This value can be approximately corrected by subtracting the value shown when driving in the fill or a conservative value assumed after comparing with other piles passing through no complication of strata.

19 Unless Batter piles are used, it is necessary to drive far enough into the solid material to give stiffness to the structure against vibration, especially where the fill above is soft.

20 Concrete piles should be used where the distance to good bearing material is great, where wood is scarce or the ground water low. Considerable expense may be saved in the foundations under the latter condition.

l The three kinds of material commonly used in foundations ry rubble masonry, rubble masonry laid in mortar and concrete. work is unsuitable for industrial buildings. Stone laid in nt mortar with bedded joints is very satisfactory and should sed where there is an ample supply of stone at a low price. rete is most commonly used and is usually the most convenient rial to handle, and under ordinary conditions it can be used at west cost.

2 A foundation for a chimney and other isolated structure ng a small base and heavy pressure should be carried to the t depth to which it would be necessary to go with any other reasonably near it, since if it is on sand there is a liability of mining it.

3 The preceding is a very general consideration of the subject and the following pages are devoted to more extended de- ions and data.

TESTING SOIL

4 *Wash Borings.* These are made with the aid of a tripod, or steel casing, drill rod, hose, force pump, bucket, etc. The d used to support the casing and drill rod usually stands 12 to high. The casing is usually made of heavy pipe, 2 to 2½ in. nal diameter, and inside it works a heavy hollow drill tube or 1½ in. to 1¾ in. outside diameter. This drill rod is fitted at ottom with a chopping bit having openings in it for the water while the top is connected with a water hose and force pump, atter usually double-acting. In action, the water is forced through the drill rod, jetting through the holes at the chopping nd carrying up the loosened material in the annular space een the rod and case.

5 The method of forcing the casing and rod down depends on character and density of the material encountered. In soft rial very little effort is required to work the rod down, while rd material more or less lifting and dropping or churning will be red.

6 In order to record changes of strata and take samples of oil, it is necessary to know the level of the bottom of the drill times and to watch the overflow for the color and character of oil. In taking samples the overflow is caught in a bucket and ed to settle. The samples are taken and placed in glass bottles ully marked with the boring number, sample number, char-

acter and thickness of the stratum which the sample represents. To be complete the records should also include the level of the ground water and the elevation of the surface of the ground at the boring referenced to some datum.

27 Wash boring samples do not always represent closely the true character of the soil as the water jet and chopping bit change it radically by breaking it up and separating the fine from the coarse material; the coarse parts are mostly pushed aside while the fine parts are taken to the surface by the water flow. The presence of clay in sand may be easily overlooked, while a hard clay suitable for supporting a structure may be reported as a soft one and unsuitable.

28 More reliable samples may be obtained by withdrawing the drill rod and forcing a pipe into the soft soil, then bringing the pipe to the surface, thus obtaining a dry sample in nearly its natural state. In hard soils, such as very hard clay, soft slate or shale, the pipe for taking samples may have its lower edge sawtoothed, and the most satisfactory samples may be obtained by working this without the water jet.

29 *Test Pits.* Test pits furnish the opportunity of observing the character of soil, its degree of compactness, amount of moisture, etc., but to be of full value they should be carried well below the level of the bottom of the foundations. In cases where the strata change with the depth, a test pit gives no sure indication of the soil below the foundations unless carried deeper than the level of the footings.

30 *Test Rods.* Testing soil with a rod is an unsatisfactory method and cannot be relied upon to give accurate information except in a limited number of cases. In a homogeneous material, not too hard, the method is valuable in determining the constancy or varying density and resistance. In a heterogeneous soil, the sinking of a rod may be stopped by a layer of hard gravel, a rock or log and would not furnish reliable or complete information.

31 Fig. 1 shows the form of record to be kept of borings or test pits.

EXCAVATION

32 Work on foundations consists of excavation of earth or rock, including shoring, sheet piling, or coffer dams, and a structure of stone, concrete, brick or timber at the bottom of the excavation, including bearing piles. In nearly all cases the expense of excavation will increase with the hardness of soil and inconvenience for working;

but if the excavation is in sand or soft earth, with considerable water to contend with, the cost is largely increased by the necessary structures for enclosing the excavations and sustaining the banks.

33 Earth is hard in proportion to the amount of cementing material which it contains and its temporary stability also depends on the amount of this material, while its permanent stability depends upon the friction of the particles on each other. The disadvantage

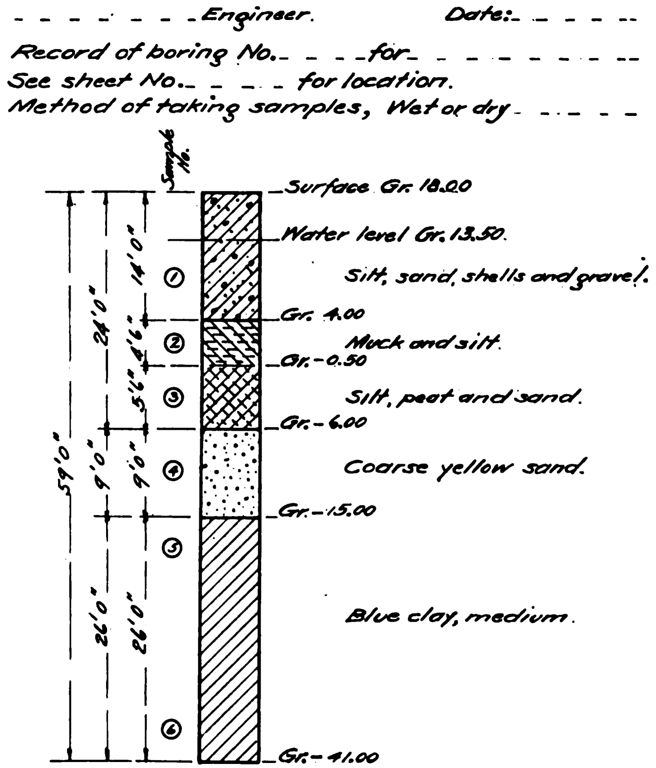


Fig. 1 Form of Record to be Kept of Borings or Test Pits

of hardness for excavation is offset in many cases by the advantage of the self-sustaining power of the vertical cut for a time sufficient for the work on the foundation to be completed. If a vertical cut is exposed for a long time to the weather, it may become dangerous. The effect of an excess of moisture, freezing and thawing, or drying out is to crack off the bank and gradually to approach the natural slope of permanent stability. If the time required to lay the foundation be very long, or the weather unfavorable, it may become necessary to shore up even in firm earths, but in a much less expensive manner than in material which has a tendency to flow.

34 Usually, in working in clean sand or gravel below a depth of 5 to 6 ft., shoring may be done by laying in planks horizontally along the sides of the banks, putting in vertical timbers or planks at short intervals and opposite each other, and bracing between them. Even with this sort of shoring, it is well to make it secure, so that no braces or other pieces may drop out and injure any person or perhaps cause a slide.

35 When sand containing water, or soft clay with running water, is encountered, the saving of soft excavation is entirely absorbed by the expense incurred in sustaining the banks and preventing damage to any adjacent buildings. Sheet piling must be used here, which consists of planks or timbers driven closely together and of a thickness varying from 2-in. plank to large timbers, according to the depth, pressure, and soil. Four-in. plank is about the thickest used in ordinary work. Three to 6-in. plank should be grooved on edges, the grooves to be filled with splines.

36 The usual way for driving sheet piling is to lay out the line for the sheet and on it drive some guide piles, the excavation being carried down as far as possible before commencing to drive. To the guide piles, inside and outside, are bolted or spiked stringers or rangers, which serve as guides for the piles at the top; a second ranger on the inside serves as a guide while the soil presses the pile from the outside. The bottoms of the piles are tapered off, and also cut slanting, so that there will be a tendency to crowd against the next one already driven. To prevent splitting while driving, the tops should be protected with an iron ring, the piles are driven far enough below the grade of the bottom of foundation so that no outward pressure can break them below the bottom ranger. As the excavation is carried down, these rangers are put in at a distance of 5 or 6 ft. apart and the bracing from side to side done on them. Although this is only a temporary structure, all the piles should be new and sound, as they are subjected to severe strain while being driven. New piles can be used again elsewhere if extracted, while shaky timber would cause trouble and perhaps cripple the sheeting, besides being useless after extraction.

37 Where any large amount of sheet piling is done, the steam hammer is the best driver to use, or if this type is not available, the ordinary pile driver with falling weight can be used to advantage.

hammer should be lighter than that for driving bearing piles. ›re the work is not very large, the piles may be driven with ›y beetles or mauls, but this method is extremely slow for thick ting. When driving by steam, a good many light blows of the mer is better than a few heavy ones and the practice is not so apt ripple the planks.

38 In commencing construction of a building, it is customary ut the trench around for the outside walls, leaving the earth ›h comes inside the building for the support of the sheeting until walls are built and set enough to receive the outside pressure.

39 *Coffer dams* are built for the exclusion of water while work eing done. The kind employed depends upon the nature and nt of the work and the strength should be somewhat in proportion to the amount of damage or delay from failure. As the space amount of puddling material are usually limited, the best and .l form will be a bank of puddle enclosed and supported by a of sheet piling on each side. Experience has shown that 4 to is sufficient for the puddle to exclude the water; but unless the is supported independently, its width must be in proportion ıe depth of water, so that it will not be overturned. Good timber ld be used here as in ordinary sheet piling and for the same ›ns. Where there is room, a bank of sand against the inside ting will assist in supporting the dam.

0 Single-sheeted dams are sometimes used successfully. › are made from planks, tongued and grooved, or carefully ‹ed, but they can only be used successfully where the soil is ›f a flowing nature; otherwise, when the pressure is relieved from nside, the flow will start under the bottoms of the sheeting and er the dam useless. This sort of dam should never be trusted 'e its failure would cause much damage or expensive delay. A ›le dam will, in nearly all cases, pay for itself.

1 *Bag Dams.* Where the depth of water is not great, bag s can be used to advantage. They can be cheaply and quickly tructed and in some cases are almost indispensable. They can sed for shear dams for turning water away from foundations, ›ially where sheet piling cannot be driven, for repairing breaks ınks and for many other purposes. They are made from strong ty cement bags or gunny sacks filled with sand or other suitmaterial and securely tied and deposited in the place where they o be used.

2 *Disposal of materials* of excavation should be made in the

cheapest and quickest manner. If by carts, and the inclination is not too great, a run should be made from the surface to the bottom of the excavation, and the carts backed down and filled. If by wheelbarrows, the run should be the same. If loaded into cars on a side track, the material may, on any sizable job, be hoisted by derricks operated by steam power, in scales, and dumped into the cars. If it is impossible to load carts in the excavation itself, they may be loaded in the same manner as the cars. In excavating the trenches in soft material, the hoisting method must be used. Where the amount of excavation is large, steam shovels can be used to advantage.

43 Material to be used again for backfilling should be put in a convenient place, and backfilling should be begun as soon as possible, to protect the foundation from the weather and for convenience in working.

44 *Piling.* Where the depth of good bottom is too great to be reached economically by the foundations, approximately 10 ft. or more, it becomes necessary to use piles. The determination of the type of piles depends upon local conditions. If it is necessary to spread the load over as much area of the underlying land stratum as possible, wood piles should probably be used. If it is not necessary to spread the load, a fewer number of concrete piles with higher bearing value can be used. If the ground water is comparatively low down, it may be much more economical to use concrete piles and carry the foundations down to the ground water level, than to use wood piles.

45 The kind of wood to use for piles is governed by the kinds which are obtainable at the location under consideration and the character of the soil through which it is to be driven. Soft wood like spruce and white pine can be driven into soft soils safely, but in hard soils there is danger of brooming the points or crippling the pile, and oak, southern pine or some hard wood should be used. As stated above, an exploration of the site should be made by borings in order to design properly the pile work.

46 The driving is done by either a drop hammer or a steam hammer. In sandy soils and soils containing gravel, the driving can be assisted by the use of a water jet. The final blows to test the rate of penetration should be made after the water is shut off. Indication of over-driving is shown by the bouncing of the hammer and by bending and kicking. The length and size of the piles and character of the soil determine the weight and drop of the hammer.

17 In hard soils it is well to protect the tip of the pile with a cast or steel cap, and in all cases where the driving is at all hard it ;cessary to protect the butt with a ring or cap.

18 All wooden piles must be cut off at or below the ground ɜr level. If they are constantly wet they will last practically ver.

19 Complete specifications, definitions and principles of pile ing are those adopted by the American Railway Engineering ciation.

STRUCTURES

;0 Of the structures at the bottom of the excavation there are y forms and kinds. Those most commonly used are concrete, e laid in cement mortar or bedded rock, stone laid with outside ts pointed and then grouted full, and stones laid dry.

;1 Dry rubble work is unfit for industrial buildings. As there constant vibration, the few bearing points of the stone, when dry, will get worn off and thus cause slight settlements, which, ıgh perhaps not unsafe, might lead to annoying cracks.

;2 The cheapness of suitable building stone in some localities sometimes make it desirable to use stone for the foundations. ıe laid in cement mortar, with bedded joints, is the surest kind ;onework. With grouted work there is not that surety of filling he joints. Grout, when made thin enough to run freely, takes ry long time to set and strengthen, and never attains the strength ment mortar. The sand is liable to settle away from the cement, making the final strength variable. Old foundations laid dry be greatly strengthened by pouring in grout, and pinning them The more recent method of forcing grout into old work is with great satisfaction in many cases.

;3 Whichever of the two above described methods is used, the ıod of laying in the stone is the same. All stones should be laid heir natural beds to obtain the greatest strength, the largest stones being used for the footing courses. In order to distribute pressure over several stones and to tie the wall together, they ıld be laid to break joints, having no two adjacent vertical ;s in the same vertical plane. The trench must be prepared ɜceive the footing course, and a bed of mortar should be spread re the stone is finally lowered. Not the slightest rocking motion ıld be allowed. Every stone should be washed clean, and this ıing will moisten the stone so that the moisture will not be absorbed from the mortar. Joints should all be filled, and care taken that they are as small as possible, not more than $\frac{1}{6}$ the whole contents being mortar. In stepping back, the lap should be so small that there is no danger of cracking off the courses below. The step should not exceed $\frac{1}{2}$ the depth of the courses. The outside must be laid up as smooth as the inside if a fairly water-tight wall is expected. If the outside is rough, water will work into the interior, tending to make the entire mass disintegrate.

54 In some cases it may be necessary or desirable to carry the brick wall below grade to the footing, in which case the brick should be selected, hard burned, and very carefully laid in cement mortar.

55 The bricks should be cleaned and wetted before laying and the mortar should be fairly rich in cement and have slightly more water than usual. The bricks should be laid in thick beds of mortar, shoved into place, tamped and all joints filled.

56 Before backfilling, it will be well to apply to the outside of a brick wall below grade a waterproof coating which may vary in character from the application of hot tar to a membranous coating. Generally two coats of hot tar or asphalt are sufficient protection against ground water, which if allowed to get into a wall will probably keep the inside surface moist or wet and discolor it.

57 When putting an asphalt compound on a wall, it will be found desirable to make the first coat of hot tar which has a greater penetrating power and acts as a good bond between the wall and asphalt. The latter is a heavier material and is generally not satisfactory if applied directly to a wall surface.

58 There are now on the market a number of damp-proofing compounds which have more or less merit, but, when possible, all coatings which are intended to permanently keep moisture or water out should be placed on the outside of the wall. Where it is necessary to place the waterproofing on the inside, it will be found desirable to back it up with a lining of sufficient weight and strength to prevent the hydrostatic pressure from forcing it away from the wall surface.

59 These remarks on damp-proofing apply as well to concrete or stone foundations. Concrete, plain and reinforced, is now used almost exclusively for all kinds of foundations owing to its economy, availability, suitability, ease of handling and rapidity of construction and strength.

STABILITY OF FOUNDATION WALLS

60 The question of the stability of mill foundation walls against sures of ground water or a fill seldom troubles the designer, ex- where the building is located on a river, in an embankment or iires a high basement wall below grade. In such cases it is ad- ble to design walls for proper stability during or immediately r construction, before the superstructure is placed, and after superstructure is in place. It is frequently the case that a new rete wall, which has not attained its full strength, has the fill placed against it on one side and must act purely as a re- ing wall. At such times it may have its greatest overturning ient, due to the backfilling having been put in and puddled the aid of water.

i1 A wall which may be stable with the superstructure load 1 it may be unsafe without it. The section of a building wall, ng the backfill in place but no superstructure upon it, should uch that the resultant line of pressure passes through the middle l of the base or very close to it. The design would better be that when the superstructure load is added to the wall the *tant* passes through the base very near its center line, as this lition gives a better distribution of pressures on the soil and eases the possibility of cracks in walls. This must be care- considered in soils having a low supporting power, such as wet sand, silt, loam, soft clay, etc., as these materials will read- themselves under unequal pressures and tend to cause settle- ts.

2 The principle that should not be forgotten, especially for lings used for industrial purposes, is that the axis of pressure ld coincide with the axis of resistance, in order that uneven ements may be avoided with their attendant evils. This iple should not only apply to the cross section of an outside of a mill, but to the longitudinal sections of all walls. Cracks ilding walls are often due to the violation of this principle.

3 Concrete walls are subject to shrinkage as a result of setting drying out, which create stresses of sufficient strength to crack at intervals, unless heavily reinforced. Long stretches of of plain concrete will crack at intervals of 30 to 50 ft., the nce depending upon the area of the cross section, and, in order the cracks may not appear directly under a wall column or d pilaster, it is good practice to locate joints in the foundations

according to a design, thus making sure that the line of resistance remains under the line of pressure.

64 Reinforced spread footings are economical where great spread is required and where the soil is poor, or where placed on piling the heads of which are near the surface.

65 The following are rather brief specifications for concrete, plain and reinforced. For more complete specifications, reference should be made to the Report of the Special Committee on Concrete and Reinforced Concrete of the American Society of Civil Engineers and to text books on concrete.

Cement. All cement should meet the standard specifications for Portland cement of the American Society for Testing Materials, as from time to time revised.

Fine Aggregate. This should consist of sand, crushed stone or gravel screenings, graded from fine to coarse, and passing when dry a screen having holes ¼ in. in diameter. It should be of silicious material, clean, coarse, free from dust, soft materials, vegetable loam, or other deleterious matter; and not more than 6 per cent should pass a sieve having 100 meshes per linear inch.

Tests of Cement and Fine Aggregates. Samples representing each car load of cement and furnished by the builders should be tested and held unused for at least 12 days for approval.

Representative samples of the fine aggregates to be used should be furnished by the builder for testing and mortars composed of one (1) part Portland cement, and three (3) parts fine aggregates, by weight, when made into briquettes should show a tensile strength at least equal to 90 per cent of the strength of 1-3 mortar of the same consistency made with the same cement and "Standard Ottawa" sand. If the sand be of a poorer quality, the proportion of cement in the mortar should be increased to secure the desired strength. If the strength developed by the aggregate in the 1-3 mortar is less than 70 per cent of the strength of the "Standard" sand mortar it should be rejected.

Coarse Aggregate. This should consist of crushed trap rock, granite or other hard rock. It should be that part which is retained on a screen having holes ¼ in. in diameter and passing a screen having holes 1 in. in diameter for reinforced concrete, and passing a screen having holes 2½ in. in diameter for plain concrete, and to contain all sizes from the smallest to the largest permissible particles. It should be clean, hard, durable, and free from all deleterious matter. No flat, elongated or soft material should be permitted and all dust should be excluded.

Gravel without crushing may be used for the coarse aggregate in plain concrete, providing it fulfills the above specifications and is suitable for the place.

Water. All water used in mixing concrete should be free from oil, acid, alkalies, or organic matter. Water should be supplied to the batches in a uniform and accurate manner.

Measuring Aggregates. The measurement of the fine and coarse aggregates should be done separately by loose volume in a uniform and accurate manner which can be approved.

Unit of Measure. The unit of measure should be the cubic foot, and a bag of cement containing 94 lb. net should be considered as having 1 cu. ft.

Proportions. The following mixtures may be varied slightly at the works if it is found that a denser mixture will result:

For reinforced stone concrete use one (1) part cement, two parts (2) sand, four (4) parts stone.

Concrete for reinforced concrete may be made richer.

For plain stone concrete to be used in retaining walls, foundations, etc., use one (1) part cement, two and one-half ($2\frac{1}{2}$) parts sand, and five parts (5) stone.

For large chimney foundations or other large mass work, a rubble concrete is sometimes permissible. This is made by adding stones to the regular concrete mixture; stones to be sound, hard, clean, rough and not less than 50 lb. in weight and placed in the mass no nearer than 8 in. to any surface or 12 in. to each other. All such stones should be thoroughly cleaned and wetted with a hose before placing.

MIXING CONCRETE

66 Thorough and complete mixing should be given all concrete to get maximum density and strength. Machine mixing is preferred in all cases, but a mixer producing uniform consistency should be used and not less than 20 turns given the mixer after all the materials are assembled in it. If hand mixing is permitted, it should be done on a watertight platform and all the ingredients should be turned together at least six times and more if it is necessary to get them homogeneous in appearance and color.

67 Ordinarily the concrete should be wet enough to be of such a consistency as will flow into forms and around all reinforcement when used, but at the same time as can be conveyed from the mixer to the forms without separation of the coarse aggregate from the mortar.

68 Retempering mortar or concrete with water should not be allowed after it has partly set nor should the material be used after it has partly set.

PLACING CONCRETE

69 Before placing concrete, and reinforcement when used, all debris should be removed from the forms and they should be thoroughly wetted, except in freezing weather, or oiled with mineral form oil. After mixing, concrete should be placed as quickly as possible, in small masses and in such a manner as to permit thorough compacting by suitable tools. The concrete should be made compact and dense and all surplus water forced to the surface. Formation of laitance should be avoided if possible; laitance should be removed if formed. Concrete should be especially cut against the forms of

all exposed surfaces to remove all voids and to make it dense. All concrete should be deposited without separation of mortar and aggregates.

70 Concrete should be deposited in a continuous operation for the portion fixed as the day's work. When the placing of concrete is suspended, all necessary grooves for joining future work should be made before the concrete has had time to set. All horizontal and vertical joints should have grooves of approved design and when work is resumed all joints should be roughened, cleaned of foreign material and laitance, thoroughly wetted, and then slushed with a mortar made of 1 part Portland cement and 1 part sand mixed with water to a creamy consistency.

71 *Freezing Weather.* In freezing weather all aggregate should be heated and all frost removed before mixing with the cement and water, and special precautions should be taken to prevent the concrete from freezing while it is setting and hardening.

72 *Placing Concrete Under Water.* If possible, concrete should not be placed until water has been removed but if it is necessary to place it in water it should be done under the special supervision of the engineer in charge. It is essential to maintain still water where concrete is deposited. It is better in such cases to use a small gravel aggregate having a maximum size of 1 in. and mixed very wet.

73 The most satisfactory method of depositing concrete in water is to flow it into place continuously by means of good spouts, keeping the mouth of the spout buried in the concrete so as to seal it at all times, preventing water from mixing with the concrete and permitting a good control of the flow.

FORMS

74 Forms should be substantial, tight and unyielding, so that the concrete will conform to the required dimensions and have a full surface without porous places.

75 *Removal of Forms.* Forms should not be removed until the engineer in charge has examined the condition of the concrete and pronounced it strong and safe. No exact time may be stated for keeping the forms in place as it differs with the location, atmosphere, composition, thickness, work the concrete is to do, etc.

76 Concrete which has not been frozen will generally be hard and strong enough to uncover when it gives a good ring under a blow of the hammer. Frozen concrete should be supported by

ms until it has thawed out and allowed to set up to the proper 'dness.

SURFACE FINISH

77 Foundations very seldom require any special surface treat- nt. Exposed walls may, however, be finished with a simple and xpensive treatment in order that they may avoid being unsightly l may weather better. As soon as the forms can safely be re- ved from the surface to be treated, remove all fins and projections nortar and fill all porous spots to an even surface, then while the crete is still "green" give the whole surface a thorough wetting h a mixture of cement and water of the consistency of cream and ıard rubbing with a carborundum stone. The wetting with ıent and water should not be applied much ahead of the scrubbing t is undesirable to have the surface dry out before the scrubbing es place.

FOUNDATIONS FOR PIERS

78 Foundations for piers which support columns are subject to re shock than foundations for the side walls, and the footings ɔuildings which contain machinery which causes much vibration uld be ample so that there will be no settlement. In some cases footing course is made continuous for the full length of the lding. If built of concrete the location of the joints should be determined; they should be about half way between column ters and not over 20 ft. apart. Where the bays are large, rein- ɛed footings can be used. It is probably safer and more con- ient for construction to design independent piers of liberal area low soil pressure suited to the load and soil.

79 The piers above the footings may be of concrete or brick, ally now of concrete. If of brick they should be good hard ned brick laid in cement mortar and capped with stone or cast ı. The pressure should not exceed 15 tons per sq. ft., and as the ;ht increases the area should be increased for stiffness.

80 With piers or other footings of concrete, if built in steps, the izontal lap should not exceed ½ the thickness of the step.

DISCUSSION

M. M. Upson presented a written discussion in which he stated t, in his experience, on most sites where the problem of founda- ı is serious, it is difficult to use the information from either wash or dry borings unless they disclose a substantial bearing stratum. The reasons for this are many, but in general are due to the inability of the average engineer to determine from a physical examination of a small sample of the soil what safe bearing strength it has, or what friction per square foot of pile surface may safely be imposed upon it. Many engineers of wide experience are reluctant to place heavy structures on a site where rock, hardpan, or stiff clay cannot be reached, and frequently valuable real estate remains undeveloped because engineers have not been able to find a good bearing soil within economical reach of the surface.

The idea of carrying heavy monumental buildings or costly steel structures on the friction of piling has not until recently been accepted as good engineering practice, and there is at the present time a large school as yet unwilling to subscribe to the assumptions on which the principle is based. Piling is usually conceived of as a means of carrying the loads from the surface of the excavation through a soft material to a substantial underlying base. It is therefore frequently a shock to be told that the piles supporting certain structures have their points in a material of no greater density and frequently of less density than the material which surrounds their butts.

The engineering prejudice against all friction pile foundations is more or less justifiable, since little information is available in engineering literature on the allowable friction per square foot of pile, and unless the individual has had a wide experience in driving various kinds of pile in all characters of soil, it is quite impossible to safely predetermine the length or the proper pile for the work.

A large percentage of piling work is done in ground which does not have any substantial lower stratum to which the point of the pile can reach. This means that the pile must support its load by friction. The amount of friction per square foot of the surface of the pile is dependent upon two variables, namely, the taper of the pile and the character of the soil surrounding it. Given the same soil, the surface friction increases as the taper increases. This is due to two causes.

a The greater the taper, the greater the tendency to compress and increase the density of the surrounding earth, since a tapered pile subjects the surface of the ground to the maximum initial pressure and thereby overcomes the tendency of the earth to heave around the pile. The density of the underlying material is thus greatly increased. It is obvious that a heaving of the soil releases the interior compression, which in turn reduces the surface friction.

b A unit area on the surface of a tapered pile naturally has a greater vertical component than the same unit on the surface of a straight pile. In the latter instance the vertical strength can be no more than the shear of the earth, while in the former the compression strength of the earth becomes a material factor.

The above statements are based on the assumption that the type of pile used conforms to the essentials of a frictional piling member, and may be tersely as follows:

(1) The form which acts as a penetrating element or the pile itself must not be so cushioned that the major part of the energy of the stroke of the driving hammer is absorbed in the cushion, and must be of sufficient strength to withstand severe abuse.

(2) The compression of the earth secured by driving the form must be maintained while the latter is being removed and concrete put in place. In other words, the substitution of soft plastic and unset cement in a hole made by the driving form will not hold the compression of the soil which has been obtained in the driving.

From the above it may be observed that a lineal foot of piling may represent a supporting power of ½ ton or 3 tons, all depending upon the resistances to which the pile is driven, and on whether the conditions which obtained while the pile is being driven are continued throughout the entire process of placing the pile.

Much thought and money has been expended in experimenting on a simple and inexpensive method to quickly predetermine the probable length and carrying capacity of piles in a given soil. The most feasible method thus far developed is the driving of a series of ¾-in. rods joined together by large cast iron screw couplings. This driving is accomplished by the use of a heavy sledge of given weight and swung through a definitely determined arc. A record is carefully kept of the number of blows required for the penetration of every foot, and the test is considered complete when the resistance of the rods has attained a certain number of blows to the last inch. The extending shoulders of the couplings are analagous to the taper of the pile and test the resistance of the various strata of soil penetrated. By carefully comparing these tests with the actual results of driving piles on the same site, a very interesting analogy is formed, making it possible to predetermine with a fair degree of accuracy the length and the loading.

Charles H. Bigelow said that a good many times sufficient attention is not given to foundations, and he had seen quite a number of buildings under which there had been practically no foundations at all. The reason is probably that foundation work is covered up out of sight and low first cost desired.

He recalled a case in which a building was put up directly over a 24-in. water pipe, where the owner desired a cheap building but there was constant liability of the pipe leaking and washing out the entire structure.

He thought the matter of foundations required a good deal of attention and care, and that the paper would be of service to anyone desiring to take up the facts of the subject on any occasion.

A. G. Monks (written). It may be worth while to point to the interesting complication that arises when buildings are constructed on land so valuable that the outside walls or columns must be placed on the lot line and the foundations must be kept entirely within the lot, a condition frequently met with in cities. It becomes necessary then to place a column on the edge of a foundation, the effect of which is similar to that of a heavy-weight swimmer standing on the edge of a raft. Two courses are open to the designer. He may either construct the foundation large enough and strong enough to carry the column, notwithstanding its eccentric position, or he may combine the foundation with one or more others in such a way that the center of the combined column loads coincides with the center of uniformly distributed pressure on the soil under the combined foundations.

Either of these types is, of course, much more expensive than the ordinary concentric foundation, and they are for that reason to be avoided when possible. The first method was more prevalent some years ago when heavy stone masonry was chiefly used for foundations, and still meets some favor when the soil is particularly firm or the loads light. Concrete reinforced with steel bars, which has come so extensively into use in recent years, both for foundations and for the superstructure of buildings, is particularly adapted

the combination type of foundation, and this type has conse-
ently grown in favor along with concrete.

The design of a combination foundation is often intricate, par-
ularly at the corner of a building adjacent to two sides, both of
ich are on the lot lines. Four columns have then to be carried
one foundation. In some cities the space below the surface of
street is regarded so valuable for present or future subway pur-
es that by the building law foundations are not allowed to pro-
t beyond the street line, and the combination foundation is more
quently required for that reason.

The facility with which reinforced concrete can be handled by
experienced designer, and its adaptability to a wide variation
conditions in the design of foundations, have made possible the
y solution of many problems formerly exceedingly expensive and
ricate.

SANFORD E. THOMPSON (written). Almost as important as the
ring power of the soil is the structural part of the foundations.
ere the construction rests on columns, a thorough study of the
ditions must be made to determine whether single or combined
tings are best. In combined footings, that is, footings upon
ich more than one column rests, the problem of obtaining uniform
tribution of the column loads on soil or piles is frequently so
plicated, especially where column loads are unequal, as to call
the application of principles which the engineer seldom meets
where. The arrangement of steel and the computation of
esses also present many difficulties.

In general, where piles or unstable soil require a large distribu-
n of area the combined footing is preferable. This also applies
general to a line of columns under the conditions indicated. The
ting of course must be designed for the upward reaction between
umns. In a floor the distributed loads are resisted by concen-
ted reactions of the columns. In footings the concentrated loads
the columns are resisted by the distributed reactions of the soil.
simplifies the problem for the designer who is inexperienced in
s type of work to think of the footings as upside down and to
sider the supports to be the columns and the loads to be the
vard pressure of the soil or the piles. The reinforcement of the
bined footing must be placed in just the reverse position from
t in the ordinary beam or slab. It must be in the top of the

footing between the columns and at the bottom of the footing below the columns. In simple cases the computations then are similar to those required for floor design. Unequal column loads and especially the arrangement of three unequal columns on the same footing may further complicate the problem.

A retaining wall, as indicated by Mr. Main, is frequently a part of the foundation. A reinforced concrete retaining wall must be designed to resist not simply the stresses due to the pressure of earth, but also the stresses due to shrinkage in concrete and changes in temperature must be taken into account. This is particularly necessary where the foundation wall projects above ground and the building is placed directly upon it. Of course, if a structure permits, definite joints may be placed between different sections. In building construction, however, it is frequently necessary to avoid joints or any noticeable cracks. By placing reinforcement to the amount of 0.3 per cent, or better still 0.4 per cent of the cross section of the wall, it may be built monolithic and noticeable cracking may be avoided even in long walls.

The greatest difficulty is in providing for a bond between sections of the wall laid on different days. Positive adhesion cannot always be assured, but by running full or better still excess reinforcement through the end forms and then spreading a bonding layer of neat cement upon the vertical surface of the old concrete, just before the fresh concrete is placed it is usually possible to prevent separation.

Of special importance in the design is the reinforcement above and below openings in a long retaining wall. As much, or preferably a little more, cross sectional area of longitudinal steel must be used in the sections above and below the opening as in the total section of the blank wall. The forces caused by shrinkage and temperature, which are proportional to the cross-sectional area of the solid wall, are just as powerful at the openings as in the solid wall, so that the section is weaker unless as much steel (which means a larger percentage) is used there as elsewhere. In fact an excess in quantity may be needed because the area, and therefore the strength, of the concrete itself is reduced by the opening in the wall. In addition it is advisable to put diagonal bars across the corners of openings to assist in preventing the starting of any cracks at these points.

F. A. WALDRON (written). The use of the Engineering News mula is dangerous unless full knowledge of the character and tion of the underlying strata is at hand. Buildings erected on e foundations in peat or silt have been known to go down two or ee feet; in fact, piles have sunk away from the building from ir own weight and weight of soil, when driven to the require- nts of this formula.

Special care should be taken to avoid driving taper concrete es in peat formation, as it is like driving a taper nail into a bber heel and demands too much punishment of the mandrel. ese piles will settle if given double the amount required by the gineering News formula, unless driven to hard pan, which is xt to impossible in thick strata of peat.

Piles driven in peat or silt or other formation of varying thick- ss, where the mass is liable to change, by the fill sliding and rting a force sidewise, should be tied together, where driven gly on each side of the building, or the groups so arranged or signed to resist the unbalanced lateral pressures.

Particular care should be taken, where soil is filled, in analyzing d counteracting forces that tend to spread or force the piles out position by lateral pressure, due to weight of fill on strata above rd pan.

No. 1509

OIL ENGINE VAPORIZER PROPORTIONS

By Louis Illmer, Reading, Pa.

Member of the Society

This paper is a synopsis of a research study to determine the proper proportions of hot bulbs for oil engines. The study was begun by the author a number of years ago when he was called upon to fix vaporizer dimensions suitable for Hornsby-Akroyd oil engines of large capacity. At that time the 32 b.h.p. engine of this type, cylinder 16 in. by 20 in., represented the largest size that could be made to operate successfully, and, while several engines of 50 b.h.p. cylinder capacity had been previously built, their manufacture was abandoned owing to lack of knowledge of suitable vaporizer proportions. A careful investigation of the underlying principles brought out finally the design data here presented and from these data hot-bulb oil engines in sizes up to 100 b.h.p. have been built and operated with satisfactory results. Recently this investigation has again been taken up and has been largely extended so as to include proportions for high compression oil engine vaporizers.

2 In the hot bulb oil engine, of which the well known Hornsby-Akroyd engine is the prototype, the vaporizer heated by the gases of combustion provides a hot surface for the double purpose of evaporating the heavy mineral oils in the fuel oil and of maintaining the confined mixture charge at a temperature high enough to enable self-ignition to be induced with moderate compression. While such self-ignition may be due, in part, to direct contact of the explosive mixture with the walls of the hot bulb, it will be shown that the principal function of the vaporizer is to confine a certain volume of hot gases from stroke to stroke, so that when this volume becomes mixed with the explosive charge, the resulting average temperature will be raised to a point which will cause self-ignition to take place at or near the end of the compression stroke. In starting the engine, the vaporizer is preheated from an external source.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

3 Fig. 1 is a sectional view of the vaporizer generally applied to a single-acting 4-stroke cycle Hornsby-Akroyd stationary oil engine. The vaporizer is of cast iron and forms a chamber, the outermost end or cap of which is unjacketed and maintained at approximately incipient cherry-red heat by the absorption of heat from the working charge. The remaining portion of the vaporizer, which is jacketed, communicates with the engine cylinder by a contracted passage or neck. The air is fed directly into the cylinder through a side port which is in connection with a suitable valve box. The fuel oil is injected into the vaporizer during the suction stroke by means of an oil pump. The spray nozzle is arranged to direct the oil against the hot surface of the vaporizer cap, as may be seen in Fig. 1. The hot gases confined within the vaporizer from the

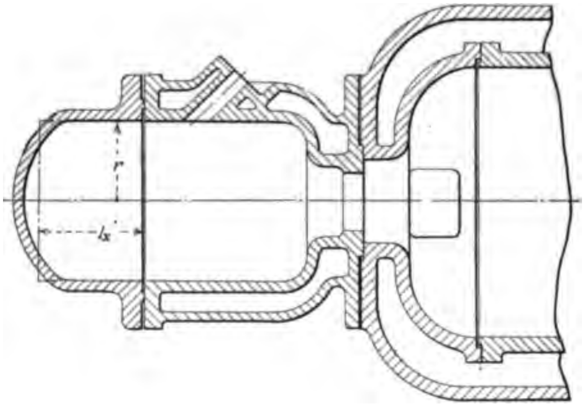


FIG. 1 VAPORIZER FOR S. A. 4-CYCLE OIL ENGINE

previous charge evaporate the highly atomized particles of the injected oil, while the heavier particles impinge against the hot cap wall. Part of the heat required for evaporation of the oil is therefore supplied from the gases in the vaporizer and part from the vaporizer cap; the latter is the major part.

4 The essential principles of operation of a low compression 4-stroke Hornsby-Akroyd oil engine are: The oil is sprayed into the vaporizer chamber in the same period that a volume of air is sucked into the power cylinder. At the end of the suction stroke, the superheated oil vapor or gas in the vaporizer will have been mixed with approximately half of the hot products of combustion, while the body of air in the cylinder is confined there by the vaporizer neck. During the compression stroke, a certain portion of this

is forced through the neck at relatively high velocity, and is ojected into the center of the highly heated oil-gas. This results first in a supersaturated mixture of oil-gas and air being formed thin the cap enclosure, the mixture being kept at a temperature nsiderably above ignition point by the heating effect of the procts of combustion. Self-ignition will occur, therefore, at the tant the further addition of air makes this mixture of such prortions as to be explosive. Owing to this rather complicated action, vaporizer dimensions must be exactly suited to the requirements, the engine will not operate satisfactorily.

5 Table 1 presents some design data deduced from a fully de-oped line of horizontal 4-stroke Hornsby-Akroyd oil engines. om the proportional analysis included, it was found that the one nple relation suitable for design purposes was that of vaporizer lume to piston displacement. The other design factors depend on more involved relations and center about the average temrature attained by the unjacketed cap portion of the vaporizer ll. As will be shown, this temperature depends principally upon proper adjustment of contour length to thickness of the cap ll.

6 The temperature relations existing in a hot vaporizer cap are st treated on the basis of a thin flat disc receiving a uniform heat out over one of its faces. Under conditions of equilibrium, the ulting heat flow will cause a temperature drop between the ter and edge of the disc; this drop may be quantitatively deter-ned without the use of complicated mathematics:

t θ_1 = temperature of disc at center, deg. fahr.

θ_0 = temperature of disc at edge, deg. fahr.

$\theta = \theta_1 - \theta_0$ = temperature drop from center to edge of disc

r = radius of disc, in.

s = uniform thickness of disc, in.

$\dot{H}$ = uniformly distributed gross heat input in B.t.u. per hr. per sq. in. of area

k = specific thermal conductivity in B.t.u. per hr. per sq. in. of section at 1 deg. fahr. head

= about 3.82 for cast-iron.

en, since the temperature gradient, i.e., drop per inch of length, proportional to the rate of heat flow, the gradient in the case of a st-iron vaporizer cap will be equal to 1 deg. fahr. for each 3.82 B.t.u. w per hr. per sq. in. of section. Applying this to a disc of radius r, heat input will be measured by the area enclosed, multiplied by

TABLE 1 VAPORIZER PROPORTIONS FOR HORNSBY-AKROYD OIL ENGINES
(STATIONARY HORIZONTAL S. A. FOUR-STROKE CYCLE TYPE)

Column	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
				Vaporizer Vol. Incl. Neck V_V		Ratio V_V to		Cap Volume		Cap Dimensions—In.						
Size of Engine In.	Rated b.h.p.	R.p.m. N	Capacity Constant = Displ. / b.h.p. $\times N$	Total Cu. In.	Per b.h.p. Cu.In.	Clearance V_C	Piston Displ.	Total Cu. In.	Per b.h.p. Cu. In.	Inside Dia. $2r$	Cyl. Depth l_X	Equiv. Tube Length l_e	Inside Cap Surface A	Full Load Oil $\div A$ lb. sq. in.	Dissipated heat h B.t.u. sq. in.	Av. Cap Temp. t_C by Eq. (17) deg. fahr.
7 x12	5	260	24,000	164	33	0.80	0.36	78	15.7	4⅝	4.65	5.81	84	0.048	24	1270
9½x12*	8	260	27,600	(213)	(27)	(0.61)	(0.25)	(83)	(10.4)	5¾	(3.2) 4.75	6.19	102	0.063	28	1280
11 x15	13	225	24,600	422	33	0.71	0.30	191	14.5	6½	5.75	7.38	150	0.069	30	1300
12 x16	16	225	25,400	515	32	0.69	0.28	216	13.5	7¼	5.23	7.04	160	0.080	33	1270
13½x16	20	220	25,200	637	32	0.66	0.28	248	12.3	7¾	5.25	7.19	179	0.089	36	1260
14½x17	25	215	24,100	825	33	0.68	0.29	362	14.4	8⅜	6.55	8.65	223	0.090	36	1320
16 x20	32	200	25,100	1199	37	0.68	0.30	425	13.3	9½	6.0	8.38	250	0.102	40	1270
25½x33	100	145	24,500	4567	46	0.68	0.27	1415	14.1	14¾	8.28	11.97	553	0.145	52	1220

* Ribbed cap, values without brackets based on equivalent plain cap surface.

Design Constants:
- Stroke in in. = $6.5 \sqrt{\text{b.h.p./cyl.}}$
- R.p.m. = $350 \div \text{b.h.p.}^{1/6}$
- Piston speed = about $150 \sqrt{s}$
- Full load oil = about 0.8 lb. per b.h.p-hr.
- Ratio of expansion = about 3 ½.
- Cl. in per cent displacement = about 43 per cent.
- Compression = about 45 lb. per sq. in.
- Full load m.e.p. = about 42 lb. per sq. in.
- Mechanical efficiency = 82 per cent.

the uniform rate of input per sq. in. of surface. Assuming all of this heat is to be conducted to the edge of the disc, the rate of heat flow through the peripheral sectional area will be equal to

$$\pi r^2 H \div 2\pi r s = \frac{H r}{2 s}$$

Dividing this equation by the constant k, the corresponding temperature gradient at the edge radius, r, will be equal to

$$\frac{H r}{2 k s}$$

This shows that the temperature drop per unit of length is directly proportional to the disc radius; and since the heat flow at the center of the disc is zero, the average temperature gradient will therefore be ½ that at the full radius, r.

7 The total temperature drop from center to edge of a thin disc will then be equal to the average drop per unit length multiplied by the disc radius, r, that is, for the disc

$$\theta = \theta_1 - \theta_0 = \frac{1}{4} \cdot \frac{H r^2}{k s} \quad \ldots\ldots\ldots\ldots [1]$$

8 Applying the same reasoning to the case of a relatively thin internally heated tube whose inner radius is r and length is l (both in inches) and in which all the heat is conducted away at one end, it will be found that the total temperature drop over the full length is equal to

$$\theta = \theta_1 - \theta_0 = \frac{1}{2} \frac{H l^2}{k s} \quad \ldots\ldots\ldots\ldots [2]$$

The temperature drop over a tube is dependent only upon the length of the tube, and for unit length the drop is twice that for a disc of unit radius.

9 The corresponding multipliers of equation constants for any other plane geometrical figure will be proportional to the ratio of the heated input surface to the peripheral section provided for the heat outgo, multiplied by the length of heat travel.

10 The temperature distribution as determined by either of equations [**1**] or [**2**] is plotted in Fig. 2. The maximum temperature, θ_1, corresponds to that of the center of the disc or of the free hot end of the tube, while the disc edge or attached tube end assumes the temperature θ_0. The latter temperature is dependent upon the means provided for the heat offtake.

11 Numerical values for the counter-temperature, θ_0, vary with the size of the engine and the character of joint between the

cap and jacket part of the vaporizer; they should be taken proportional to the change of heat flow with load. In the case of the Hornsby-Akroyd vaporizer cap, which is bolted to the jacketed part of the vaporizer and sealed by a copper wire gasket, a fair average value for θ_0 is 450 deg. fahr. under full load conditions.

12 If to the counter-temperature, θ_0, be added the average temperature ordinate under the curve, Fig. 2, then the average temperature, t_c, attained by the cap may be taken as approximately equal to

$$t_c = \frac{2}{3}(\theta_1 - \theta_0) + \theta_0 \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

13 The various equations given above apply only to caps

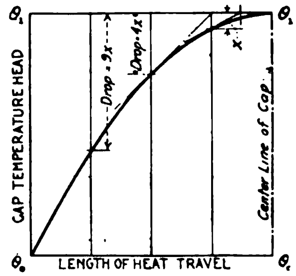


FIG. 2 VAPORIZER CAP TEMPERATURE HEAD

whose thickness, s, is small in proportion to their radius, r. The derivation of formulæ for thick internally heated tubes is somewhat involved, and for usual vaporizer proportions numerical values obtained would differ but little from those derived from the formulæ given.

14 In deducing corresponding formulæ for vaporizers with more complicated outlines, such as relatively thin hemispherical shapes, due allowance must be made for the more involved relation of the average to the maximum temperature gradient. A value of this relation for such shapes may be conveniently determined by dividing the contour length of heat travel into an equal number of parts, finding the temperature gradient for each part and plotting each as an ordinate on a contour length base. For thin hemispherical shapes, this constant is about 0.45 as compared with 0.50 for a disc or tube. Since the input surface of a hemisphere is twice that of a disc of equal radius, and the contour length of heat travel $\pi/2$ as long, the temperature drop will be $0.9 \times 2 \times \pi/2 = 2.8$ that of a disc. Accordingly, the temperature drop from center to edge of a

thin hemispherical shape having an inner radius r is approximately equal to

$$\theta = \theta_1 - \theta_0 = 0.7 \frac{H r^2}{k s} \quad \text{[4]}$$

15 In treating the case of a bumped head, such as is shown in Fig. 1, a simple and fairly reliable method is to resolve the cap volume into an equivalent cylindrical depth, l_x, enclosed by a flat plate end, as indicated by the dotted line in the figure. The corresponding contour length for heat travel, taken in terms of tube length, will then be equal to

$$\frac{r}{2} + l_x = l_e \quad \text{[5]}$$

where l_e = equivalent tube length in in. of the combined contour of tube and disc

16 The input surface in this case will be the area of the disc plus the area of the cylindrical length l_x, which is exactly equal to the interior surface of a cylinder having a length l_e. Hence this factor may be inserted into equation [2] without further surface correction, thus

$$\theta = \theta_1 - \theta_0 = \frac{H l_e^2}{2 k s} \quad \text{[2a]}$$

17 It will be seen that the thickness, s, is an important factor in fixing the temperature assumed by the vaporizer wall. In larger engines this dimension is dependent upon strength considerations, and on account of heavy internal temperature strains the allowable tension stress, as based upon the maximum explosive pressure, should be limited to about 1500 lb. per sq. in. Owing to casting considerations, smaller caps must be made relatively thicker, and in the line of Hornsby-Akroyd engines referred to, the uniform thickness of cap metal is fairly well represented by the relation

$$s = 0.075\, l_e \quad \text{[6]}$$

Where the cap thickness is not uniform, a limited variation may be approximately compensated for by substituting the average thickness of metal as taken over the entire contour length of the cap wall.

18 Internal vaporizer ribs serve to increase the average temperature of the cap, the effect being equivalent to an increase in the tubular length factor, l_x. It is better practice, however, to avoid the use of ribs and to work with plain surfaces.

19 When the cap is projected into the vaporizer and receives heat over a portion of its face in the manner indicated in Fig. 3, the counter-temperature, θ_0, is increased. The derived formulæ apply only to the disc portion of the surface which is directly exposed to the heating action of the hot gases, while the temperature drop over the length, *d*, of the protected tubular section of Fig. 3 should be taken proportional to the amount of heat conducted to the edge of the disc.

20 The case of an offset cylinder head, Fig. 4, receiving heat at a uniform rate of input over its entire surface is best treated by considering the shape as a whole and finding its equivalent tube length having an internal radius r_b. The equivalent tube length to be added to compensate for the flat closing head is given by the

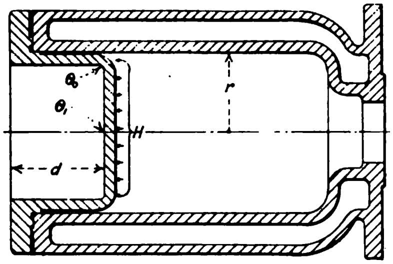


FIG. 3 VAPORIZER CAP PROJECTING INWARDS

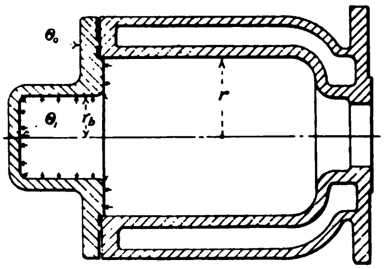


FIG. 4 VAPORIZER CAP PROJECTING OUTWARDS

above formulæ; but in determining the equivalent length to be added for the annulus at the other end of the tube, due allowance must be made for difference of thickness, area and contour length as compared with that of a disc.

21 Before numerical results can be obtained by means of equation [**2a**], a value must be fixed for the heat input factor, *H*. Owing to the cyclic nature of the heat flow, the quantity of heat that will pass into the vaporizer cap wall may be taken as proportional to the average temperature head maintained with respect to the hot cap multiplied by the square root of the rate of such temperature applications. Other factors of lesser importance, having to do with the ratio of surface per unit of volume, etc., will not be taken into account here because of their rather involved and uncertain effect.

22 For the purpose of further simplification, it is assumed that the temperature of the exhaust gases is approximately equal to that of the cap, so that the interchange of heat to and from the cap during all but the power strokes may be regarded as negligible. This holds

true for Hornsby-Akroyd engines, because the temperature of the hot exhaust gases enclosed by the cap is approximately 1300 deg. fahr. at full load.

23 Under the assumed conditions, fairly reliable values for the uniformly distributed heat input factor, H, may be based upon experimental determinations of gas engine jacket losses and may be taken as follows:

24 For 2-stroke cycle engines

$$H = \frac{t_h}{8}\sqrt{\frac{N}{100}} \qquad [7]$$

and for 4-stroke cycle engines

$$H = \frac{t_h}{11}\sqrt{\frac{N}{200}} \qquad [8]$$

where N = engine speed in r.p.m.

t_h = effective temperature head in deg. fahr. as measured by the average temperature maintained during expansion with respect to the heat-absorbing jacket wall

H is in B.t.u. per sq. in. of cap surface per hr.

25 These formulæ hold good only for relatively hot caps, for should the jacket wall be comparatively cold, it would result in a material increase in the heat absorbed due to the temperature head existing with respect to the jacket during the idle or non-power strokes.

26 In case the vaporizer is subjected to external radiation, all of the heat absorbed by the vaporizer cap wall will not be conducted to the edge of the disc, as is assumed in the derivation of the above formulæ The radiation loss of a vaporizer at red heat may be approximately compensated for on the basis of 10 B.t.u. per sq. in. per hr. uniformly distributed over the interior cap surface.

27 Another modifying factor of heat flow is the cooling action due to the evaporation of such portion of the injected fuel oil as may take heat out of the cap wall. This loss is likewise most conveniently allowed for by assuming it to be uniformly distributed over the entire surface of the cap.

28 As previously pointed out, the injection of fuel oil tends to cool the hot products of combustion confined within the vaporizer. The degree to which these hot gases contribute to the evaporation of the fuel-oil depends in part upon the thoroughness of atomization. Without elaborating on this point, it might be said that in the Hornsby-Akroyd engine apparently about $\frac{1}{3}$ of the full load oil is

vaporized by the hot gases and the hot cap imparts the necessary heat of evaporation to the remaining $\frac{2}{3}$ of the full load fuel oil and superheats this oil gas to a temperature of about 900 deg. fahr.

29 The quantity of oil required by an oil engine operating on the explosive cycle can be determined approximately from the following full load indicated efficiency formula:

$$E_i = 1 - \frac{1}{R^{n-1}} \qquad [9]$$

where E_i = indicated efficiency

R = ratio of expansion (about $3\frac{1}{3}$ for the Hornsby-Akroyd engine)

n = exponent in formula $P\ V^n = C$ as based upon the expansion line. For most explosive oil engines of the vaporizer type, n may be taken at about 6/5.

30 The Hornsby-Akroyd oi engine to which the tabulated proportions apply is essentially of the low compression type, because the early injection of the fuel oil does not permit the compression to exceed about 50 lb. per sq. in. gage without causing pre-ignition. For this low limit of compression, the indicated efficiency is only about 21 per cent, making the full load fuel oil consumption at 82 per cent mechanical efficiency equal to about 0.8 lb. per b.h.p-hr.

31 The total heat of evaporation for fuel oil remains fairly constant throughout a wide range of densities, and for the purpose of the present investigation it may be taken as approximately

$$q = 165 + \frac{t_f}{3} \qquad [10]$$

where q = total heat in B.t.u. required to evaporate 1 lb. of fuel oil from 70 deg. fahr.

t_f = deg. fahr. to which the oil-gas is heated.

32 An estimate can now be made of the total cooling action resulting from external radiation plus that due to the gasification of fuel oil by the cap. This heat taken on the basis of unit cap area, will be designated by h, and its numerical value as given in column 15 of Table 1, varies from 10 to 25 per cent of the gross input heat, H.

33 Having thus fixed upon that portion of the input heat that is not conducted to the edge of the disc, a correction for the dissipated heat can be embodied in equation [2a] by taking the tem-

perature drop from end to end of the tube proportional to the net or actual heat flow, thus

$$\theta=\theta_1-\theta_0=\left(\frac{H-h}{k}\right)\frac{l_e^2}{2s}\ldots\ldots\ldots\ldots\ldots\ldots[11]$$

where h = dissipated heat in B.t.u. per hr. per sq. in. of cap area, i.e., difference between the gross input heat and the net quantity actually conducted to the end of the tube.

34 By the terms of equation [8], the factor H of the equation [11] is proportional to the temperature head, t_h, which in turn is approximately equal to the difference of the average temperature during the expansion stroke and that of the cap, t_0. Numerical values for the average expansion temperature may be fixed by the following considerations:

35 The heat consumption of an engine per cubic foot of cylinder displacement can readily be shown to be equal to

$$0.185\frac{P_m}{E_i}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[12]$$

where P_m = mean effective pressure in lb. per sq. in.

36 The maximum attainable mean effective pressure is dependent, in part, upon the amount of fresh air per cubic foot of displacement that is available for fuel combustion. In a 4-stroke cycle explosive oil engine in which the compression pressure crosses the atmospheric line at about 9/10 stroke the full-load mean effective pressure to be expected from a properly proportioned explosive mixture bears a constant relation to the indicated efficiency, and may be taken as approximately equal to

$$P_m=200\ E_i\ \ldots\ldots\ldots\ldots\ldots\ldots\ldots[13]$$

37 Combining [12] and [13], it will be seen that the full load heat consumption of a 4-stroke cycle oil engine is practically constant at 37 B.t.u. per cu. ft. displacement, and that this is independent of the compression pressure. This means that, in terms of fuel oil having a heating value of about 18,500 B.t.u. per lb., a 4-stroke cycle engine may be expected to burn about 0.002 lb. oil per cu. ft. of displacement at full load, a limit that can be readily exceeded when all the air is thoroughly and uniformly mixed with the oil-gas.

38 The theoretical air required for the combustion of oil-gas is about 15 lb. per lb. of oil, and in an oil engine the excess air de-

sired to insure rapid and complete combustion is usually taken at a factor of 3/2 for full load conditions.

39 The maximum temperature of combustion may be approximately estimated on the basis of a full-load heat input of 37 B.t.u. per cu. ft. displacement, as found above. The air sucked into a hot 4-stroke cycle cylinder may be taken at about 0.05 lb. per cu. ft. displacement, and allowing for the high temperatures prevailing in the cylinder after the instant of explosion, the specific heat at constant volume may be assumed at 0.25. The resulting theoretical temperature rise must be reduced to allow for the considerable cooling action of the cap wall; in the case of the Hornsby-Akroyd engine, the actual temperature rise does not exceed ¾ that expected on the unjacketed basis.

40 As will be shown, the full load working charge in the vaporizer of the Hornsby-Akroyd engine reaches a temperature close to 800 deg. fahr. at the end of the compression stroke. During combustion, the temperature rise of the preheated gases within the vaporizer will be further augmented by the supersaturated condition of the explosive mixture. From a study of the rather complex conditions existing at this time, it appears that the full load temperatures occurring in the vaporizer are from 10 to 20 per cent higher than those corresponding to the resultant average pressures of the entire cylinder charge as determined from the indicator card. The full-load temperature rise in the cap of the Hornsby-Akroyd vaporizer, resulting from the combustion of the oil-gas mixture, is probably not far from 2000 deg. fahr. and if this be added to the initial temperature, the maximum temperature of combustion may be fixed at about 2800 deg. fahr. At lighter loads the temperature rise will naturally be reduced in proportion to the oil requirements.

41 The temperature at point of release or end of expansion period is then equal to

$$T_4 = (1 - E_i)\ T_3 \quad \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

where T_4 = temperature of release, absolute

T_3 = temperature of combustion, absolute

and hence the desired net temperature head driving the heat into the cap wall is given by the equation

$$t_h = t_a - t_c = \left(\frac{T_3 + T_4}{2} - 460\right) - t_c \quad \ldots\ldots\ldots\ldots [15]$$

where t_a = average temperature during expansion stroke, deg. fahr.

42 Numerical values for all the required factors having been finally fixed upon, the various equations may now be combined to find a value for the cap temperature, t_c. This must be taken with due allowance for the decreased heat input with rise of temperature in the cap wall. For a 4-stroke cycle engine, this condition is satisfied by substituting in equation [3], thus

$$t_c = \frac{l_e^2}{2\,k\,s}\left[\left(\frac{t_a - t_c}{11}\right)\sqrt{\frac{N}{200}} - h\right]\frac{2}{3} + \theta_0$$

Transposing,

$$t_c = \frac{\left(t_a\sqrt{\frac{N}{200}} - 11h\right)\frac{l_e^2}{33\,k\,s} + \theta_0}{1 + \sqrt{\frac{N}{200}}\cdot\frac{l_e^2}{33\,k\,s}} \quad \text{. . . . . [16]}$$

43 Taking k for cast iron equal to 3.82 and confining the equation to the line of Hornsby-Akroyd oil engines, the above can be further simplified by substituting for the thickness, s, equation [6]. Accordingly at full load,

$$t_c = \frac{0.13\ l_e\left(t_a\sqrt{\frac{N}{200}} - 11h\right) + 450}{1 + 0.13\ l_c\sqrt{\frac{N}{200}}} \quad \text{. . . . . [17]}$$

44 The corresponding derivation based upon equation [7] shows the cap temperature for a 2-stroke cycle engine to be

$$t_c = \frac{\left(t_a\sqrt{\frac{N}{100}} - 8\,h\right)\frac{l_e^2}{24\,k\,s} + \theta_0}{1 + \sqrt{\frac{N}{100}}\cdot\frac{l_e^2}{24\,k\,s}} \quad \text{. . . . . [18]}$$

45 From these formulæ, the expected cap temperature for an oil engine vaporizer may readily be estimated. The equations also hold good for partial loads, provided the numerical values of t_a and θ_0 are suitably modified. A check shows the cap temperatures to vary approximately as the square root of the per cent of full-load oil used.

46 It will be seen that since the cap temperature formulæ contain a speed factor, the cooling off of the cap under slow speed and light loads tends to limit the speed control of the engine.

47 Column 16 of Table 1 shows the full-load cap temperatures for the line of Hornsby-Akroyd oil engines, as determined by equation [17] for rated speeds. The average cap temperature, t_e, is found to be about 1275 deg. fahr., which corresponds to a maximum temperature, θ_1, of approximately 1690 deg. fahr., or cherry red heat.

48 Table 1 further shows that the total vaporizer volume is equal to about 0.3 of the piston displacement, or about $\frac{3}{4}$ the clearance volume of the engine. The jacket portion of the vaporizer allows the relation of contour length to cap thickness to be proportioned independently of the volume content.

49 The air sucked into the hot cylinder of a Hornsby-Akroyd engine may be assumed to have an initial temperature of about 250 deg. fahr. This air, taken at 0.05 lb. per cu. ft. piston displacement, when compressed to about 45 lb. per sq. in. gage, will be confined in a volume just about equal to that of the vaporizer. Since a portion of this air must remain in the clearance space behind the piston, only about ½ of the cylinder intake air will be forced into the vaporizer by the time the piston reaches its inner dead center. This amount of air weighs only about 12 times as much as the injected full-load oil and is insufficient for complete combustion. However, immediately after explosion of this charge, a portion of the ignited supersaturated mixture is sent forth through the contracted neck, when it is made to intermingle with the required amount of surplus air lying in the space behind the receding piston.

50 Oil-gas assumes explosive proportions when the air content of the mixture reaches about $\frac{2}{3}$ of the air required for complete combustion, i.e., about 10 parts of air to one of oil by weight. Such a mixture will ignite of itself at a temperature of about 750 deg. fahr. Assuming that air in proportion of about 10 times the fuel weight to have been forced into the vaporizer at the end of the compression stroke, the hot products of combustion may be expected to raise the temperature of the entire vaporizer content of a Hornsby-Akroyd engine to 750 deg. fahr., ignition temperature when running on $\frac{1}{3}$ to ½ of the oil used at full load. Below this critical point, the temperature head, t_h, is reduced to such an extent that the cap no longer keeps the enclosed gases sufficiently preheated to reach the ignition temperature at the end of the compression period.

51 A corresponding check made for full-load conditions shows that, due to the higher exhaust and cap temperatures, the average mixture temperature in the vaporizer will be raised to about 800 deg. fahr. This increase of temperature causes an advance in the ignition

timing, and at heavy loads may lead to the characteristic pounding that accompanies serious pre-ignition unless counteracted by water injection.

52 As stated, the Hornsby-Akroyd engine is essentially a low compression engine. Notwithstanding its praiseworthy reliability in service, especially under non-fluctuating load conditions, its limit of indicated efficiency and resulting mean effective pressure are so restricted as to make it quite cumbersome in sizes of 50 b.h.p. and upward. From a design standpoint, the large vaporizer proportions are also objectionable. These inherent limitations may be overcome by injecting the fuel-oil near the end of the compression stroke, which allows the use of higher compression and so increases both the efficiency and capacity of the engine.

53 The timed injection of the fuel-oil affords the further advantage that the ignition timing is made independent of the vaporizer proportions. On the other hand, the delayed injection of oil so reduces the available time for forming a thorough explosive mixture that it generally becomes necessary to inject the fuel-oil by means of highly compressed air; in that event, the hot vaporizer gases must be raised to at least 900 deg. fahr. prior to injection, in order to compensate for the resulting cooling action and still have sufficient temperature left for self-ignition.

54 When the temperature of 900 deg. fahr. is reached solely by high compression of the working charge, as in the Diesel engine, this will necessitate carrying the compression to at least 400 lb. per sq. in.; and in practice, the pressure must be raised beyond this to allow for cold starting and other contingencies.

55 When the temperature rise by mechanical compression is insufficient to reach ignition temperature, the difference may be made up by preheating a portion of the working charge by the vaporizer method, as explained. The vaporizer volume required for a high compression engine is relatively small, due to the fact that after a sufficient quantity of fuel-oil has been injected and burned within the vaporizer proper, the resulting expansion of the products of combustion will cause the rest of the working charge to undergo adiabatic compression to a degree that will allow the remaining portion of fuel oil to be injected directly into the air lying outside of the vaporizer, where it may be ignited in the manner of a Diesel engine.

56 The application of this principle to high compression oil

engines requires certain critical proportions for the vaporizer, which may be deduced on the following basis:

If V_0 = total clearance volume of engine (Fig. 5)

V_v = required volume of vaporizer

V_c = clearance volume external to vaporizer

V_v^1 = volume of combustion products of the vaporizer charge, V_v, after pressure equalization

V_c^1 = volume of charge V_c after compression by the expanded combustion products

On the basis of the exponent $n = 5/4$, the critical compression pressure required to reach the ignition temperature of 900 deg. fahr. is

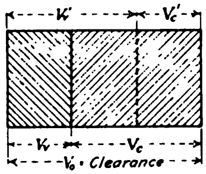


Fig. 5 Vaporizer and Clearance Volumes

about 400 lb. per sq. in. absolute, assuming the initial charge temperature in a hot 4-stroke cycle cylinder to be about 250 deg. fahr. At constant volume, the maximum attainable explosion pressure is directly proportional to the relative temperature increase as expressed by the following

$$P_3 = \frac{T_3}{T_2} . P_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [19]$$

where P_3 = maximum explosion pressure, absolute

P_2 = compression pressure, absolute

T_2 = compression temperature, absolute, at full load as measured in the vaporizer

$T_3 = T_2 + 2000$ = corresponding explosion temperature, absolute.

57 Accordingly, the relation between the volume of the gases in the vaporizer before and after they have been expanded adiabatically to the equalization pressure of 400 lb. absolute, may be fixed thus

$$\frac{V_v^1}{V_v} = \left(\frac{P_3}{400}\right)^{\frac{1}{n}}$$

58 Assuming the vaporizer to be of such proportions that the

adiabatic expansion will just raise the gases in the remaining clearance space to the equalized pressure of 400 lb. absolute, then,

$$\frac{V_c^1}{V_c} = \left(\frac{P_2}{400}\right)^{\frac{1}{n}}$$

hence,

$$V_v\left(\frac{P_3}{400}\right)^{\frac{1}{n}} + \left(\frac{P_2}{400}\right)^{\frac{1}{n}} V_c = V_v + V_c = V_0$$

and, by transposing, the desired relation may be expressed as

$$\frac{V_v}{V_0} = \frac{1-\left(\frac{P_2}{400}\right)^{\frac{1}{n}}}{\left(\frac{P_3}{400}\right)^{\frac{1}{n}}-\left(\frac{P_2}{400}\right)^{\frac{1}{n}}} \qquad [20]$$

59 The minimum allowable vaporizer volumes for high compression engines are given by equation [20]. Instead of using two separate fuel-oil nozzles, the clearance space may be so arranged that a sing e nozz e will serve the space both within and without the vaporizer, as done for instance in the De La Vergne oil engine, type FH.

60 Numerical results derived from equation [20] are shown plotted in Fig. 6, from which it is evident that equalization of pressure cannot occur in the sense assumed unless the compression pressure exceeds 135 lb. per sq. in. gage. Hence engines operating at less than this compression should have their vaporizer volume made equal to the entire clearance space. This requirement may be obviated and the vaporizer volume reduced by about ⅓, when working with a supersaturated mixture and projecting the ignited oil-gas into the surplus air lying outside the vaporizer, as practiced in the Hornsby-Akroyd oil engine.

61 The moderate preheating requirements of the high compression oil engine allow self-ignition to be attained without maintaining the vaporizer cap at full red heat. This reduces internal strains : nd makes the cap better able to withstand fatigue without cracking. Furthermore, the increased ratio of expansion in the high compression engine lowers the exhaust temperature, and in case the vaporizer cap is to be maintained at an average temperature considerably higher than that of the exhaust gases, due allowance must be made in the given formulæ for the cooling action occurring during the idle strokes.

62 The elimination of the idle suction and exhaust stroke make the 2-stroke cycle especially applicable to hot-bulb oil engine it not only improves the frequency of impulse, but also admits c wider speed variation without ignition failure. By carrying th compression in the vaporizer beyond the self-ignition limits, th engine can be proportioned to operate through a much greate speed range than is possible with the low compression Hornsby Akroyd type of engine.

63 Comparing the high compression vaporizer type of oil en

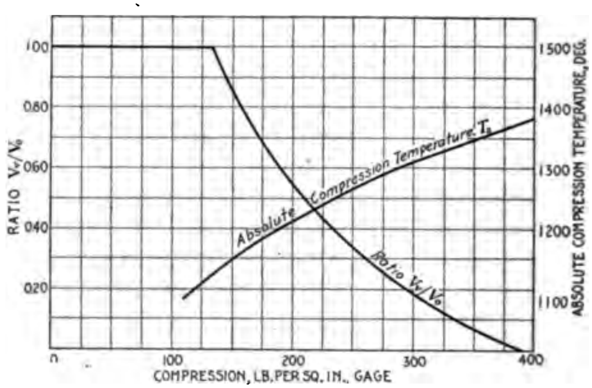


FIG. 6 CURVE GIVING VAPORIZER VOLUME

gine with the Diesel engine, it may be pointed out that the former lower in first cost and the net fuel economy is little, if at all, inferi when working with a compression pressure above 300 lb. per sq. in gage. Among other advantages, the vaporizer engine embodi simplified construction, obviates extremely high pressures, increas mechanical efficiency, lowers the power loss required for air injectic of the fuel oil, lessens operating skill required on the part of th attendant and to a considerable degree reduces the danger o destructive explosion.

DISCUSSION

WM. T. PRICE (written). The author has invaded a very diff cult field and has carried his calculations as far as reasonable gener assumptions will permit.

The subject of vaporizer proportions involves so many facto requiring for practical mathematical treatment so many specif assumptions, that the only safe method to follow is backward calc

›n and interpolation. Exterpolation, even to a moderate degree, ıve found to be not very reliable.

Mr. Illmer assumes for the Hornsby-Akroyd vaporizer a uniform ›ut of heat over the entire inner surface of the cap and a marked ıperature gradient from center to edge. This is not correct. th this engine the spray strikes the cap on the side at an angle, l at this point instead of an input of heat there is a momentary ›traction, as heat is given up to gasify the liquid fuel. At full d a dark area is usually plainly evident; around this region the ıperature is higher, and then the color dies out to a black at the ;e. Mr. Illmer has pointed out that the contact of the cap with water-cooled portion is only over the area of a narrow copper ket; the temperature gradient must therefore be explained as ng due to radiation and convection from the large flange area.

With the De La Vergne type FH engine, using air injection, input of heat to the vaporizer does appear to be uniform over interior surface, but examination of the hot vaporizer shows no ·ceptible temperature gradient. At full load in a dark room there ısually a dark red color of uniform shade from end to end. The ·put of heat seems to be uniform over the whole surface, therefore, flow of heat to the edge being negligible.

There is a great difference in oils and often with Hornsby-royd engines the caps must be changed to suit the different fuels. e writer has known cases where it was necessary to insert a metal l in the vaporizer to maintain the temperature at light loads. ıen the temperature of a Hornsby-Akroyd vaporizer shows symp-ıs of dying, first the water circulation is reduced, then the com-:ssion is increased by inserting plates between crank pin box on ınecting rod, then a ribbed cap is substituted for the plain cap.

The author's comparison of the high compression engine with the ›orizer engine is in the writer's opinion entirely correct, but he ›ears to lean toward a 2-cycle vaporizer engine with a compres-n pressure of about 300 lb. per sq. in. and with air injection.

The addition of the vaporizer to allow of reducing the compres-n from 500 to 300 lb. per sq. in. is only a half-way measure. The ıpression must be reduced still lower and then we must go further l eliminate the air compressor. Already there is in fairly wide nmercial operation a new oil engine which operates without air ection with compression of 150 lb. and with fuel consumption der favorable conditions very slightly below ½ lb. per b.h.p-hr., a consumption of 0.55 lb. at three quarters to full load being a conservative guarantee.

Mr. Illmer has pointed out the advantages of timed injection and has explained the difficulty of forming a thorough explosive mixture in the reduced time interval allowed. The logical solution of the problem is *first,* an abundance of oxygen to increase the combustion rapidity, suggesting at once a 4-cycle design. *Second,* an efficient mechanical spray to divide the oil into the smallest possible particles so as to present the greatest oil surface to the oxygen; this suggests a certain shape of spray and a uniform distribution of oil particles throughout the spray volume. *Third,* a combustion chamber conforming as perfectly as construction considerations will allow to the natural shape of the spray. *Fourth,* as completely as possible confine the entire charge of air in the vaporizer at the time of injection. *Fifth,* a vaporizer cap located so as to complete the vaporization of heavy oil particles and ignite the charge.

The Author. In deriving the formulae presented a series of alternative assumptions was tried out, but the given equations were found to be in best agreement with the Hornsby-Akroyd vaporizer proportions. The primary aim has been to show that rational vaporizer proportions must rest upon something better than a purely empirical basis.

The exception taken by Mr. Price to the assumption made as to the uniform input of heat over the entire inner cap surface of the vaporizer cap appears to be valid inasmuch as the theoretical temperature gradient would not occur in practice. The discrepancy noted resulted from the simplification resorted to in deriving the formulae. A more refined basis for derivation should make allowance for the fact that this temperature head, t_h, is not uniform over the entire cap surface, but is actually smaller at the center than at the edge of the cap.

In addition the oil spray is usually directed against the hot center portion of the cap, which sets up localized cooling and thus further contributes to a reduction of the temperature drop from that expected on the assumed basis of a uniform distribution of input and output heat.

A more complete analysis, taking into account the varying rate of heat input from center to edge of the cap, would no doubt show a temperature distribution more nearly in accord with Mr. Price's observations, but it is thought that such rather involved basis of calculation would not materially alter the average cap temperature t_c as given. This view is based in part upon Professor Hopkinson's[1] series of non-cooled piston temperature determinations, which show a temperature gradient for the head-plate to be in fair agreement with the drop indicated in the present paper.

Experimental measurement along the lines of Hopkinson's investigations on pistons would no doubt offer the most satisfactory method of finding the actual temperature gradient in vaporizers. Such data would readily make it possible to fix upon the degree of cooling due to the fuel injection and show its effect in reducing the ratio of maximum to average cap temperature, besides providing a check for the series of other dependent assumptions which had to be made as to existing temperature relations.

In closing, the heat flow from center to edge of the cap is not due primarily to the increased external radiation into the atmosphere from the flange surface, as indicated by Mr. Price, but is rather to be explained by direct metallic conduction of heat from the cap flange into the jacketed portion of the vaporizer. While the contact surface of the copper gasket is small, the conducting power of such restricted metallic contact may nevertheless be relatively quite large under the existing temperature head, the effect being analogous to the cooling action exerted by the seat upon the head portion of a poppet valve.

As indicated by the quantitative figures given in the paper, the external radiation of heat into the atmosphere can at best account for but a relatively small portion of the total heat input into the cap. Hence while the restricted gasket contact does serve to prevent excessive cooling of the cap edge, it still has sufficient capacity to conduct away the major portion of the heat received by the cap wall.

[1]On Heat-Flow and Temperature-Distribution in the Gas-Engine, Proc. Inst. C. E., Feb. 2, 1909.

No. 1510

STANDARDIZATION OF SAFETY PRINCIPLES

BY CARL M. HANSEN, NEW YORK, N. Y.
Member of the Society

During the last half of the nineteenth century, a complete revolu-on took place in the industries of the world and particularly of the nited States. Efforts of engineers in all lines of production were ncentrated on the one single problem—efficiency.

2 With the dawning of the twentieth century, it was seen that had achieved an efficiency in our industries, surpassing that in other countries. The machine tools of the United States, for stance, were recognized as superior to any manufactured elsewhere. rough the clogging of our courts with litigation involving injuries operators of these machines, however, we began to ask ourselves in our zeal for productive efficiency of machines we had not over-oked one vital factor. Were these machines reasonably safe for ose workers who had to give them constant or intermittent atten-n?

3 With an annual toll of upward of fifty thousand workers killed d approximately two million other more or less serious, but non-al accidents, we realized that we were rapidly on the road to creat- a nation of cripples, and that in order to measure true efficiency our industries it would be necessary to take into account the nomic loss occasioned by all these work accidents. In taking ck to that end we found that an amount of between five and six ndred million dollars had to be added to our cost of production in ler to collate an exhibit of our true efficiency.

4 Some of our larger corporations, particularly the United ates Steel Corporation, The International Harvester Company d several others, at the same time, and even earlier began to study

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Presented at the Annual Meeting, December 1915, of THE AMERICAN CIETY OF MECHANICAL ENGINEERS. Contributed by the Sub-Committee on rotection of Industrial Workers.

lutely prevent something from being done, rather than simply serving as a suggestion. That, in other words, they can be depended upon for unfailing action with as little consideration as possible for

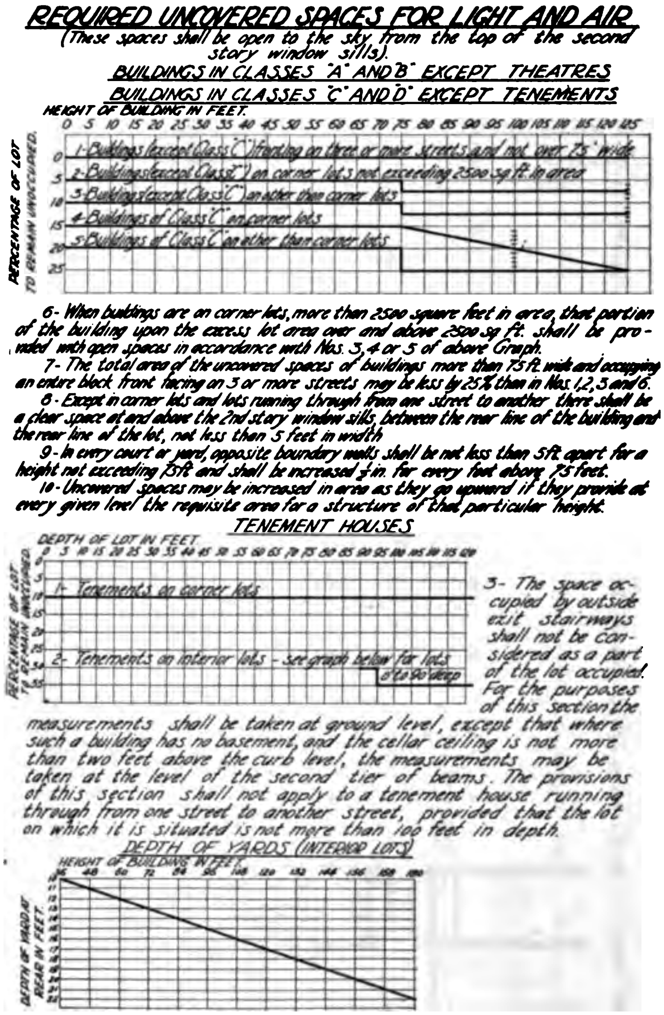

FIG. 1 SAFE AREAS OF BUILDINGS ON CITY LOTS

the universally uncertain human factors, such as knowledge, memory and involuntary acts, etc.

11 In the following an attempt will be made to illustrate these principles in actual application. Commencing with a typical factory

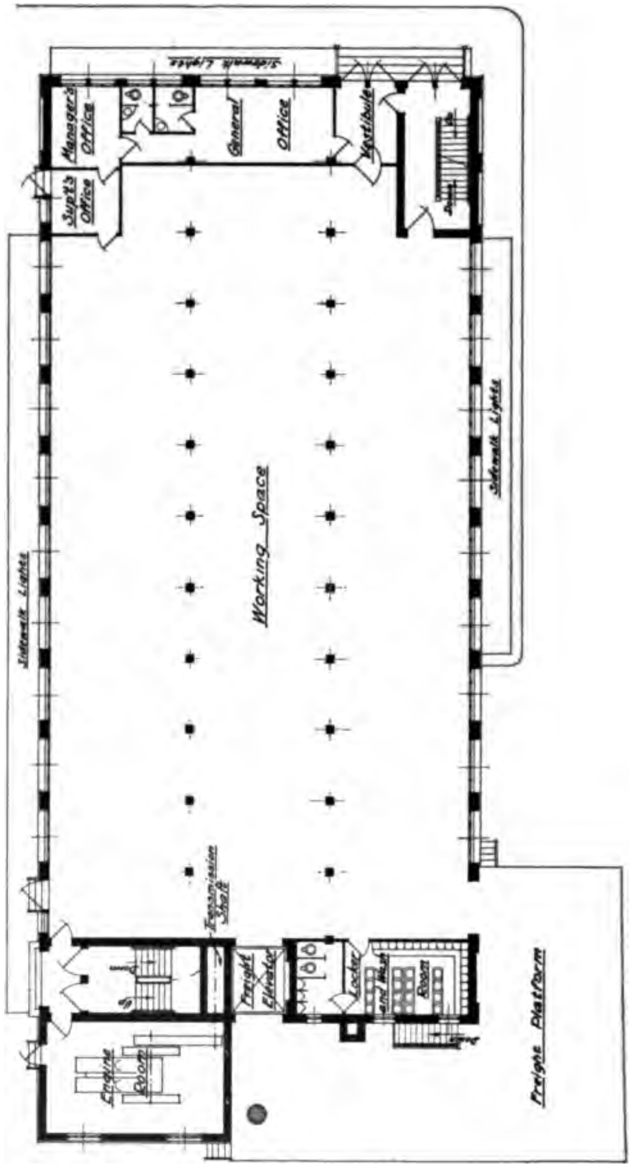

FIG. 2 FACTORY BUILDING MILL CONSTRUCTION. FIRST FLOOR PLAN

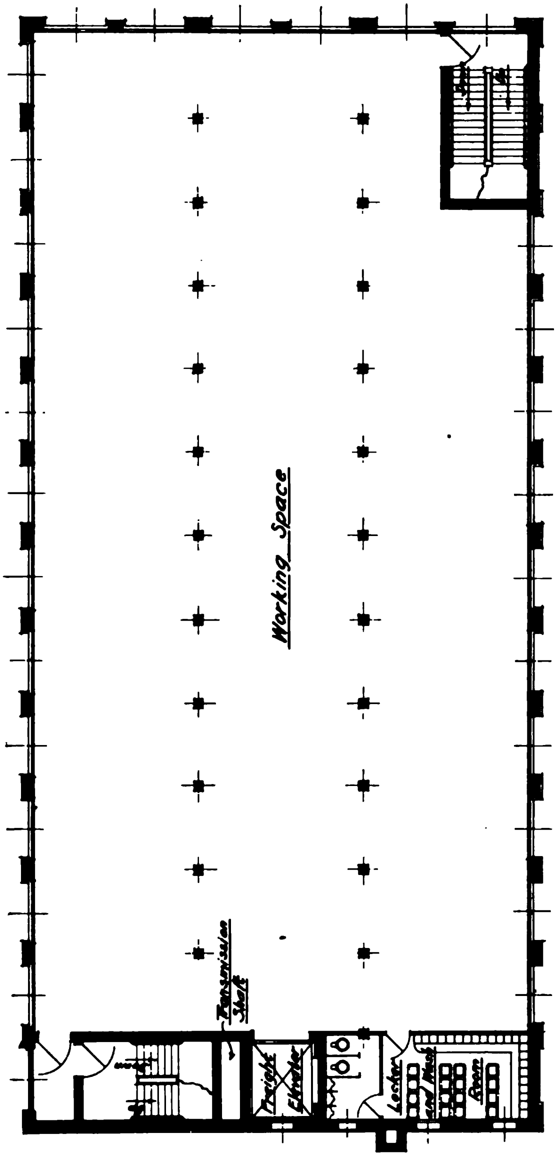

FIG. 3 FACTORY BUILDING MILL CONSTRUCTION. TYPICAL FLOOR PLAN

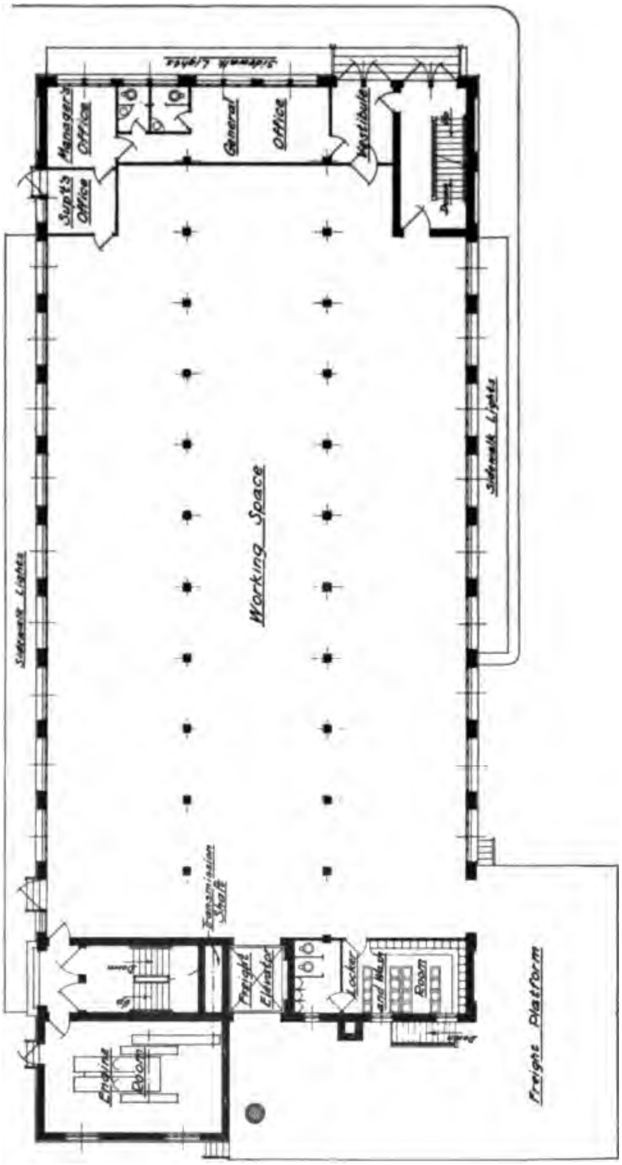

FIG. 2 FACTORY BUILDING MILL CONSTRUCTION. FIRST FLOOR PLAN

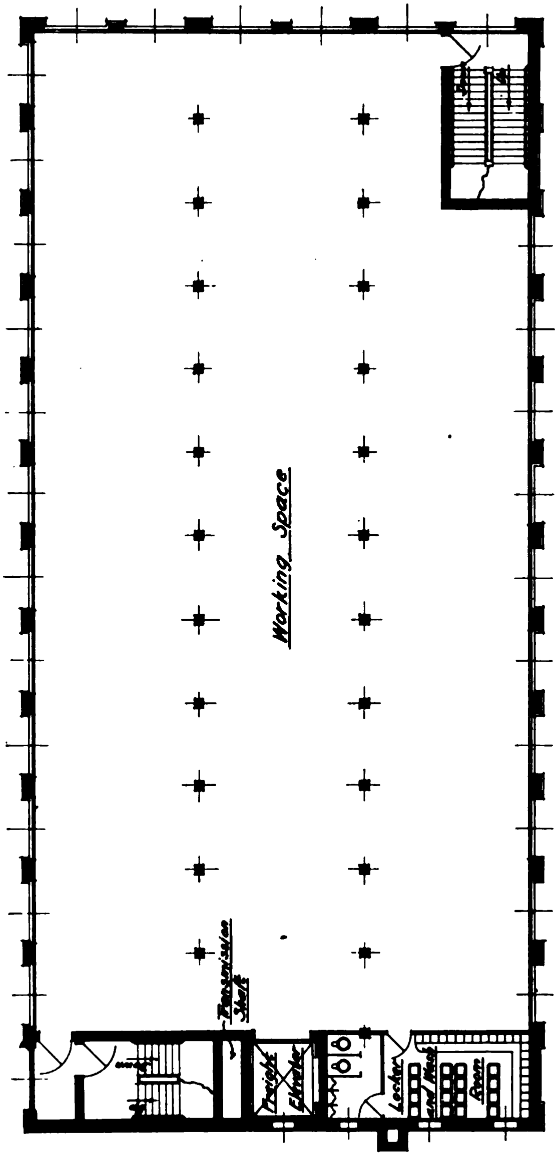

FIG. 3 FACTORY BUILDING MILL CONSTRUCTION. TYPICAL FLOOR PLAN

building, the supposition is that the site has been selected and that the problem is to design and equip the building for maximum safety without interfering with efficiency or convenience.

12 An isolated factory presents few difficulties, but the city factory offers problems in planning, due to the proximity of other buildings, which must be regulated by standards. It has been determined that a certain percentage of the lot must remain uncovered in order to provide sufficient air and light. What that area of unoccupied space should be, given the size of the lot, is indicated in tables in Fig. 1. This illustration shows the percentage of area which can safely be occupied with any class of building on any type of a city lot.

13 Having determined the outline of the plan, the next step is to determine means of rapid and safe egress which at the same time will not interfere with the efficiency of the manufacturing processes and are based on total occupancy of building. Figs. 2, 3 and 4 show suggested arrangements of such a plan by means of stairs enclosed in fire resisting walls and smoke proof towers.

14 As will be noted, the power plant and the vertical transmission shafts are separated from the rest of the building to prevent the spread of fire, and the elevator shafts are inclosed in fire resisting walls and equipped with automatic fire doors.

15 The windows exposed to fire hazard from other buildings are glazed with wire glass set in metal sash. All lintels are far enough below the ceiling so that the heated gases will be contained to operate automatic extinguishers and so that flames breaking out through the window are diverted from the other wall openings.

16 Even allowing for this and for the minimum width of piers, ample lighting facilities are obtained. Space is provided for toilets and wash rooms having outside light and ventilation and with a sufficient number of wash basins and other toilet facilities to accommodate the employes.

17 The construction should be fire resisting to a high degree, as illustrated in Fig. 5, but in case this should prove too expensive, owing to location or other circumstances, a good type of slow-burning construction as indicated in Fig. 6 may safely be adopted. Whether it be fire proof or slow-burning mill construction, fire fighting equipment should be provided, such as automatic sprinklers, wall and yard hose and portable extinguishers, etc.

18 It is admitted that no specific standard can be laid down which will embody specifications covering any or all of the foregoing.

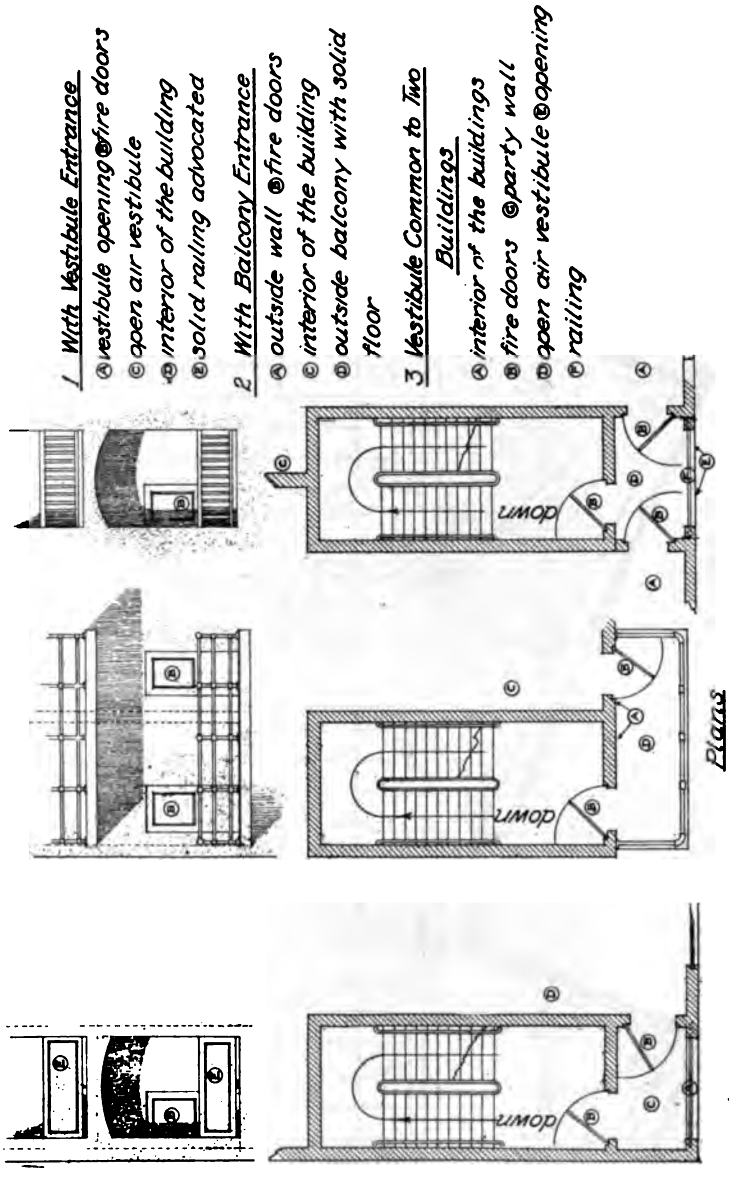

FIG. 4 SMOKE PROOF TOWERS

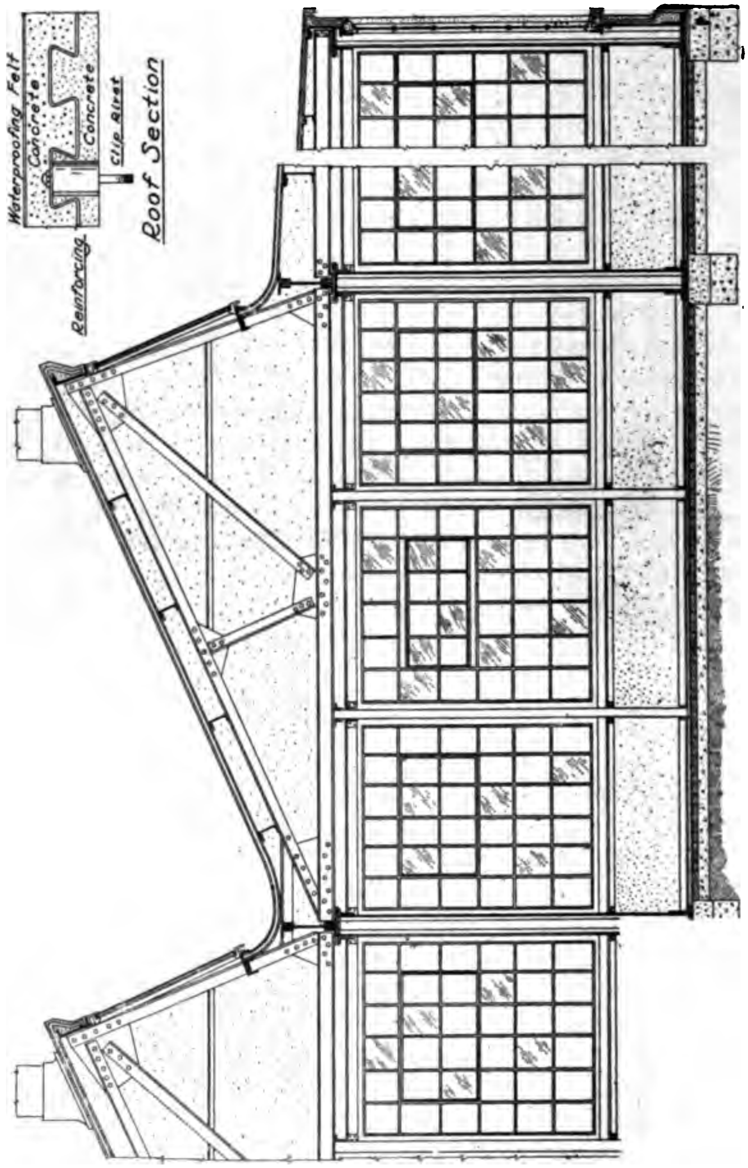

FIG. 5 SAW TOOTH FACTORY CONSTRUCTION

All that the author is endeavoring to show in these illustrations is the principle which must be applied to the particular problem confronting the architect or engineer in designing a building and he desires to emphasize the fundamental fact that these principles are essential.

19 It may be maintained that all these factors are having careful

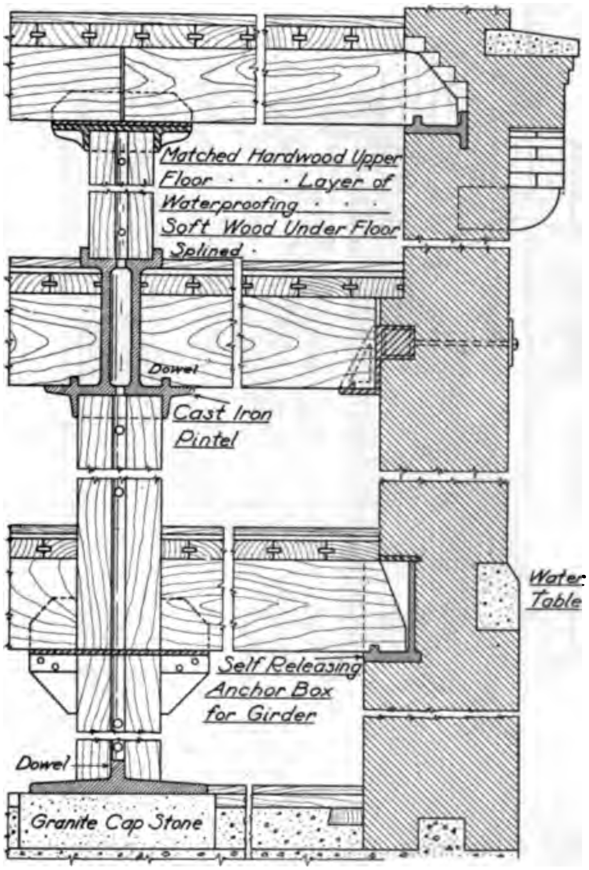

FIG. 6 DETAILS OF MILL CONSTRUCTION

attention at this moment and are being embodied in modern structures. It is admitted that such is the case in individual instances, but as a general rule—No! Facts are against such a contention. Very few factories and even few office buildings which have been designed and built in this country in the last few years are truly safe for the occupants. It is not asserted that they fail to comply with existing codes, laws and regulations, but as long as each individual city has its own building code, often based upon the particular market of its locality and dominated by the building products of the controlling political parties, is it not too much to assume that scientific results will be obtained?

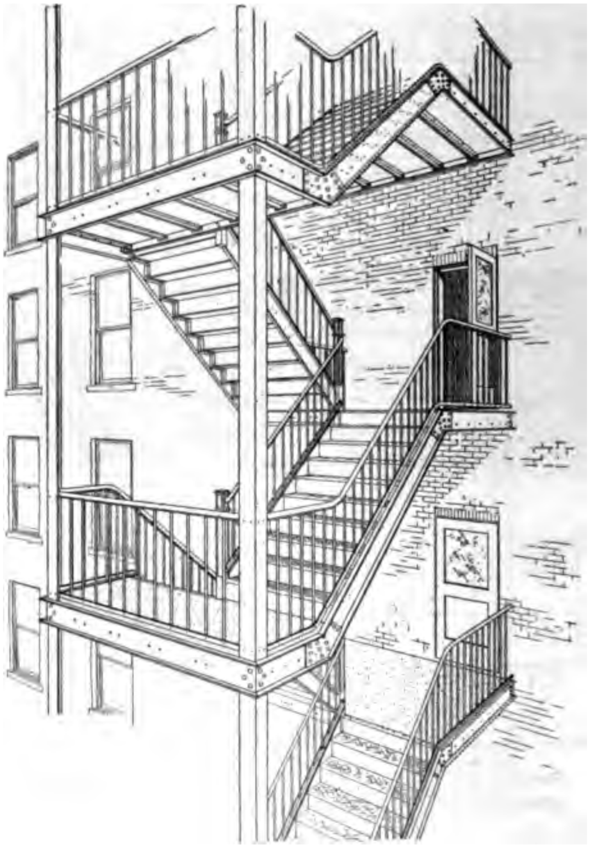

FIG. 7 APPROVED TYPE OF OUTSIDE EXIT STAIRWAY
Stairs Perpendicular to Wall

20 As practical men, however, it would be impossible to condemn all buildings now in existence which do not embody the ideal safety principle. Our plan must therefore include existing buildings as well.

21 Taking the building, for example, where adequate means of egress has not been provided, it may be necessary to overcome this

deficiency by means of outside exit stairways of a type as shown in Fig. 7, but under no circumstances should the so-called "fire escapes" as commonly known be countenanced, that is, exits with such objectionable features as grid-iron balconies connected by open stairs or ladders, or exits so located that they adjoin windows.

22 The horizontal exit obtained by means of bridges connecting

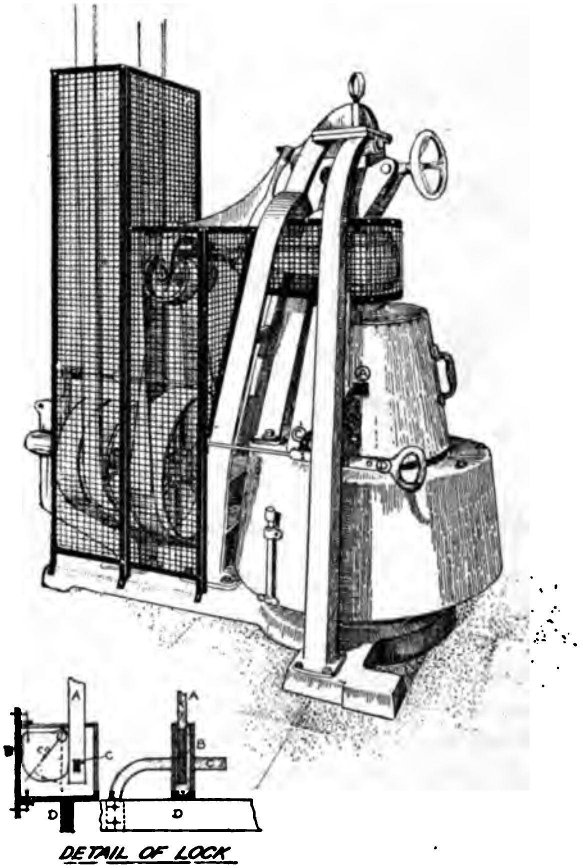

FIG. 8 DOOR-LOCK ON OVER-DRIVEN EXTRACTOR

adjoining buildings, or in the case of large floor areas the breaking up of the area with fire resisting partitions and automatic fire doors, are very effective means for promoting safety from the fire hazard in many existing buildings and it is quite incomprehensible why those plans are not more extensively used. This method places safety zones immediately next to the danger zones.

23 It is naturally impossible in a paper of this nature to go largely into details. There are many more questions which properly should be considered in connection with buildings, but it is hoped the foergoing will illustrate the points involved.

24 We next come to the selection and installation of machinery and equipment generally. Where machine tools are to be used, only those should be specified which have all possible safeguards embodied in original design; that is, as an integral part of the machine itself. A safeguard on a machine should not be an extra attachment or an after-thought; it should be as much an original part of that machine as are the gears or the pulleys. A manufacturer of

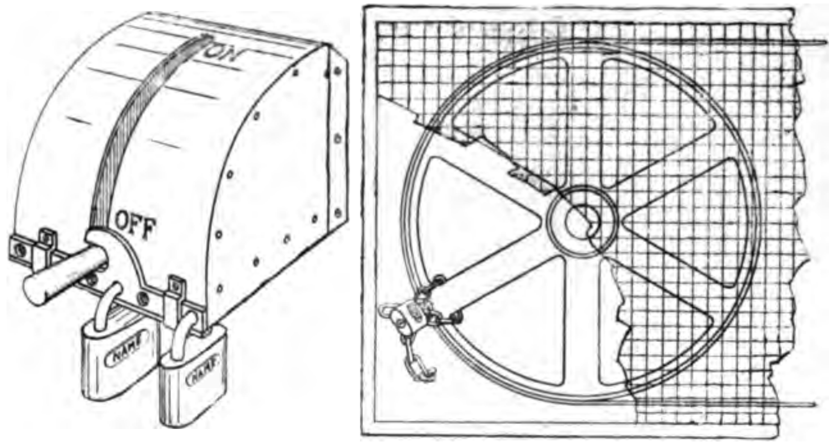

FIG. 9 TWO EXAMPLES OF PADLOCKS AS SAFEGUARDS

machine tools should consider it an insult to have a purchaser ot his product add parts to it after it has been installed.

25 Gears are necessary on certain classes of machines for the transmission of power, but no engineer will admit that these gears have to run openly in order to perform their function. Knowing that as long as they run openly they present a hazard, and knowing further that it is an unnecessary hazard and that it will not reduce the efficiency of the machine to eliminate it, it must appeal as a logical proposition that such elimination should be effected in original design rather than after the machine has been installed. This applies equally to all other points creating an accident hazard.

26 In order to meet the principle of safety a guard for a machine should be so constructed as to prevent the removal or misuse of its essential parts. Wherever possible, it should be so designed in

relation to the machine to which it is applied that the machine cannot be operated unless the safeguard is in position. This may seem to present insuperable difficulties in the design of certain types of machines. It has been the author's experience, however, that where concentrated study has been given to that particular principle in machine design, it has been met in every instance. Fig. 8 indicates the principle applied in the door-lock on a laundry extractor. As formerly stated, it is the human element which must be overcome in the application of safeguards. It is often noted in advertisements of certain guards that arguments are used such as:

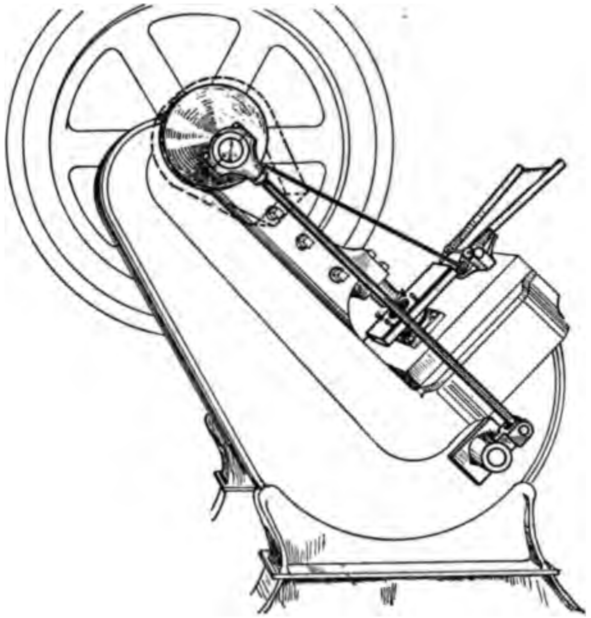

FIG. 10a STAMPING PRESS GRAVITY FEED CAM-ACTUATED EJECTOR

"Can easily be removed"; "Can be thrown out of the way at will"; and so forth. The real arguments for superiority of safeguards should be: "Cannot be removed;" "Can only be removed or adjusted by key"; "Is necessary for the operation of the machine"; etc. This principle is again illustrated in Fig. 9.

27 A further study of the question of safeguarding has brought out the fact that the application of a safety principle often increases the efficiency, as, for instance, in the case of stamping presses equipped with slide and gravity feeds, as shown in Fig. 10. It is a fact well established in stamping press operations that that method of feed

naturally increases the output on the machines where applie often produces that result. It must be noted, however, tha automatic feed as such does not in itself form a complete gua of that press. Some further protection must be provided, m it impossible for the attendant to put his or her hands in the danger zone during the downward stroke of the ram.

28 If from a construction standpoint the building is pr provided with all safety features, and the machinery and equip with its belts, pulleys, gears, set-screws, couplings, etc., embo safety principles, there are still other hazards which must be

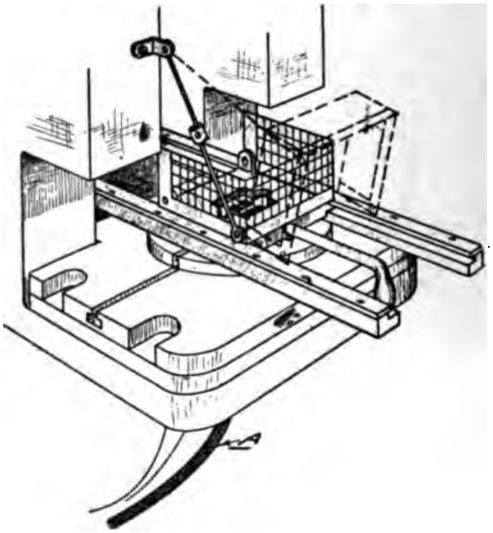

FIG. 10b STAMPING PRESS SLIDE FEED AIR EJECTOR
A = Air Ejector Hose

fully considered and the principle for elimination of these h established. The slipping hazard is one of the most importan careful study of many accidents as reported has brought out th that slippery surfaces constitute one of our largest contrib causes to such accidents. That the slipping hazard is easil can be practicably eliminated by the introduction of anti-slip su on all stairs and on all floor surfaces where employes or other pants of buildings frequent is an established fact. It is partic necessary that such anti-slip surfaces be used around machiner

29 Poor light is also a potent contributory cause to acci It must be admitted, however, that more has been done i

particular phase of hazard elimination than in any other, thanks to our exceedingly efficient illuminating engineers.

30 Having proven, so the author hopes, the necessity of safety as an essential principle in the design of buildings and their equipment; having further proven the practicability of such safety principles; and having reinforced this by showing the apparent need of standardization in the premises, the question is before us of where and how shall we obtain such standards? The answer lies in another question: Where did we get our industries? Who is responsible for the hazards in these industries? We—the engineers. We have created the machines used and it is up to us to eliminate any unnecessary hazard which these machines have introduced.

31 The unfortunate phase of the whole subject is that up to now safety has been dealt with as a legal subject. We have left it to our courts, to our lawyers, to decide whether or not a building or a machine was safe for its occupants or operators. Now, the author maintains that safety never was and never should have been made a legal question. It is not a question of law; it is in every instance a question of fact and who is more competent to deal with questions of fact than engineers, especially when these facts pertain to their own business?

32 To give a typical illustration of legal and engineering standards: The hazard of moving elevators and open shaft doors has long been recognized through occurrence of hundreds of fatal and serious non-fatal accidents. Legislators in different municipalities and states have passed statutes to the effect that it shall be deemed unlawful for an elevator operator to start his car until the doors of the shaft are closed and latched. This is a typical standard produced by legislators. But does this prevent the occurrence of elevator accidents from open doors? It does not, because it is impossible to enforce the act in time to prevent the accident. The only thing that such a statute accomplishes is to establish responsibility for the accident after it has occurred.

33 Let us consider briefly how an engineer would solve the same problem. He would analyze the mechanism of the elevator, the door, and the motive power. He would find the contributing causes which produced the hazard and re-design his elevator so that it would become impossible for the operator to reproduce the dangerous conditions, and would compel him to perform a certain act before he could get motive power—the act in this case consisting in closing

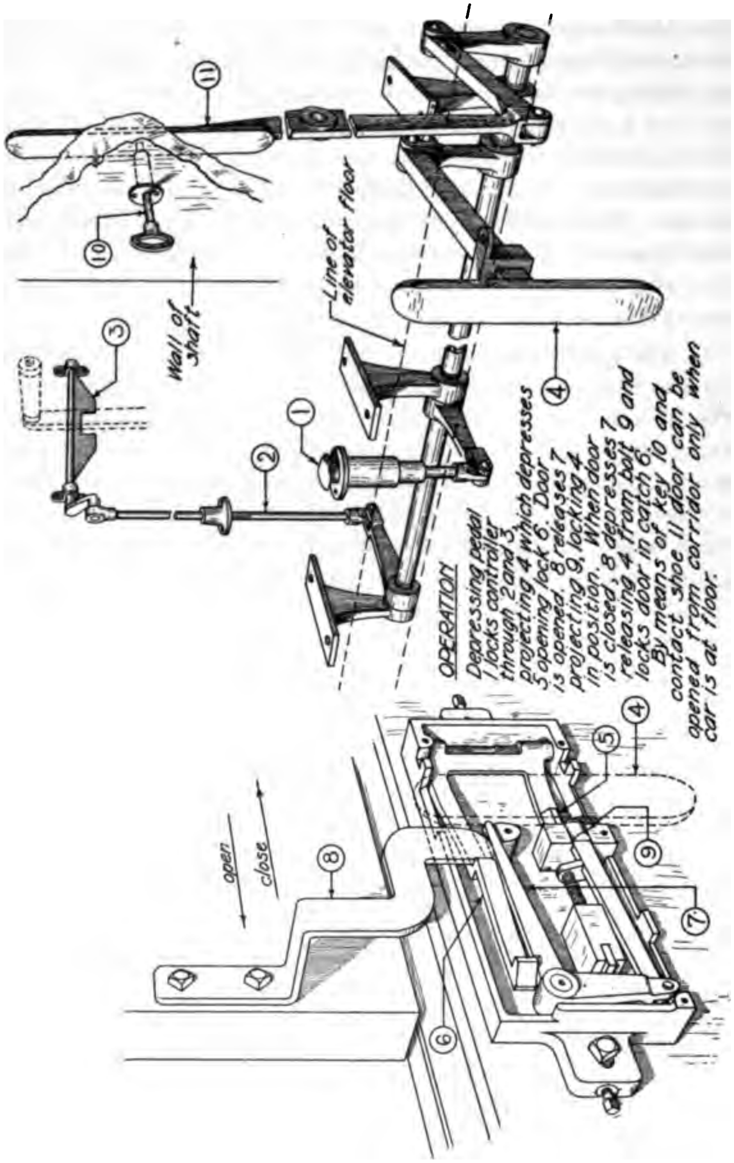

Fig. 11 Automatic Elevator Door and Controlled Lock with Foot Release

the door in order to move his operating lever. A device to illustrate this point is indicated in Fig. 11.

34 When engineers have developed such a principle and shown it to be a practical one, our legislators may well assist them in giving legal force to it, but this should be done without any unnecessary phraseology. To write a law book on a question of fact which can be expressed in a line is folly.

35 Before closing, it is desired to express a further word in respect to building design. Our architects may well say this is an architectural problem and one which should be left to them to deal with and solve. Let it be clearly understood there is no desire upon the part of the author to dictate what is and what is not proper building design. Let it further be understood that there is no intention to advocate that our architects should sacrifice art, utility or any other desirable feature of building construction in the interest of safety. But it is maintained,—and this with a fair degree of conviction,—that the architect of the future must combine safety, convenience, efficiency and art in order properly to serve the interests of society, with always,—as in the case of the engineer,—"Safety First."

36 Let it be further understood on the part of the author's brother engineers that no attempt is made or will be made on the part of the organization which he represents, namely, the Workmen's Compensation Service Bureau, to enter into specification details in any standards which that organization may be responsible for. In fact, the term "standard" as used under the auspices of that organization refers solely to standardization of principles, leaving it then to the engineers of the various industries to apply these principles in their particular field, a task for which they are far better qualified than any one else.

37 The Workmen's Compensation Service Bureau, however, has a vital economic interest in the premises. It is an organization composed of the leading liability insurance companies on this continent. It deals with the matter in a concrete sense in-so-far as its constituent company members insure the liability occasioned by any hazard present in buildings and equipment. To value these hazards properly and to eliminate them, as well as to avoid discrimination, specific standardization is essential. We have undertaken it in the interest of our own business, but the interest is equally as great on the part of the industries insured because the rates for insurance are in direct ratio to the presence or absence of any hazard.

38 In conclusion, the author believes it a self-evident fact that the engineers as creators of our industries owe it as a duty to these industries and the nation of which they form so important a part to exert their utmost efforts in the conservation of life and property which are so essential to the economy of the individual plant and so essential to the nation as a whole. The subject should be treated with the consideration which some of us at least feel it justly deserves. Our methods for reducing waste, whether it be material or vital, must be improved through preventive methods and none is better fitted to lead such a movement than the engineers. It is not proposed that a new branch of engineering shall be established for this problem. It is not proposed that any new societies shall be formed to deal with the question. We have too great a variety of societies already. The real object must be to make each branch of the engineering profession in the true sense of the word "Safety Engineering."

DISCUSSION

Frederick R. Hutton. In the historic development of any productive industry, the first consideration had to be to make something that would do the work,—that was the condition for the seventies to the nineties. Then came the period of improving the efficiency with which the result was attained. In the early part of the twentieth century, the third stage was reached; that is the evolution takes the forms of, *first*, getting the result; *second*, getting it economically and efficiently, and *third*, getting it with safety to the men concerned.

Mr. Hansen has not mentioned what I know he very clearly sees, and that is that the most effective way for designing engineers to attain the desired results is by means of what is called the interlocking principle. In a laundry machine, for example, it should be impossible to shift the belt to start, until you have closed the machine and locked it, and got your hand out. It should be impossible to start the elevator until the door has been closed; correspondingly, it should be impossible to open the door until the elevator has been stopped.

Mr. Hansen has not referred to one unfailing method of securing safety, and that is that it should require the use of both hands to start a machine. If the operator must pull a lever with the right hand and then pull a lever with his left, to either start or stop the machine, he has no hand free which he can get in the way of maiming forces.

Supplementing the author's statement of basic principles, and his definition of "practical" safety devices, I would say that these devices must be such that they do not diminish production.

In closing, may I express my pleasure that the Society should have devoted a session at this meeting to the discussion of this most important subject. I have been interested in the subject of safety since about 1895, when nobody thought much about safety engineering. I did not myself until it was brought to my attention that the practice in Europe was far ahead of our own. We are catching up, and it is to the credit of the engineer that this result should come about, and to organizations such as those of which Mr. Hansen is representative.

FRANK E. LAW contributed a written discussion, stating: I doubt whether safety standards should be absolute, should be hard and fast. It seems to me that there are often several ways of accomplishing a desirable end, and one way should not be adopted to the exclusion of others. All that should be necessary is that the particular way adopted will achieve the desired end. If some latitude is allowed in the choice of means, we can often secure the cooperation of a workman through his pride of authorship. A workman is much more likely to use willingly and effectively a guard devised by himself than one imposed upon him by authority. I think, therefore, that Mr. Hansen is proceeding along the right lines when he announces "that no attempt is made or will be made on the part of the organization which he represents, namely, the Workmen's Compensation Service Bureau, to enter into specification details in any standards which that organization may be responsible for. In fact, the term 'standard' as used under the auspices of that organization refers solely to standardization of principles, leaving it then to the engineers of the various industries to apply these principles in their particular field. . . ."

I would urge upon engineers the propriety and desirability of checking all designs of structures and machinery prepared in their offices from the safety standpoint, and when purchasing machinery to include safety requirements in the specifications.

I would express doubt as to the advisability, in legislating with regard to safety measures required of employers, of setting forth in detail the guards to be provided, but would instead recommend a statement of the principles, imposing upon the Commissioner of Labor the duty of issuing regulations covering the details.

It constitutes a distinct reproach to the engineer that he has done so little in the field of safety engineering. At the same time it is only fair to bear in mind that he has often been hampered by his employers and by the limitations imposed by competition and expense.

It is interesting to note, however, to what a considerable degree the work of the Society has had to do with things that lie within the realm of safety engineering. The design and construction of structures and machines to secure maximum strength, the choice of proper materials, is safety engineering. A large part of the work of the Boiler Code Committee had to do with the proper design and construction of boilers from the safety standpoint and the choice of proper steel for the shell, furnaces, stays, rivets, and other component parts. But there is much more that the members of the Society can do, and the field of subjects for papers dealing with safety problems is practically unlimited.

Safety engineering, unfortunately, like every other effort for betterment, often runs afoul the matter of expense. The expense of perfect design, materials, construction, conditions, is prohibitive. We have constantly to bear in mind, therefore, the distinction between what is desirable and what is feasible, having regard to the matter of expense. The problem of the engineer is to secure the maximum of safety at the minimum of cost. It is easy to wreck a promising safety program on the rock of expense.

The insurance companies have done and are doing a great deal to help solve the problem of the prevention of accidents. They have published literature on the subject, have made and are making safety inspections, and have established service bureaus and inspection boards to make merit or schedule rating inspections for the purpose of determining discounts and additions to the rates, as safety measures are or are not adopted by employers. The companies earnestly hope that they will have the full and complete coöperation of engineers everywhere in the great work of promoting industrial safety. Mistakes will be made at the outset, for we are doing pioneer work, but we hope these will be as few as possible, and we hope also that we shall have the benefit of constructive, not destructive, criticism of our friends and fellow-workers, the engineers.

JOHN H. BARR (written). The author states that only those tools should be specified which have all possible safeguards embodied in original design. It may be difficult at the present day to get always the machine required with all danger points guarded in accordance with modern ideals, but the statement that safeguards *should* be included by the maker and as a part of the original design is absolutely sound. Not only on aesthetic considerations, but in the interests of effectiveness and true economy should this be done. Until builders generally adopt the practice of supplying "built in" safeguards (and for some years afterwards while old machines are still in action) the poultry netting, picket fence and ashcan types of makeshifts will persist in offending the eye. It is not necessary to dwell on the unsightliness of these contraptions, much as we may rejoice to see them where no protection previously existed. In many cases they are not as effective as would be well-considered designs made in the shops of the builders.

It is, however, the consideration of real economy which may be expected to make consistent well designed safeguards the rule rather than the exception. In the first place, the introduction of the "merit rating system" will be an incentive to purchasers to call for all necessary safeguards with the original purchase. The history of factory fire insurance proves that owners will respond to higher requirements when convinced that there "is money in it." When purchasing agents show a preference to the bidder whose product has these features properly incorporated, the seller will begin to take notice.

I believe the progressive builder is disposed to anticipate such demands and that he would voluntarily go much further in the matter than he has yet done if there were adequate assurance that *any* design he can offer will meet the statutory requirements, the insurance standards and the idiosyncracies of inspectors wherever his product is sold. At present, unfortunately, he has no such assurance. The chaotic condition of today must be remedied before the situation can become really satisfactory.

Such work as that done by the National Machine Tool Builders Association, the Abrasive Wheel Manufacturers and the Machine Shop Practice Committee of this Society, on the Safety Code for the

Use and Care of Abrasive Wheels[1] will have great influence on legislation in the various states and will tend strongly toward uniformity of requirements.

In a letter to the sub-committee of this Society on Protection of Industrial Workers, Dr. S. W. Stratton, director of the U. S. Bureau of Standards, said:

We are in entire sympathy with your feeling that the great obstacle to the adoption of more complete standardization from the safety standpoint by manufacturers, is the diversity of requirements and the conflict with existing regulations in different sections of the country and even by different jurisdictions in the same territory.

This statement sums up the matter quite conclusively.

Adequate, yet reasonable and practicable codes, generally acceptable to all interests, must be worked out for each class of mechanical equipment. If this work is well done, it is not unreasonable to expect that eventually it may be generally adopted. While other organizations may be better adapted to promote the important work of safety propaganda and while the prosecution of various phases may be properly left to different agencies, it is believed that this Society is peculiarly called upon to take a leading part in formulating several of these standard codes and in exerting its influence toward securing their adoption throughout the United States.

JOHN W. IRWIN[2] (written). The effort to standardize safeguards would be more immediately effective if there could be devised some way to show manufacturers the waste resulting from lack of standards. There are so many men who have their own ways of doing things, and who are unwilling to look about them and see what has already been done by others. Many know the end to be attained and have a general idea of how to reach it, but have neither the knowledge nor the equipment of the organizations which specialize on the subject.

The matter of mechanical safeguards and their standardization is an engineering question in the sense brought out in the paper, but the profession can render a further service for industrial safety. Engineers can develop methods of guarding, but they can also be teachers and leaders. They can disseminate information about what has already been done. They can utilize countless opportunities of pointing out the waste of experimental, haphazard safeguarding.

[1]Published in this volume.

[2]Vice-Pres. and Secy., Universal Safety Standards Co., Philadelphia, Pa.

The waste from ignorance in guarding is not to be compared with the cost of industrial accidents, but it is sufficiently large to warrant the attention both of engineers and safety men generally. This waste may be reduced, and gradually eliminated, by concerted action on the part of all who come in contact with people who are trying to find their own ways of doing things that have already been successfully accomplished.

The desirability of standards is quite sufficiently obvious. The thing that is now needed is a widespread knowledge that a long step in standardization has already been taken. I refer to the Universal Safety Standards as already developed by Mr. Hansen and his associates.

Our company takes the attitude that mechanical safeguards constitute the beginning of accident prevention; and the education of the workmen towards the elimination of accidents resulting from ignorance and carelessness is of equal, if not of more importance.

We have found that safety work as a whole must comprise the application of mechanical guards, the elimination of such general hazards as may exist in a plant, and then a systematic, progressive, never-ending campaign to develop the minds of the men in the direction of safety—of carefulness through force of habit.

The men must be reached by varying methods. Frequently, general welfare work, plant sanitation, social features and the like, are effective in proving good faith on the part of employer, and bringing the men to believe that the influences for safety are for their own good.

An educational plan cannot be operated until the matter of mechanical guards has been covered. You, as an employer, cannot consistently ask and require that your employes exercise caution, and cultivate the spirit of universal safety, until you have proved your good intentions by doing what you can to make accidents impossible. In a recent issue of a prominent magazine an illustration showed a workman operating a machine bearing a placard: "Think Safety"—a flywheel revolved within a foot of the operator's head, and it was not guarded; there were no belt guards visible, and exposed gears were easily discernible in the picture. This merely shows that there are multitudes of little things, and big things too, yet to be learned, both by the average plant operator and by the one supposed to know better.

No better "Safety Method" can be conceived than an ideal combination of high engineering skill, 100-point common sense, and the experience of the successful teacher. The devising of efficient guards, removal of building hazards, and the like, are engineering subjects purely. What we urge upon all engineers is to go further than that, and lend influence, example, and earnest coöperation in extending the work beyond the point of engineering as such, thus advancing the ideal condition where there is no machine unguarded, no unprotected building hazard, and where every workman knows what to do to protect himself and his fellows—and invariably does it.

WALTER M. KIDDER contributed a written discussion, viewing the question from the commercial standpoint. He stated that opposition to measures intended to prevent accidents arises from self-interest both of workers and of those in authority over them.

The owner, the stockholder, fears in the compulsory adoption of accident preventive appliances and measures a new drain upon the earnings of his business. Nevertheless, he may become a powerful ally of this latest move toward industrial efficiency when he realizes the practical economy of accident prevention in the long run.

To every serious accident there are many minor ones and each represents an interruption of industry—idle machines while fellow workers aid the injured one, damage to product, damage to machinery, and often a gang or a department thrown out of balance by the absence of an essential worker. These things do not merely add to the difficulties of the foreman in charge and of his superintendent; they cost the stockholder money. Just so far as he realizes this will he desire accident prevention, because it increases his profits by decreasing their impairment.

If the possible potent aid of the stockholder is utilized by engineers and a broad view is taken of *the commercial aspect* of the standardization of safety principles, the added impetus of this powerful influence will lighten the task of the engineer and leave him the more free to deal with the technical side of the matter, which holds quite enough problems to engage his best energies.

DAVID S. BEYER[1] (written). The author, in stating that the real goal must be to make each branch of the engineering profession

[1]Mgr. Accident Prevention Dept., Mass. Employees Insurance Assn., Boston, Mass.

Safety Engineering, has established a motto which might well hang over the desk of every engineer in the country. It is a primary duty of the engineer to eliminate needless expense and waste, and during the past few years it has been shown conclusively that one of the most appalling wastes in industry is that caused by preventable accidents.

The most effective time to prevent accidents is when the construction drawings are being evolved under the pencil of the engineer. Then the erasure of a line, or a few additional strokes of the pencil, may mean the saving of a human life or the elimination of hundreds of dollars of unnecessary expense in making changes later.

It might be said that there are two equal doors to the House of Safety, the first marked Mechanical Safeguards, the second Safety Education. The engineer has always held the key to the former through his control of the design and construction of new buildings and machinery. This phase of his work is usually most prominent in our minds, and I notice Mr. Hansen has confined his remarks to it. The attitude of the man who runs a machine is just as important, however, from the standpoint of accident prevention, as the condition of the machine. On account of the ever-increasing influx of engineers into all industrial lines and the fact that executive positions are being filled more and more by men of engineering training, the matter of safety education of employes should accordingly be mentioned as a vital form of standardizing safety principles.

The safety organization has become an integral part of the working organization of hundreds of our foremost industrial plants, and the knowledge of how to organize the working force for safety and how to interest the employes in accident prevention is an essential part of the engineer's equipment, if he aspires to the largest fields of usefulness.

James O. Gibbons (written). One of the most interesting points brought out in this paper is that safety should be considered as an essential element of good engineering design, rather than as something to be added just because the customer or the law demands it. Of course this idea has been gaining ground for some time, but if it can be shown that the liability of a machine to cause injury calls for an actual cash charge against it as insurance of the risk, we are putting safety efficiency on the same plane as production efficiency.

As has been pointed out, the loss to the community through preventable injuries is very great. It is perhaps not quite so easy to show just what the loss to the individual employer is, because in addition to the time taken laying off for recovery, there is the moral effect produced by accidents, the effect of which on efficiency is hard to estimate.

Knowledge on this subject of safety is a great help, because however much we may wish to do the right thing for its own sake, the surest way to put safety on a sound foundation is to show that it pays as a business proposition.

GEO. M. PRICE[1] (written). I regard this movement for the standardization of safety principles by safety engineers as one in the right direction and an important advance in the safety and sanitation of industrial establishments. Your Society and the Workmen's Compensation Service Bureau are doing what the German Trade Associations have been doing for a great many years, and the results of such standards of safety devices, etc., are bound to be a great improvement in the safeguarding of the workers' lives and limbs in factories and workshops.

May I not suggest that there is also a great need in extending the standards not only to safeguards and safety devices, but also to the sanitation and general hygienic environment in industrial establishments. May I also make the suggestion that your Society endeavor to introduce the study of safety engineering in the various technical and engineering schools so that the future managers and heads of industrial establishments should be fully prepared to put into practice all possible safety devices for protection of workers.

Another suggestion would be that the description of safety devices should be couched in a simple and, if possible, non-technical language so that the workers and mechanics may be able to understand them and put them into practice.

THE AUTHOR. I desire to express my appreciation of the interest which my paper has created, as indicated by the discussion.

[1]Director, Joint Board of Sanitary Control, New York, N. Y.

No. 1511

MODERN MOVEMENT FOR SAFETY FROM STANDPOINT OF MANUFACTURER

BY MELVILLE W. MIX, MISHAWAKA, IND.
Member of the Society

The successful salesman is the man who first sells himself on his proposition, who becomes so saturated with its merits, so enthusiastic in his beliefs that conviction is automatically transmitted to the buyer. Substantially the same principle holds with the modern safety movement; no manufacturer will make real progress in the movement until he takes a personal interest in it, and is able to transfer his enthusiasm and earnestness to those directly in charge of work and, through them, in turn, to the workers themselves. A form of evangelism is required that arouses the spirit of all concerned.

2 In these days of Workmen's Compensation and high damage awards in employer's liability suits, the manufacturer will not be slow to grasp the main idea that "saving is earning," and act accordingly. He will find that the market is full of "guards" of all kinds for all sorts of protecting purposes and places, and a large supply of propaganda calculated to encourage him to spend money in an endless way on preventive appliances. He will find an army of inspectors ready and active with unlimited recommendations and orders for every machine and for every nook and corner of his factory. He may be persuaded to buy them all and to protect without limit, but if he is not himself converted to their efficiency and need, and if he has not the coöperation in spirit of his employes, foremen and superintendents, his financial investment in them counts for little.

3 In no place is there a better or more practical way to develop the "save my pal" movement than in a factory or mill. Not only must the worker look out for himself, but his every thought, when engaged in hazardous work involving operations with which others

Presented at the Annual Meeting, December, 1915, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Contributed by the Sub-Committee on Protection of Industrial Workers.

have to do, must be "how about my fellow worker?" When in a factory organization such thoughts become automatic from president to "sweep" then money spent for guards becomes a real capital investment; but without that spirit its principal effectiveness is to expand the inventory and keep up the insurance rates.

4 We spend thousands of dollars in fire protection in order to reduce the hazard and save in operating expenses. We drill our factory fire brigades without thought of the cost, because we are educated by actual experience to realize what such losses mean in actual money and stoppage to business. It has taken years to learn the lesson, but we have done so, and fire protection is a recognized essential part of wise business administration. We do not stop at the installation of fire prevention appliances, but we put them under constant, careful inspection; we instruct every man in their location and uses; we leave no stone unturned to insure the effectiveness of the apparatus when called upon.

5 But, with safety "guards," until we can see the same benefits in providing the same relative protection to the individual, until we can bring him to see how it affects him physically and financially to "keep right with the guards," we fail to realize the full benefit of the investment of money and time spent in education. As a concrete example of what may be accomplished in this direction, I give here figures (Table 1) showing the experience for the past five years of the company with which I am associated.

TABLE 1 COST OF ACCIDENTS COMPARED WITH PAYROLL, AND TIME LOST COMPARED WITH TIME WORKED

FIVE YEAR RECORD

Year	1910	1911	1912	1913	1914
Total cost of accidents for each one hundred dollars of annual payroll, including first aid, hospital, bills and claims, if any...	$0.503	$0.228	Score board adopted $0.112	Score board in use. Actual cost $0.079	Score board in use. $0.070
Time lost due to accidents beyond the fraction of the first day. Percentage of number of men-days lost to the total number of men-days worked.....	No record kept	No record kept	Fraction of 1 per cent 0.394	Fraction of 1 per cent 0.1924	Fraction of 1 per cent 0.1165

A wage bonus of two days extra pay for all men in a department having a perfect score for one year was based on the general effect of a cash bonus, but more particularly on the fact that a man will do for the benefit of the gang or department what he would consider beneath his dignity to do to safeguard himself against minor accidents.

6 We have numerous cases on record where men have cautioned each other with the admonition that such recklessness would endanger the standing of the department. One instance where a foolish act of one man lost two days' pay for a whole department has proven a wonderful object lesson to the entire plant.

7 In these days of scientific management and administration, there may be a tendency to the thought that, as such things do not seem to show immediate cash credits on the balance sheet, their importance may be set back for the moment for motion studies, rewards on operations, etc., but there is no economy in tolerating a condition that may require new men to be broken in to replace those impairments that a little time spent in educating men and in arousing them to the Safety First pitch might have avoided. It is impossible to place a money value on a pair of eyes, a leg or an arm; and while the expression "the human scrap heap" displeases us, we must know that there is such a thing, and that the right word at the right time may be effective in saving a fellow worker from that untimely and unseemly misfortune.

8 Why not be practical and do all we can whenever we can to prolong the usefulness from the physical standpoint, if from no other, of every man or woman that we employ. Experience has developed that guards are not especially effective if, with them, we do not have the coöperation of the workman. If we make things automatically safe, we are quite as apt to do the worker an injustice in weakening his own spirit of self-preservation. Workers must recognize danger,—they must respect it, and be ever watchful in every phase of life as well as when at work. The family must have it drilled into them; schools must preach and teach it; the citizen must be a party to the campaign of education, as only in that way may we be assured that "Thinking First" comes before Safety First.

9 Out of 30,000,000 male workers in the United States, about 7,250,000 are engaged in the general manufacturing lines. The annual loss due to fatal accidents is about one to four thousand, the lowest of the eighteen principal occupational groups; yet from the standpoint of public attention, the manufacturer is held up as the cause of nearly all physical suffering and loss due to occupational disaster.

10 This situation would not exist if the manufacturer were

more personally interested and inclined to fight for his rights and a proper recognition of his standing on the score board of national employment. Out of 22,500 fatal accidents in 1913, only 1,819 are attributed to the manufacturing occupations,—comparatively small indeed. Yet it can be much smaller if we will only approach the subject from the personal standpoint, evangelize with our employes, and arouse in them a spirit of coöperation and thoughtful, prudent regard for themselves and their fellow workers that will be automatic, and a habit for protection and carefulness, of which they cannot break themselves even if they should wish to. That is the manufacturer's work today.

11 Numerous plans have been devised for arousing and maintaining this personal interest; score boards, bonuses, special favors to winners, lectures, moving pictures and many other plans are being used. To the extent that a management believes in these plans and encourages their uses, just to that extent will it be rewarded with the association of an unimpaired, efficient organization of contented, prosperous workers.

DISCUSSION

Luther D. Burlingame agreed that while we are giving attention to the matter of guarding and protecting our machinery and buildings, we should give equal, if not greater, attention to engendering the spirit of safety among our employes, and should consider the various means which can be used to develop the safety habit among the workmen.

He was not fully in accord with the author as to the statement that putting guards on machines might make the workmen lax in regard to safety matters, and therefore increase the hazards rather than otherwise. He believed we should do all the guarding possible, and protect the machines as fully as possible. Even then there will be hazards enough to keep the workman alert and make it advisable to preach safety in the organization, and preach it so hard that the spirit of the workman shall be that a part of his value lies in working in a safe way, and that the spirit of the foreman shall be that a part of his value lies in seeing that his workmen work in a safe way.

Frederick R. Hutton. I would like to contribute commendation to the statement that "a man will do for the benefit of the gang or department what he would consider beneath his dignity to do to safeguard himself against minor accidents." There is a splendid philosophy in that sentence. The difficulty with many of us is that we are inclined to take the position that we are immune from accidents which might befall someone else less fortunate. If, therefore, we can safeguard the less gifted alongside of ourselves and make that protection our responsibility, we shall have done a great deal in securing safety.

To this end, I would like to emphasize the creation among the working people themselves of committees of safety. After a workman has served on such a committee, and has himself seen the philosophy and importance of the habit of safety, he will not only find it very much easier to protect his reckless neighbor, but he himself will not offend. He will keep himself from doing that which, as a member of the safety committee and serving to prevent accidents for a week, or for a month, he was called upon in an administrative way to prevent in someone else.

Frank E. Law emphasized the point that securing the coöperation of the workmen was often more important in preventing accidents than installing safety devices. He instanced the case of a large manufacturing corporation which had spent upwards of five million dollars in safety work. It had been estimated that not more than 15 per cent of this vast sum had been *directly* effective in preventing accidents, but the money had been well spent because the expenditure resulted in securing the coöperation of the workmen. When the employer himself does all that is reasonably possible, he is pretty certain to evoke coöperative response from his employes.

John H. Barr fully agreed that the importance of the educational work is frequently greater than that of merely providing guards. He said, however, that the Committee on Protection of Industrial Workers had decided its work should be confined, at least for the present, to an attempt to develop and bring about the adoption of standard Safety Codes, because so many other organizations and agencies are working effectively on the educational side.

The influence of this Society, he believed, would help in the adoption of such standards, whereas other organizations have made a specialty of propaganda work. He said the committee is fully in sympathy with this work and desires to coöperate, but it has selected what is conceded to be the narrower field, because it seemed to be the one in which it can work with the best effect at the present time.

F. D. Patterson[1] said that from the standpoint of manufacturing, safety paid dividends not only in dollars and cents, but also as a matter of fact in added happiness to every employe, by the prevention of industrial accidents.

He dwelt on his experience at the plants with which he is connected, showing that before active safety work was undertaken the accident curves were mounting upwards, but upon adoption of safety measures, these plants not only had their experienced men saved from injury and available for productive labor, but thereby avoided the necessity of having inexperienced workmen laboring in their place; and he emphasized the fact that from the accident standpoint the new employes form one of the perils of modern industry.

He considered that the work in accident prevention in industrial plants comprises two distinct features:

a Guarding of all accident hazards by mechanical guards which will reduce the number of accidents 20 per cent.

b A campaign of education bringing home to every workman the facts that he is personally responsible for the accidents that occur and that he ought to constitute himself a safety committee of one to see that accidents do not occur either to himself or to his fellow employes.

Leonard Waldo said that he had noticed that in some of the great steel works the Safety First idea has so permeated the administrative staff and the men that you will find notices constantly posted for semi-weekly, weekly and bi-monthly meetings on safety subjects. Prizes are offered for the best safety work; certain men are appointed to supervise safety matters around the different parts of the plant; highly organized hospitals are established, and there is a wonderful movement on the part of the intelligent control of great works towards safety among operatives.

A bad accident almost invariably occurs when the sense of danger is not present because of some new modification or some novelty of the process. If we codify the principles of our rules of safety, we are of the greatest possible help to the administration of the Safety First movement.

[1]Director, Dept. Sanitation & Accident Prevention, Harrison Bros., Philadelphia, Pa., The J. G. Brill Co., and the Elec. Storage Battery Co.

No. 1512

THE ATTITUDE OF THE EMPLOYER TOWARDS ACCIDENT PREVENTION AND WORKMEN'S COMPENSATION

By W. H. Cameron,[1] Chicago, Ill.
Non-Member

There was a time when workmen were expected to assume the hazards of their work, and employers, through insurance companies, paid as little as possible for accidental injuries or deaths. In one State only seven deaths in the hundred were paid for. Employers were not inhuman—workmen were careless or reckless and simply suffered for their foolishness.

2 This condition has changed. In thirty States the democracy has said that the financial burden shall be borne by the industry. This meant social and industrial revolution. Many employers predicted that the whole movement was a transient outburst of humanitarianism and that it would die out in the course of time, just as fashions in clothes changed. There were others, however, who had an overwhelming sense of the safety movement, who realized that it spelled industrial efficiency and justice and was a symptom of progress in civilization.

3 The census reports do not show the number of employers for or against Workmen's Compensation Laws, or industrial safety measures, but a radical transformation in relationships to this problem has brought about many and varying attitudes towards it. This paper is an attempt to interpret these tendencies in as unbiased a manner as possible.

4 Interest in accident prevention is not new. For many years our Government has been a leader in providing a measure of protec-

[1]Secretary, National Safety Council.

Presented at the Annual Meeting, December, 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Protection of Industrial Workers.

tion for mariners through the lighthouse and life saving services, the Interstate Commerce Commission has ordered the use of certain safety appliances for railroad cars, and other agencies have contributed to the general movement.

5 About twelve years ago, however, America began to realize that preventable industrial accidents were a national evil of appalling magnitude. The agitation for Workmen's Compensation Laws brought forth astounding facts and figures. It was shown that both the number of accidents and the economic losses occasioned by them were increasing in ratio more rapidly than the output of our industries, but it was not until the extent of the evil became fully apparent that the safety campaign began.

6 For many years labor unions had demanded safeguards for machinery, and we had on our statute books various kinds of safety requirements, which, though perfunctorily observed, were not really enforced. As soon, however, as Workmen's Compensation Laws transferred the burden of industrial accidents from the workman to the industry the "Safety First" movement began. Mention should be made, however, of some of the large corporations who had "seen the light" several years previously, and had voluntarily taken decisive steps not only to safeguard their employes, but to compensate every injured workman.

7 Under the stimulation of self-interest, national commissions, Governmental bureaus, employers' organizations and chambers of commerce began a serious study of the causes of accidents and a possible means of preventing them—the methods of minimizing the losses from accidents. Finally, there has resulted a full and free interchange of ideas through various engineering and industrial organizations and through the National Safety Council, which has made the work immensely more effective.

8 The effects of this work were another triumph for method when applied to the social and industrial question. The work had previously been done in a haphazard and unscientific way, and was, in consequence, very unsatisfactory.

9 It should be further stated that it was commonly assumed that the rapid increase in industrial accidents was inevitably caused by the growing use of power driven machinery. This view has been shown to be far from the truth, because the majority of industrial accidents are not caused so much by machinery as by carelessness in its use—carelessness proven to be very largely preventable through extensive and thorough educational methods. The employe "took

chances" and the employer expressed his attitude on the subject by the slogan "Get out the work." The man who received his approbation was the man who "did things" and accidents were accepted as part of the price of success. This fallacy had to be very obvious before it was exploded.

10 This attitude of mind was reflected in the state of the law, which held that while the employer was bound to carry on his work in a reasonably safe manner, in a reasonably safe place, with reasonably safe tools and reasonably safe workmen, reasonable safety did not require him to take undue precautions, the hazard being put on the workmen through the three archaic common law defenses, viz., the fellow servant doctrine, acceptance of risk, and contributory negligence. In other words, the employer was not bound to abolish the danger spots even if they were vital to production, and did not place him outside of the "reasonableness" test. The common law held that the employer was not under any obligation to provide guards, unless they were required by a special statute. The real cause of the acceptance and growth of accidents was the general and philosophical acceptance on the part of both employers and workmen that nearly all accidents were due to the "human equation" and an inevitable part of the business, and that it was impracticable to guard against or prevent them. There was no true appreciation of the causes of accidents, and, therefore, no logical plan put forward as a remedy.

11 The first step in accident prevention was to disprove this doctrine and to show that accidents were unnecessary. This could only be done by serious study of the causes of accidents, and a willingness to invest time, money and brains in remedial methods. Sentiment may make men anxious to abolish accidents, but "self interest" spurs them on to collect and classify accident statistics, to hire competent engineers to devise methods of accident prevention, and to spend a great deal of money in making these methods productive. That is why effective safety work coincided with the introduction of Workmen's Compensation Laws. Credit should be given, however, to the concurrent growth of a larger belief in the doctrine of fellowship; and feeling of responsibility for the welfare of their men on the part of a large number of employers has been a powerful adjunct to the force of necessity in bringing activity to a focus.

12 It was also feared that if machinery could be safeguarded, or changed in character, or operated at lower speed, economic

operation of industry would be interfered with. This, too, has happily been proven untrue, for good safety work has been found the partner and part of efficiency; for, being simply the study of the right and orderly way to do things, it has aided shop methods and economy in a very marked degree. Safety work has, therefore, evolved as a great constructive economic force in industry, as well as a social and humanitarian one.

13 What then is the present attitude of employers toward effective methods for reducing industrial accidents, and paying for every personal injury suffered by workmen under Workmen's Compensation Laws?

14 The ultra-conservative type of employer is not yet awake to the modern methods of accomplishing results. He is often the man who has come up from the ranks, or the one who has inherited the established customs of his forefathers—the employer who feels that he only gets as much work out of a man as he can force out of him—that industry is a struggle between two men as to who will get the "best of the bargain." This type of employer has not taken kindly to the suggestion of his responsibility for safe methods of doing work in his shop. His attitude is that he owns the business,—that he has the exclusive privilege of directing it, and the suggestion that his foremen or workmen be asked to tell him where the dangers of the business may lie is obnoxious to him and wrong in principle. He is afraid that if his workmen are organized into accident prevention committees this will encourage the union idea among his men. The Workmen's Compensation Law forces this type of employer to study more carefully the causes of accidents suffered by his employes, and because he has to pay the cost of the accidents he may be converted and study the results secured by other employers through more humane measures. He will first have to learn to do business by the coöperative method, and if he wants to eliminate the excessive cost of accidents he must realize that his employes should be recognized as having a part in the organized plan to eliminate accidents. He must be willing to educate his workmen in safe practices—the right way of doing things—to study their operation of the work in his shop from the standpoint of safety, and make his workmen understand that "Safety must be the first consideration."

15 A second group of employers may be said to occupy the "middle ground." These are the employers owning modernly constructed plants, well ventilated and illuminated and clean and orderly—and who take a moderate interest in the safeguarding of

machinery, if it does not cost too much money—but who do not believe in "new-fangled notions." This employer is shocked by the horrible details of an accident, but considers himself to be a model of propriety in his own personal relationships. He would resent any insinuation that he was not vitally interested in the welfare of his employes. At the same time he is rather slow to accept new conditions—he believes in all the rights of the "boss." He is averse to publicity methods of accidents; he dislikes criticism or any intimation that there may be anything wrong with the operation of his plant. A large percentage of the owners of small plants may be included in this category. This applies particularly to employers whose businesses are not generally considered to be of an especially dangerous character. Employers in this group have not actively opposed Workmen's Compensation Laws but are apt to support associations who boost propaganda against radical legislation.

16 The attitude of the third group may be termed that of the progressive type—the pioneers in the accident prevention movement. The large employers of labor, the leaders in industrial life, and the employer who is himself working earnestly for the welfare of the race, are among those who have taken the lead in making an effort to solve the accident prevention problem. They are in favor of a fair and equitable Workmen's Compensation Law. They have equipped their plants with sanitary toilets, locker facilities, modern washrooms and recreation centers, and do everything within their means to make their work places attractive and comfortable.

17 This class of employers asks safety engineers to review plans for new shop buildings from the standpoint of safety and comfortable working conditions. Tacitly they confess to their employes that they may not have done their full duty towards them in the past in providing safe working conditions, and urge coöperation by encouraging the men to work with them in establishing safe working practices to eliminate as many of the hazards of the industry as may be possible. They organize their workmen into committees for the purpose of getting their suggestions in making conditions safe, and carry on a vigorous educational campaign to teach the workmen the real causes and remedies for accidents. These are the employers who reduce accidents in their plants from 25 to 85 per cent, and whose efficiency account shows savings up to many million dollars per year. These employers unanimously declare that results are securable from safety work commensurate with the time and effort

put into the task. One of them has recently stated that his investment for safety has brought a tangible return of 80 per cent.

18 The increase in insurance rates for protection from liability under workmen's compensation has brought about a revision in the methods of establishing premium rates, and now, through the (so-called) "merit rating system," this group of employers realizes the benefits of making their plants safe.

19 To summarize, from the anciently accepted attitude of indifferent or ignorant acquiescence in accidents has come

I An awakening, caused by
  - *a* publicity that has compelled a realization of the inhumanity and fearful economic waste of accidents
  - *b* the discovery and growing conviction that accidents are preventable.

II The employer has proceeded to prevent accidents
  - *a* impelled by his humanitarian impulses
  - *b* spurred by compensation laws that put the burden and responsibility on the industry and employer, and sweeping away the old outgrown customs and laws

III The result has been
  - *a* a great and gratifying reduction of accidents through mechanical safeguarding and education of workmen
  - *b* the discovery that good safety work has been a big money saver over the old accident conditions
  - *c* the discovery of the great constructive value of safety work, it proving to be a study of the proper way to accomplish manufacturing results and, therefore, a great aid to industrial efficiency and economy
  - *d* the bringing of employer and employe together on a common ground of mutual interest, thus producing a fellowship and good feeling and coöperation in industry that nothing else could have accomplished.

20 Aside from the attitude of the three groups of employers referred to, it should be stated in all fairness that when the history of the safety movement is written a great deal of credit will be given to the employer. It has been due to his study and interest that the cause of accident prevention has grown to immense proportions in an amazingly short time.

21 Many illustrations could be cited to indicate the rapid and widespread development of the safety movement through the establishment of new businesses to provide such safety equipment as

safety spectacles, safety devices, articles of clothing, etc., and the testimony of these salespeople is that their customers are weli satisfied with the results secured through the use of such preventative devices. Generally speaking, however, the safeguard devised in the shop or factory where it is to be utilized is the most effective educational feature of safety work.

22 No better evidence could be given of the attitude of employers toward real interest in accident prevention than the development and growth of the National Safety Council. The council was organized by a group of representative employers as the result of a safety meeting held in Milwaukee in 1912. The headquarters of the council were not opened until October 1913, but within a period of two years this organization has secured a membership of over 1600 employers, representing 150 industries, included within which are 5500 representatives located in every State of the Union, except two, and in many foreign countries. These employers have voluntarily joined the council for the purpose of formulating information to promote the cause of accident prevention. Through the dissemination of the collective experience of the members of the council of various methods adopted to eliminate accidents, a great deal of light has been thrown upon these problems. The National Safety Council has organized twenty-three active local councils in various parts of the country from San Francisco to Boston. The Fourth Annual Congress was attended by over 1500 employers, or their representatives, and for attention and interest this meeting has rarely been excelled.

23 Some employers predict the most far-reaching effect from the safety organization plan in bringing the employer and employe together into closer relationship, by obliterating the class feeling which has kept employer and employe apart. When both sit down at a safety meeting to discuss a problem of common interest, there is apt to grow up a better understanding of the aims and purposes of each party. If the employe has imagined in the past that the employer's indifference has been the cause of accidents, he soon learns that the remedy lies principally within himself.

24 While it is realized that many employers are not yet in sympathy with many of the aspects of the accident prevention problem, and that the Workmen's Compensation Laws are defective in many ways, it is believed that both employer and employe will profit by the new status of the work and will gain a new and better view of their industrial obligations.

DISCUSSION

John Price Jackson. I believe that this paper is a clear review of the attitude of the employer in Pennsylvania toward safety, and the work that is being done. From the point of view of safety legislation, the situation today is that practically every State in the Union now has something in the nature of what may be called an Industrial Board. In many cases there are obligatory laws which are enforced by these boards and the accompanying inspection forces, but many of these laws are often imperfect and without uniformity over the whole country. Suppose Pennsylvania, through its Board, calls for a certain kind of guarded lathes, and Illinois calls for another kind, and Indiana for a third kind, then undoubtedly waste and useless expense is involved. There are a great many kinds of machinery where a great many types of guards can be used, and therefore, where the Boards can make a great differentiation and cause unnecessary lack of uniformity by specific requirement.

If such a body as this Society, through its Committee on Protection of Industrial Workers, can put on record a reasonably sound practice for providing all of our industrial machinery with safety devices, then the Boards, instead of making haphazard rules and regulations, may be guided to make rules according to this practice.

We have another group of laws growing up—compensation laws—which have been adopted in nearly every State. Under these laws an employer who saves human loss of life and capacity as carefully as he now saves fuel loss, is actually going to save many dollars in his outlay, and also in his insurance rate.

There is a very material opportunity for financial saving by having available acceptable practices for the installation of safeguards and safety devices.

The subject of safety organizations and safety methods, though more a matter of management than design, is of even greater importance than that of safeguards. It should, therefore, have the careful consideration of this Society in order that the best methods may be outlined for the development of carefulness on the part of all connected with an industry.

Summarizing, my thought is that this Society, with the help of its whole membership, should put on record conclusions and opinions which will be the directive force in the safety movement of the Nation, and will keep this movement in the right direction, both as to the laws that may be passed in the several States, and as to the methods that may be used by the manufacturers themselves, and by the Industrial Boards and safety and inspection officers.

CHARLES WHITING BAKER. In the general discussion of safety, a fact which is often ignored, is that safety is always a relative term, and that there is no such thing as absolute safety.

We cannot prevent all accidents, but what we can do is to make reasonable requirements that will raise our standards of safety. The question then arises as to where shall we draw the line. This is a matter which can only be determined by the engineer from experience and expert knowledge.

There is another serious side of this subject, and that is the question of fixing responsibility for accidents. As engineers, we should always remember to have some charity for the men who are responsible. It very often happens that the accidents which occur may have come from some cause for which any of us might have been responsible. Ignorance of some of the simple principles of physics on the part of some of the wisest is sometimes responsible for serious accidents.

LEW RUSSELL PALMER.[1] The engineer is playing a greater part every day in the work of accident prevention. Engineers, as a rule, stand for efficiency. There is nothing that indicates inefficiency in such a marked degree as a useless waste in human energy. Plant managements woke up several years ago to the fact that they could produce double the amount of material with the same number of men if they installed proper appliances and safeguards. Safety work is another application of the efficiency idea.

Mr. Cameron states that coöperation is the keynote of the work, and we need coöperation, both from safety organizations and State and National organizations of engineers.

[1]Ch. Inspr. Pa. Dept. Labor and Industry, Wilkes-Barre, Pa.

No. 1513

INDUSTRIAL SAFETY AND PRINCIPLES OF MANAGEMENT

By W. P. Barba, Philadelphia, Pa.
Member of the Society

The matter of accident prevention has come to be considered as a point vital to the success of any business—through the operation of reasons, economic, humanitarian and sociological. Only a few years ago, say ten years, little was heard in this country of more than desultory efforts to minimize the waste of human effort through accidents. Now the slogan *Safety First* has been given widespread currency, and has lately been improved by coupling with it another, without which the first is really of little value, hence we now hear *Safety First—Safety Always.*

2 It requires no detailed presentation to make manifest the waste to the world of productive power through the suspension from daily labor of men injured therein, of the consumption of materials without the corresponding production which normally would result therefrom, of the real suffering—both of the injured man and those dependent upon him, or of the investment laid aside and idle with every man injured. As part of a study of the principles of management (which should accompany the technical education of every man called to work amongst his fellows) much too little attention is given to the subject of methods of employing labor, both trained and untrained, and too little weight is laid upon the cost of training *any* new employee into the duties for which he is hired. Few realize that each man hired into a plant is the subject of investment for quite a time before he is sufficiently trained to the work he is to perform as to return a profit. It is probable that a study of the subject would produce a figure of $150.00 average of investment in each man before he becomes productive.

3 So when true principles of management are but little under-

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Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Philadelphia Section.

stood and rarely taught, it is customary to witness many discharges as corrective of offenses which are perhaps trivial at best, but which discharges in any case only act as correctives for the benefit of the employer with whom the discharged workman may next engage for employment. A better way, for instance, is to call up the man, convict him to his own satisfaction and yours, of the wrongdoing in question, and then and there apply the punishment which should correct the offense, using means which will inure to the benefit of the employee as well as the employer. This can be effected by lay-off or fine; discharge should be rare, and only for major offenses. Fines should be used most often in preference to either of the other more usual means. Fines should be laid with great care for the proper amount needed to effect the corrective, and a relatively small sum usually is sufficient. In no case should fines accrue to the benefit of the company, this having a very disturbing effect upon the value of the corrective, but fines in all cases, both for spoiled work and for discipline, should be converted into a fund for the benefit of the whole body of workers. Some mutual benefit plan, such as is now happily established in most up-to-date works, should receive the results of all fines. The result is then doubly beneficial, and has a balmy effect which removes all soreness and sense of *personal* injury from the transaction.

4 Referring again to the loss of investment from removal of trained men from production by discharge, let us also look at this loss through *physical* injury, and especially those arising from preventable causes. It is, of course, plain that the investment loss through injury can become a permanent loss of producing power, the loss of investment causing the employer to suffer, and the loss of producing power causing the injured man to suffer permanently, as well as through the physical pain he bears, meanwhile; his producing power and income reduced or gone, his expenses continue even on a reduced scale, his mental condition is much disturbed and he wonders what can be done, and so the rapid but all too slow growth of the safety movement, the liability and compensation laws, the preventive measures, but most powerful of all, and as yet but little developed, the education of every person engaged or even interested in industry, to the end that all preventives be used both *without* and *within* the person of the operator subject to injury.

5 Figures compiled during a term of years point most strongly to the lack of proper education as the prime cause of injury—loss

of productive power, and all the attendants of this condition. Hence, industrial safety and sound principles of management are without doubt most closely bound together, and cannot with success be considered apart. Commissioner John Price Jackson, of the Department of Labor and Industry of Pennsylvania, has shown in dollars and cents just what the large number of accidents that are occurring in the industries of the State, are costing the workers. He has not figured what they cost the employers.

6 During 1914 accidents cost employers in a score or more of the larger industries in Pennsylvania the sum of $1,048,503.96, which total is computed on the daily wage. There were 38,126 men thrown out of work and each of these men lost on the average $27.50. The accidents from which Commissioner Jackson has computed the figures do not include those reported to the State Department of Mines or the Public Service Commission. The report shows that about one man in every 28 lost time because of accidents, as there were 1,086,508 employees in the industries from which statistics were gathered. The average daily wage was $2.45 and the total number of working days lost was 426,824.

7 This brief report gives one an idea of the extent to which accident prevention may go, the aggregate for this State only being so large as to appall one, and still does not include mining risks that are notoriously great. Indeed, most of the opposition which was disclosed in an organized way to the enactment of a workmen's compensation act has been from the smaller operators of mines, whose whole capital could be annihilated by the liabilities arising from one accident on a small property.

8 From the operation of these causes, motives and products alone, there has within the last eight years, come into being a general safety movement, looking more toward the possibilities of prevention than ever before. Out of all the upward striving of humanity toward better things and better conditions comes a new principle: namely, that a trade should bear the charges and costs of its casualties, and from this, the employers' liability and workmen's compensation movements have gained much force. Rarely has there come before the public a movement involving so-called capital and labor, but in reality employer and employed, which has met with such complete support from both sides. But there *are* no "sides"—the interests of *all* employed in the world's productive work are entirely identical. To enlarge upon this thought, for

with operating a large manufacturing plant has as his first pra tical concern, the securing and retaining, as well as maintaining in good efficient condition, a well satisfied body of workmen, contente well paid, and their physical condition well in hand.

10 After this the managing executive may take up his financ ing, his raw materials, manufacturing methods, selling organizatio etc., but no manufacturer can possibly succeed without this satis fied body of operating workmen—fellow employees as just outline —to secure and maintain this condition as a prerequisite to an further success.

11 The economic as well as the social side of this problem mus be just as carefully looked after, now that Pennsylvania has a work men's compensation act, the twenty-fourth in the United States which country followed quite a long way behind both Germany an England. No longer will industrial safety be thought a fad, n longer will the courts be filled with cases crying for just relief, n more will the cheap lawyer haunt the hospitals to prey vulture-lik upon the unfortunate. Each trade is to bear its share of the cos of its casualties, and the ultimate consumer will pay the bill. Al this is exactly as it should be, the only danger lying in the chance a real danger too, of the whole game being absorbed into some form of political spoils, which God forbid.

12 Much is being said and printed about foreign trade, foreig competition, foreign methods, and many have objected that th securing of this foreign trade in normal times was difficult, due t the difference in labor conditions, rates of pay, etc. It is ver

his plant *safe;* and the penalty for failure to achieve safety of equipment will be very great. There are, however, other features in the case: present and proposed laws provide greater compensation for a man with a family than a lone bachelor without dependents, some laws providing compensation for the grandfather of a man injured, when dependent. The natural result will be to select for employment men who offer the lowest risk, both as to compensation penalties and as to physical condition of the applicant, which condition exactly reflects chances of injury, and consequently, the rate of risk. The compulsory insurance laws of Great Britain, placed in operation in 1912, immediately resulted in selection of employees whose rate of risk was the lowest, and thousands of perfectly good workpeople were turned adrift for this reason.

14 This is one of the consequential results our legislators and agitators should consider most carefully. There will inevitably result, a careful selection of the best insurance risks, and defectives of all kinds will find it most difficult to obtain and retain employment. A defective in this sense may be a man physically unfit, incipient hernia, a tendency to joint dislocation, etc. He may be defective in the sense that he just cannot keep out of trouble, and this kind of a man soon cannot keep himself in a job. He will be *selected* to be always out of a job, until he learns to *think* enough to keep himself from unnecessary injury. Study of the figures showing frequency of accident to the individual no matter where working, inevitably searches out such men, and then the employer must move the man, either to another job or else *out*, both ways meaning a loss of investment, production and earning power. Again education, both of the man and his employer, most necessarily his immediate foreman, is needed and just how best to supply the lack is a task worthy the best mind in your employ.

15 The works with which the author is intimately associated, has for years made a careful physical examination of each man offering for employment, having in mind his condition with reference to health, present and potential, his record as to previous injuries and their results, his eyesight, hearing and whatever condition will affect his value both as an individual and as affecting his fellow workmen.

16 At one works there were offered for employment, practically hired, then turned over to the physicians for this examination 2,569 men during the year 1913; of these 391, or 20 per cent were rejected

for various reasons. Of course, a man may pass inspection upon employment, and then find his health go down. He may become the subject of some form of communicable disease. The answer for this is the growing need for periodical examination of each individual worker, and his elimination when his risk becomes too great. This periodical examination of employees has been undertaken to much too small an extent, and needs attention and developing. The benefits are numerous, the first and chief being the aid and assistance to full recovery, and checking in advance of disease which a man unconsciously fights off until he drops—mastered. Beaten by exposure to a blizzard or heavy storm when in a reduced condition, many a good man is lost, who, by a regular, even though cursory, examination might have been saved through timely catching, checking and conquering, through advice and aid, of an ailment creeping up on a man almost without his knowledge of anything going wrong. In this connection arises the demand, a just one, for some form of insurance, preferably mutual, and upon a sliding scale, according to the risk offered by each man. This is but one of the many problems each executive is going to face in the future handling of employees' matters.

17 When a man is hired, he should be taken to his foreman for careful instruction in his duties, its dangers and their safeguards pointed out, and every effort made to prevent the new employee from becoming worse than a normal risk. As one concrete illustration of this, in a certain large works much fuel oil is used for heating purposes, and a comprehensive system of storage and distribution is installed. Leakage is impossible to entirely prevent, and numerous explosions and small fires have occurred. It is necessary at times to descend into the pits in which the apparatus is placed, and the explosive fumes of oil and air are frequently present; steam pipes (not air—because of the danger of air) are rigged in all such places and at stated intervals the steam is sent through these chambers entirely and safely displacing all noxious gases.

18 Again, the burners for fuel oil are merely a combination of jets and valves—some of the jets embodying the injector principle. Each man who is called upon to work with this apparatus is taken into the bureau of Safety First by his foreman, and with the safety engineer (a high priced executive) is taken through a course in his particular oil burning apparatus. The collected unit is there, it is

disassembled and reassembled by the man, and a partly cut-out section is shown, so that all the functions of the system are fully understood by the workman. Since the introduction of this system of instruction, the number of burns and fires from ignorant handling has been very much reduced. It is impossible, however, to eliminate entirely the accidents from careless handling. Here, as elsewhere, the adage "Familiarity breeds contempt" is true, and it would be easy to multiply concrete illustrations of this statement.

19 In 1907, in a nearby large works, the safety movement was given a real start, by the appointment of a safety officer, whose whole duty was to report hazardous conditions of plant and equipment and see that the conditions were corrected. It has lately been more generally recognized as the function of the factory inspector of the state to keep every manufacturing establishment in a safe condition, so far as equipment, etc., is concerned. The present incumbent in Pennsylvania, Dr. John Price Jackson, has given real value to this function for the first time, but even he, up-to-date and progressive as he is, recognizes that the correct way is to make each factory safe automatically by having liability laws which properly penalize the lax employers.

20 The attempt of 1907 to effect safety of equipment by the appointment of a safety officer and his staff, in the works mentioned, was soon superseded by a plan to make the worker himself feel his share of the responsibility, and a committee of 70 men was chosen from among the employees, there being upward of 5,000 employees. These men were always on duty while regularly working and seven of them were chosen each week and with the safety engineer gave up a whole day in the company's time and pay to actual examination of conditions of plant and equipment. Their recommendations were given priority by heads of departments and the works were soon found in such good physical shape that the committees had little or nothing more to report.

21 The number of accidents was, of course, reduced, but not to a point which was thought commensurate with the efforts put forth. During all of this time (five years) careful study and analysis were given the daily accident reports, with the result that they soon segregated into three groups:

1 Hazards of occupation
2 Hazards due to faulty equipment
3 Hazards due to personal carelessness and disregard of safety appliances.

22 These regularly occurred in almost unchanging proportions, even though the sum total went steadily downward:

> The first class—Hazards of occupation, 24 per cent.
> The second—Faulty equipment, 3 per cent.
> The third—Carelessness and neglect, 73 per cent.

23 This large proportion, 73 per cent, is purely a result of the operation of the personal equation, and, at once suggested itself as the point of attack.

24 To meet this a total change of programme was inaugurated. The plant was divided into seventeen distinct units or geographical districts and a committee of three was appointed in each for a term of two months. A datum line for each district was established from history which showed the frequency of accidents in each district. The figures were worked out in units per one hundred men employed for the period of two months, thus affording easy comparison.

25 The task set was for each district to equal or beat its previous record. No district was set against another, its record being wholly within itself. This is a vital point. Each sixty days' record is merged into the previous total and thus a new record automatically set. For a committee which equalled or beat its district record during the committee's sixty-day term, there was established for each man a cash prize of a ten dollar gold piece, or $30.00 for each winning district each sixty days (incidentally the amount thus paid during the year 1914 was $3,200.00, and no sum was ever more cheerfully paid out). In addition, the committee which made the greatest improvement upon its own record was, each period, granted a double prize, or $20.00 to each of the three men.

26 The experience of the first year was that out of the seventeen districts there were paid prizes all the way from four up to sixteen districts, the four being midway in the year, and the treatment applied by the management when this low score occurred brought the score right up, so that at the last period fourteen sets of prizes were paid. The personnel of these committees is changed each period so that the experience gained is accumulated by a larger number of men.

27 The treatment in this case was simply for the general manager to talk to the assembly of the men later referred to, and point out, from knowledge of the accidents occurring during the period when low scores were made, how greater vigilance, less laxness, more attention to the men seen to take hazards carelessly, would result

in better scores, more gold pieces, less suffering, to risk of which each of the whole number of employees is subject, whenever vigilance in these particulars is relaxed.

28 There is also a collateral advantage which is a large part of the value of this new scheme. The management hires a large and convenient hall for a meeting place, provides cigars and light refreshments, invites all the men the hall will contain (about 400) and makes a public occasion of the presenting of the prizes. A free discussion of methods and experience is had, and from 8 o'clock to 9:30 an evening is spent which is most profitable in every way. Stimulation is given the safety movement, the managers are all there, and a great feeling of the community of interest of all concerned is engendered. The spirit of full coöperation is established and fostered, a better and closer acquaintance is had on all sides, and the whole effect is most beneficial.

29 The net result in figures is curious and interesting. By an accident is meant the state requirement or definition—that a man is away from his work more than two days. The number of accidents during the year just closed was reduced by 59 per cent, the three above classes being represented by 26.8 per cent, 2.23 per cent and 71 per cent, clearly showing that there is still more work to be done in fully bringing home to the individual his personal share in the responsibility for his injury.

30 This responsibility is going to be more closely brought home under compensation acts, since it will mean the elimination of men who are thus injured through their own fault too frequently; these men will be compelled to seek other employment. It then becomes a nice point of judgment for the management to determine whether its investment in such a man, i.e., his trained capacity for his work, is a sufficient offset against his increased risk, due to his propensity for acquiring injuries to an undue degree. Some of the compensation laws refuse a man any payment if it be shown that he was injured by *his own act* for the purpose of going on the benefit list. This provision and the one denying benefit for the first *fourteen days* are about the only safeguards the employer has against unjust claims for payments.

31 Reverting to the system suggested of avoiding discharge losses through conviction and fines, the proper channel for restoring fines upon delinquents to circulation through the whole mass of employees is—the recognized need of caring in some way for the

reliable employees unavoidably injured slightly, and returning to work within the 14 days exemption period provided by recent compensation laws.

32 A fund could be provided, augmented by the company, especially so in the case of larger employers, this fund to be administered by a mixed board of employees, and a small measure of financial relief afforded the unfortunates who recover within 14 days. This period, designed by framers of the acts to prevent malingering, does not altogether effect this, and meanwhile works a real hardship on many worthy men whose needs are such that the uncompensated loss of *any* days becomes a matter of concern.

33 If ever there was a subject fit for Federal instead of State legislation, it is this one of liability and compensation. Fifty-eight states working each alone may produce such wide diversity of legislation as really to put neighboring states into competition for both manufacturers and for workmen. When John P. Neill was Federal Commissioner of Labor he worked assiduously to secure uniform state legislation on this subject because there were statutory difficulties in the way of a Federal act. This leaves it up to the several states where too often a subject of this magnitude is taken up by untrained and uninformed legislators who can, quite possibly, be swept into ill-considered action by a wave of hysterical outcry from the newspapers, professional labor leaders, and publicists who treat a situation academically and without close knowledge of the problem.

34 With especial reference to the Pennsylvania law just enacted, it is fine to recall how this law was prepared. Begun under Governor Tener by a board of broad expert business men, thrown out by the legislature of 1913, brought up again by a Governor fearless of criticism, worked out modernly upon the basis of the original commission's draft, the Pennsylvania law is, generally speaking, satisfactory to all groups, and there is in it opportunity to coöperate fully, to protect those dependent upon their labor for daily bread, automatic incentive to clean, healthful surroundings, care in safeguarding equipment, and all concerned are compelled to do their utmost to achieve the blessed result desired.

35 Any employer, or rather fellow employee, who shall disregard the plain common sense demands for a legitimate, well considered scheme for automatic compensation for every injury not wilfully incurred, is not alive to his business, to his duties, to his

men or to his stockholders, nor to his duties to the progress of humanity at large.

36 It is the author's firm conviction that the so-called *industrial unrest* is wholly preventable, is due chiefly to lack of understanding of the problem—lack of patient working in full coöperation with all concerned, and the result of following sound principles with policies based thereon is certain to prevent unrest such as has been all too frequent from just such causes.

37 The years in the immediate future will be largely occupied with the care and handling of just such problems as are here presented, and if anything said here gives a line to take hold of and follow, the author will be more than glad to have had the opportunity to discuss a matter so large as that of the subject presented. In pointing out the dangers of haste in enacting such legislation, all right-thinking people are urged to work for the passage of such laws as will compel the lax, careless, or unwilling employers to secure the coöperation toward safety of workers whose livelihood, and lives, are in the hands of those who pay wages, and as before mentioned should find it a cheerful duty to work toward the securing through coöperative effort, of the maximum of safety, comfort and happiness among all grades of employees. Only thus will the work of the world be furthered. Only thus will there be removed from among us strife, discord, class distinction, unions and non-union, and there will surely come into industrial life, the big rewards which result from careful thought along lines which go to promote full coöperation among those called to do the work of the world.

No. 1514

THE HEAT INSULATING PROPERTIES OF COMMERCIAL STEAM PIPE COVERINGS

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INTRODUCTION

In meeting the necessity for effectively insulating heated surfaces against the loss of heat, the question arises as to what material shall be used and, what is of even greater importance, what thickness of material shall be used for greatest economy.

2 There are on the market at the present time a large number of commercial pipe coverings made up ready for use. It is customary to use one of these forms and for that reason, the investigations described in this paper are confined to such commercial coverings. However, as will be shown in the mathematical treatment of the subject, the results of the experiments confirm theoretical laws which may be used to extend the scope of the work so as to include the entire field of heat insulation.

3 An enormous amount of effort has been expended in attempts to determine accurately the savings effected by the use of non-conducting coverings on steam pipes. But, even with the results of all these investigations available, little reliable information is on hand regarding the efficiencies of pipe coverings in commercial use at the present time.

4 The majority of the tests, the results of which are now used, were made from 15 to 20 years ago, and the investigators made no attempt to make their results applicable to conditions other than those prevailing at that time; so that errors up to 100 per cent in extreme cases would be made in applying their conclusions to present day conditions. Age alone is no discredit to any work, but time alters conditions. One of the first principles of scientific research is to keep all factors but one constant and vary that one within wide

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

limits, noting its effect on the results, then applying this process to each of the variables in succession. The failure to do this accounts for the inadequacy of the work of former investigators in the field under consideration.

5 Meritorious work on the testing of heat insulating materials has been done by Prof. J. M. Ordway,[1] C. L. Norton,[2] G. M. Brill,[3] Geo. H. Barrus,[4] Prof. D. S. Jacobus,[5] H. G. Stott,[6] Chas. Eberle,[7] and many others, but in general their results apply at only one temperature, or at most two, and for one or two thicknesses and are not sufficient to determine the laws which govern the flow of heat at different temperatures and for different thicknesses. Therefore, this work fails where an attempt is made to apply the results to modern conditions involving high superheat and thicker coverings.

6 The work which this paper describes has been carried on for a period of about two years at the University of Wisconsin and every effort has been made to secure accurate and consistent results. The effect on heat losses of varying the temperature difference between pipe surface and air between limits of 0 and 500 deg. fahr. has been thoroughly investigated and the conclusions reached will be fully explained. Different thicknesses of material from 0 up to 3 in. were tested and the laws confirmed by the results of these tests permit of their application to any thickness whatsoever. The drop in temperature from steam in a pipe to the inner and outer surfaces of the pipe wall under various conditions has been accurately determined. Another new fact brought out was that the loss from any covered pipe is a function of the temperature difference between the surface of the covering and the surrounding air; and this function is the same for all coverings having the same character of surface regardless of what the other properties of the covering may be, since the effects, if any, of these properties appear in the temperature difference. The value of the function has been determined for canvas covered surfaces, and a complete explanation of its significance is here included.

[1]Trans. Am. Soc. M. E., vol. 5, p. 73; vol. 5, p. 212; vol. 6, p. 168.

[2]Trans. Am. Soc. M. E., vol. 19, p. 729.

[3]Trans. Am. Soc. M. E., vol. 16, p. 827.

[4]Trans. Am. Soc. M. E., vol. 23, p. 791.

[5]Stevens Institute Indicator, vol. 19, p. 12.

[6]Power, December, 1902, p. 33.

[7]Mit. über Forschungs-Arbeiten auf dem Geb. des Ing., heft 78.

REVIEW OF FORMER INVESTIGATIONS

7 The first extensive investigation of materials used for the purpose of insulating steam pipes against the loss of heat was made by Prof. J. M. Ordway, and the results were published in three papers before The American Society of Mechanical Engineers.[1] The method of this investigation consisted in covering a steam pipe with a given thickness of the material to be tested, and then surrounding this with a calorimeter made up of a double cylinder with the space between filled with water. The amount of heat passing through the covering was calculated from the weight of water in the calorimeter and its rise in temperature in a given time. In Professor Ordway's third paper, a comparison was drawn between results obtained by this method and those from an attempt made to determine the loss of heat through the covering by measuring the amount of condensation in the pipe under certain fixed conditions of test. Even at that early day the conclusions were decidedly against the condensation tests, and it was shown that such tests could not be depended upon for highly accurate results.

8 The work of Prof. Ordway is of interest principally on account of the great variety of materials tested—some fifty-five kinds in all. However, many of the materials were not useful as pipe coverings, as for example common salt, cotton batting, anthracite coal, geese feathers, etc.

9 The principal objection to Professor Ordway's method is that tests were not made with the covering under the conditions that would prevail if it were on the pipe without the surrounding calorimeter. It would be a rare occurrence, indeed, if the temperature of the air adjacent to the covering were the same with Ordway's calorimeter in place as when the pipe was in the open air. Since the amount of heat lost is a function of the *difference* of temperature between the steam and air, and not of the steam temperature alone, a serious error might be introduced here. A further and equally important source of error lies in the fact that the calorimeter itself serves as an insulator to a limited extent. Anything that resists the passage of heat decreases the amount that would be lost from the pipe, and the metal of the calorimeter and—what is more important—the enclosed air space would have quite an insulating effect. A minor error might be introduced by the transfer of heat from the calorimeter to the air and vice-versa.

[1]Trans. Am. Soc. M. E., vol. 5, p. 73; vol. 5, p. 212; vol. 6, p. 168.

10 More tests of heat insulating materials have been made by the "condensation" method than by any other. That is to say, the covered pipes were filled with steam and the heat lost was estimated from the amount of steam condensed. The difficulties in the way of getting accurate results from such tests are at once apparent. The amount of heat represented by the condensation of a single pound of steam is so large that, in order to get reasonably accurate results, very long lengths of pipe must be used to increase the amount of condensation to a measurable quantity. This introduces differences in couplings, bends, etc., that are hard to correct for, because the losses per unit area at these points are rarely the same as for the rest of the pipe. It is true that such fittings are found on every installation of steam piping, but not necessarily in the same proportion as on the test apparatus. Therefore, it is better to consider the covering of the straight sections and of the fittings separately. A large source of error is the fact that the quality of the steam is a quantity hard to determine with great accuracy, owing to the difficulty of getting an absolutely correct sample. If dead steam is used in the pipes, then only one quality need be measured, namely, that of the entering live steam. In this case it is likely that air and condensate will collect at the low points and seriously affect the radiation from such parts. If a current of steam is maintained, the quality must be measured at exit also, and, in addition, the amount of steam circulated must be determined. The collecting and weighing of the steam condensed in the pipe is neither accurate nor convenient. The arrangement of the ends of the pipe and of the collecting apparatus so that no heat can be lost from them is not possible by the use merely of insulating materials. The best proof of the inaccuracy of condensation methods is that duplicate tests by the same experimenter check only approximately, and the average of a very long series of tests would be necessary for a really accurate result.

11 An interesting method, and a departure from previous methods, was that of Prof. C. L. Norton.[1] An electric heater consisting of coils of resistance wire was placed in a pipe filled with oil, and the pipe was then covered with an insulating material. Electric current was passed through the coils and the amount of energy required to hold the temperature of the pipe constant was ascertained. This method is capable of giving very accurate results, but as employed by Professor Norton it was not well adapted to commercial

[1]Trans. Am. Soc. M. E., vol. 19, p. 729.

coverings on account of the necessity of covering the ends with the same material as that on the main body of the pipe. Other causes of inaccuracy were the extremely small sizes of pipe used and the fact that the temperature was measured at the upper end by means of a mercury thermometer. The temperature of the oil at the top of a vertical section could not, with accuracy, be taken as the temperature of the outside surface of the pipe. It would not even be the average temperature of the oil, and it will be shown later that there might be a considerable drop from the temperature of the oil to the temperature of the outside surface of the pipe. The results cited by Prof. R. C. Carpenter in discussing Professor Norton's paper pointed decidedly to this fact.

12 Perhaps the most accurate results of pipe covering tests yet published were those of H. G. Stott.[1] Several different kinds of coverings in lengths of 15 ft. each were placed on a long line of 2-in. pipe which was heated by passing an electric current through the metal of the pipe, the pipe itself serving as an electric heater. The heat given to each individual covering was calculated from the product of the current in the pipe and the voltage drop in an 11-ft. section under the given covering. The method of measuring the temperature was unique, since the section of pipe under the covering was made to serve not only as a resistance, but also as a resistance thermometer, the temperature being calculated from the *change* in resistance of the pipe.

13 As far as they went, Mr. Stott's tests were highly accurate, and practically the only objection that can be offered is the tremendous amount of current required. This would limit the size of the covering tested to practically the 2-in. size used by Mr. Stott, and while the results would be excellent on a comparative basis, there is no ground for the assumption that there is the same loss through a square foot of any 10-in. covering as through a square foot of 2-in. covering of the same thickness, when the ratio of pipe surface to covering surface is greatly different in the two cases.

FUNDAMENTAL METHODS OF TESTS

14 It has already been pointed out that the methods employed by Norton and Stott were open to criticism. However, the principles on which their methods were based seemed fundamentally correct, and therefore were adopted as a basis for this investigation. An entirely original scheme of performing the tests was adopted,

[1]Power, December, 1902, p. 33.

however, and by this means the points of criticism were eliminat

15 It was proposed to heat a section of covered pipe by means an electric heater made up of resistance coils immersed in oil insi the pipe, and to calculate the amount of heat lost through the cov ing by measuring the energy required to hold the outside metal the pipe at a constant known temperature. Under such conditio it is evident that just enough energy is being supplied to compens for the losses through the covering; otherwise the excess or ficiency of energy will cause the pipe to heat up or cool off as t case may be. This must be true for, according to the law of co

FIG. 1 APPARATUS USED FOR PIPE COVERING TESTS

servation of energy, all the energy entering must appear as he since none is transformed into any other form and none is lo Fig. 1 is a photograph of the apparatus used for the tests.

16 The objection will be raised that the results of the tests this method do not represent actual operating conditions, since actual practice the pipes contain steam and not hot oil. The co ditions of operation in practice are these: The covering is plac on a pipe containing steam at some practically constant pressu therefore the metal of the pipe has some definite constant te perature. The surrounding air may be either still or in motion, a it too has practically a constant temperature over a limited peric If the temperature of the pipe is higher than that of the air, or vic versa, there will be a flow of heat, the quantity of which will be

function of the difference of temperature and certain constants, one of which is the conductivity of the material. Now it is so simple as to be considered axiomatic that this flow of heat will be independent of what the pipe contains so long as the temperature difference between the pipe surface and the air remains the same. This may be demonstrated mathematically using the equation $Q = k\frac{\theta_1 - \theta_2}{x}. A\,t$ where Q = quantity of heat that will be conducted in time t through area A and thickness x of a material whose conductivity is k; temperatures of inner and outer surfaces being θ_1 and θ_2 respectively. The limiting value of the quantity $(\theta_1 - \theta_2) \div x$ or $\frac{d\,\theta}{d\,x}$ is called the *temperature gradient* at any point, and inspection of the equation shows that this is the only thing that can change the rate of heat flow $U = \frac{k\,.\,d\,\theta}{d\,x}$ through a material of constant conductivity, and the statement of operating conditions made the temperature gradient constant also. Theretore, in order to duplicate operating conditions for the covering it is only necessary to maintain a *constant* difference of temperature between its inner and outer surfaces; how this is done is immaterial.

17 In order to maintain the temperature of the pipe constant, however, heat must be supplied from some source if heat is passing out through the covering. If steam is the heating medium the temperature gradient *inside* the pipe, that is, from the steam to the surface of the pipe, will be less than for oil; for the material having the higher conductivity will require the lower temperature gradient for the delivery of a given amount of heat. But the different effects of oil and steam will cease here.

18 The accuracy of any conductivity test depends upon being able to maintain constant conditions, and this may be done by electrical means better than by any other. It is necessary, however, to have a very accurate means of knowing not only when the temperatures are constant, but also the correct values of such temperatures. For room temperatures, high grade mercury thermometers were considered satisfactory. For the pipe temperatures, however, after a great deal of experimenting with various arrangements, it was decided to use copper-constantan thermo-couples and the potentiometer method of measuring the e.m.f. of the couples. This method is one giving opportunity for the highest accuracy in high temperature measurements. The couples were calibrated by com-

paring with a standard thermometer at the boiling points of pure organic substances. Points were obtained up as high as 600 deg. fahr. and the results of the calibration appear in the form of a curve in Fig. 2. The instruments were sensitive to about 0.2 deg. fahr. change of temperature, which was well within the requirements of the case.

TEMPERATURE GRADIENT INVESTIGATION

19 In the foregoing the fact was brought out that the drop in temperature from steam to the outside surface of a pipe was probably different than from oil to the same point on account of the dif-

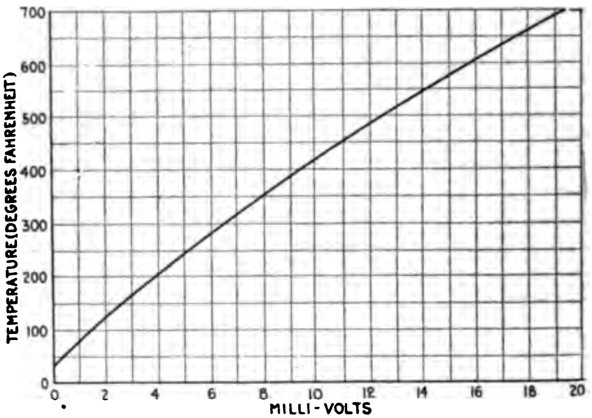


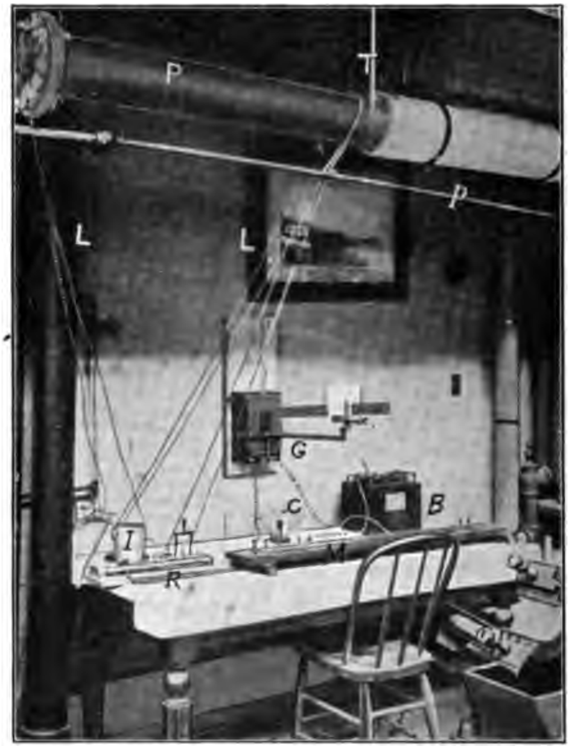
Fig. 2 Calibration Curve for Copper-Constantan Thermo-Couples

ferent conductivities of these substances. The method of test provided for the measuring of the temperature of the *outside* of the pipe and not of the oil, so that in order to make the results applicable directly to steam conditions it is only necessary to establish a relation between the temperature of steam in a pipe and that of the outer surface of the pipe wall. This difference of temperature has been entirely neglected by most investigators, being accurately measured by none and considered not to exist by most.

20 It was reasoned in the present case that in order to have heat flow from the steam to the pipe some difference of temperature was necessary, and since there must be such a difference, then it would be highly desirable to know just how great it actually was. With steam in the pipe, the temperatures of steam and of outside surface of pipe were measured very accurately by means of copper-constantan thermo-couples and potentiometer of the kind already

ntioned. However, in this case a much more sensitive galvano-
ter was used, so that a difference of temperature of 0.1 deg. fahr.
ld be detected.

21 The problem of carrying through the pipe the leads of the
iple that was to be in the steam and yet keeping the wires insu-
ed from the pipe and from each other was not a simple one. The
ice that finally proved to be satisfactory was made by screwing
-in. nipple into a ½-in. bushing in a reverse manner, passing



. 3 APPARATUS FOR INVESTIGATING TEMPERATURE GRADIENT IN THE WALL OF A STEAM PIPE

couple, each wire enclosed in a glass tube, through the nipple,
l filling the space that remained with bakelite. The plug
is formed was heated for several hours at about 175 deg. fahr.,
er which it was ready to be put into the pipe in which a ½-in.
ped hole had been provided. It was found that such a plug would
d perfectly tight for all pressures that were used, i.e., as high as
) lb. gage.

22 A photograph of the apparatus in use is shown in Fig. 3. The pipe P is arranged so that steam may be brought to it from the main at any pressure up to 130 lb. per sq. in. gage. The supply pipe is ½ in. in diameter, and for a distance of about 20 ft. it is left bare in order that no superheat due to the expansion of the steam would carry over into the test pipe. The small pipe p just below the test pipe is the drain, and a little steam was allowed to flow through it continuously in order that no air or water would collect in the test pipe.

23 The temperature measuring instruments are shown also. T is one of the thermo-couples on the outside of the pipe. L, L are

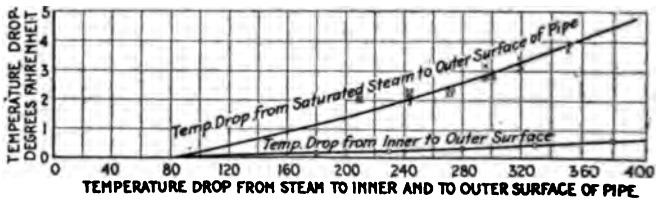


Fig. 4a Covered Pipe

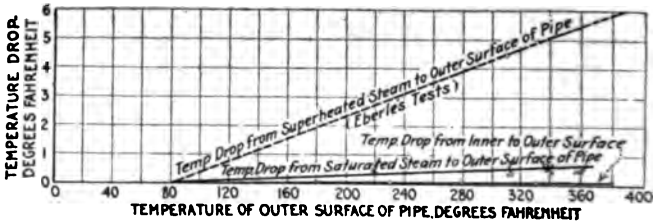


Fig. 4b Uncovered Pipe

Fig. 4 Temperature Drop from Steam to Inner and Outer Surfaces of Pipe

the leads to the couples; the leads on the left go through the plug in the end of the pipe to a couple in the steam. I is an ice bath serving as the cold terminal of the couples. M is a slide wire potentiometer and R is a variable resistance used in connection with it. C is a Weston standard cell, and B is a chloride accumulator or storage battery furnishing current for the potentiometer circuit. G is a very sensitive low resistance galvanometer.

24 Temperatures of steam and of outside of pipe were taken simultaneously at various steam pressures ranging from atmospheric to 130-lb. gage. This was done first with the pipe covered with

one inch thickness of sectional 85 per cent magnesia and later with pipe bare. In the first case from one to two hours' interval was allowed each time the pressure was changed for the temperatures to become constant before readings were taken. This was greatly in excess of the interval necessary, for it was observed that the temperatures did not vary noticeably after the first few minutes. In case of the bare pipe only half an hour was allowed, for without the covering the temperature of the pipe surface reached its new value very quickly after the steam pressure was changed. The results of the above described tests are given in tabulated form in Table 1 and the same expressed in the form of curves in Fig. 4*a* and *b*.

25 Fig. 4*a* shows the drop in temperature from saturated steam

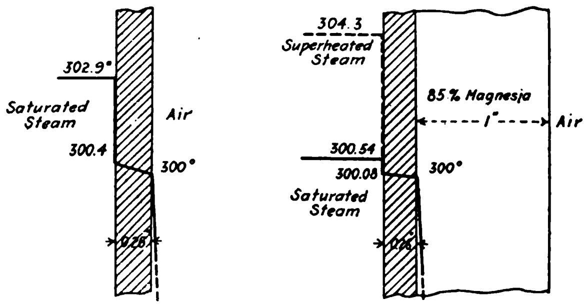


Fig. 5*a* Uncovered Pipe Fig. 5*b* Covered Pipe

Fig. 5 Temperature Drop from Steam to Outside of Pipe

to the inside surface of the pipe, the drop from inside to outside surface and the total drop from steam temperature to temperature of outside surface in the case of uncovered pipe. The drop through the metal of the pipe was so small as to be difficult of accurate measurement, and it was therefore calculated from the temperature gradient required to cause a known amount of heat to flow through the wall of pipe. This amount of heat is obtained from the results of the test of bare pipe that will be described later and from the conductivity of steel which is a quantity that has been established beyond dispute by many experimenters. The value of the latter is 417 B.t.u. per deg. fahr. temperature difference per sq. ft. per in. thickness per hour.[1] Therefore, it is a simple matter using equa-

[1]Mathematical Theory of Heat Conduction, by Ingersoll and Zobel. The value given was transformed into British units from 0.1436 cal. per cu. cm. per deg. cent. temperature difference per sec.

TABLE 1 DATA FROM TEMPERATURE GRADIENT INVESTIGATION

Barometer Rdg = 29.10 in. Hg Room Temperature = 80

Condition of Pipe Surface	Time July 31	Steam Pr. (Absolute)	Corresponding Saturation Temperature	Potential of Thermo-Couple in the Steam	Temperature of Steam	Potential of Thermo-Couple on Outer Pipe Surface	Temperature of Outer Surface of Pipe	Temperature Drop from Steam to Outer Surface	Calculated Temperature Drop through Wall of Pipe
Pipe Covered with 1" of 85 per cent Magnesia	1:21	151.8	359.4	8.215	360.4	8.20	359.9	0.5	through covered pipe = 0.027 °F; " " " = .041 "; " " " = .056 "; " " " = .070 "; " " " = .103 "
	1:23	148.8	357.8	8.17	358.8	8.155	358.3	0.5	
	1:27	146.7	356.8	8.145	357.9	8.13	357.4	0.5	
	1:29	149.8	358.4	8.195	359.7	8.18	359.2	0.5	
	1:34	151.8	359.4	8.23	361.0	8.215	360.4	0.6	
	4:30	18.8	224.6	4.42	219.4	4.41	218.9	0.5	
	4:34	18.8	224.6	4.43	219.8	4.415	219.2	0.6	
	4:39	18.3	223.2	4.425	219.6	4.415	219.2	0.4	
	4:43	18.3	223.2	4.425	219.6	4.415	219.2	0.4	
	6:20	46.0	275.8	5.855	274.3	5.84	273.8	0.5	
	6:25	45.5	275.1	5.85	274.1	5.835	273.6	0.5	
	6:27	45.5	275.1	5.85	274.1	5.84	273.7	0.4	
	6:30	46.2	276.0	5.87	274.8	5.855	274.4	0.4	
	6:35	46.5	276.5	5.875	275.1	5.85	274.6	0.5	
	7:45	81.1	313.0	6.915	313.2	6.895	312.5	0.7	
	7:47	82.1	313.9	6.92	313.4	6.905	312.9	0.5	
	7:49	82.1	313.9	6.925	313.6	6.91	313.1	0.5	
	7:52	82.1	313.9	6.925	313.6	6.91	313.1	0.5	
	7:56	81.6	313.4	6.915	313.2	6.89	312.4	0.8	
	9:15	112.9	336.7	7.58	337.9	7.565	327.2	0.7	
	9:18	114.0	337.6	7.575	337.7	7.555	337.0	0.7	
	9:24	117.5	337.9	7.67	341.0	7.65	340.3	0.7	
	9:27	117.5	339.7	7.66	340.6	6.645	340.1	0.5	
	9:31	117.5	339.7	7.65	340.3	7.635	339.7	0.6	
	9:35	116.5	339.1	7.625	339.3	7.61	338.7	0.6	
	Aug. 2								
Pipe Uncovered	2:55	14.5	111.5	4.20	211.0	4.15	209.0	2.0	At 180° Drop through bare pipe = 0.16 °F; " 230° " " " " = .22 "; " 280° " " " " = .33 "; " 330° " " " " = .46 "; " " " " " " = .61 "
	3:00	14.5	111.5	4.21	211.4	4.16	209.4	2.0	
	3:02	14.5	111.5	4.20	211.0	4.15	209.0	2.0	
	3:05	14.5	111.5	4.205	211.2	4.155	209.2	2.0	
	3:25	28.1	246.6	5.115	245.0	5.06	244.1	2.1	
	3:30	28.1	246.6	5.11	246.0	5.055	243.9	2.1	
	3:33	28.1	246.6	5.105	245.8	5.045	243.5	2.3	
	3:36	28.1	246.6	5.10	245.6	5.05	243.7	1.9	
	3:55	45.0	274.5	5.82	273.1	5.76	270.9	2.2	
	3:57	44.5	273.8	5.825	273.3	5.76	271.1	2.3	
	4:02	44.5	273.8	5.81	272.8	5.745	270.5	2.3	
	4:05	44.0	273.1	5.80	272.4	5.74	270.2	2.2	
	5:05	68.0	301.0	6.565	300.4	6.485	297.6	2.8	
	5:08	67.5	300.5	6.545	299.7	6.465	296.9	2.8	
	5:11	67.0	300.0	6.33	299.2	6.45	296.4	2.8	
	5:15	67.0	300.0	6.53	299.2	6.44	296.1	3.1	
	5:40	93.3	322.8	7.14	322.0	7.05	318.8	3.2	
	5:44	93.2	322.8	7.135	321.8	7.05	318.8	3.0	
	5:47	95.3	323.6	7.15	322.4	7.055	319.6	3.4	
	5:50	95.8	324.7	7.185	323.7	1.10	320.6	3.1	
	6:25	146.6	356.6	8.10	356.3	7.99	352.5	3.9	
	6:29	146.6	356.6	8.09	355.9	7.98	352.0	3.9	
	6:32	146.1	356.3	8.09	355.9	7.98	352.0	3.9	
	6:36	145.6	356.1	8.07	355.2	7.96	351.4	3.8	

tion $U = \frac{-k \,.\, d\,\theta}{d\,x}$, which has already been explained, to solve for the temperature gradient. For example, take a pipe temperature of 380 deg. fahr. and room temperature of 80 deg., making temperature difference 300 deg. Then from the bare pipe curve Fig. 10 it is found that the loss per degree temperature difference $=3.26$ B.t.u. per sq. ft. per hr. Therefore $U=300\times3.26=978$ and $-d\,\theta/d\,x=978/417=2.345$. The reason for the negative sign of the temperature gradient is that the flow of heat is in the direction of the fall and not of the rise of temperature. If the fall of temperature per inch $=2.345$ and the pipe is 0.26 in. thick, the fall through the pipe is $0.26\times2.345=0.61$ deg.

26 Fig. 4*b* shows, for pipes covered with one inch of sectional 85 per cent magnesia, the same relations as Fig. 4*a* does for bare pipe. In addition a curve, shown dotted, gives the drop in temperature from superheated steam to the outside surface of the covered pipe. This is only a rough approximation since it is constructed on the evidence given by a single test by Eberle.[1] However, it shows that the drop in temperature from steam to pipe is about ten times as great when the steam is superheated as when it is saturated at the same temperature. This accounts in a very large measure for the higher economy obtained by the use of superheat. It is certain that the heat losses would be lower from a pipe carrying superheated steam than the same one carrying saturated steam at the same temperature; for in the first case the pipe would be cooler than in the second, and the loss is dependent upon the temperature of the outside of the pipe and not that of the steam. The radiation losses from engine cylinders would be less with superheated steam for the same reason as above. Also there would be lower losses due to heating and cooling of the cylinder walls during the cycle, for the superheated steam would not heat them so hot, in comparison with its own temperature, as would saturated steam.

27 Fig. 5*a* is a section through the wall of a steam pipe showing the relative magnitudes of the temperature drops at various points when the pipe is not covered and its temperature is 300 deg. at the outside. Fig. 5*b* shows the same as the above for pipe covered with 1-in. of 85 per cent magnesia and for both saturated and superheated steam. This figure is of particular interest because it shows the effect of different substances in a pipe whose outside temperature

[1] Mit. über Forschungs-Arbeiten auf dem Geb. des Ing., heft 78.

is constant. The temperature gradient is greatly changed on the *inside* of the pipe, but not affected *outside.*

28 From the results given above it is apparent that the drop in temperature through the wall of a covered pipe containing saturated steam is so small in comparison with the total drop from steam temperature to room temperature that it may be neglected entirely. However, for superheated steam it should be taken account of, and the steam temperature obtained by adding to the pipe temperature the corresponding correction, which may be estimated from Fig. 4.

DESCRIPTION OF APPARATUS FOR TESTS OF COVERINGS

29 The general arrangement of apparatus used for the tests of pipe coverings is shown in Fig. 6. The test pipe is a 16-ft. section of standard 5-in. steel pipe closed at the ends and filled with gas engine cylinder oil. It contains also resistance coils which serve as an electric heater and a stirring device for keeping the oil in circulation. The remainder of the apparatus consists of the electrical instruments for measuring the energy input, a small electric motor for driving the circulating propeller, the thermometers for room temperature, and the thermo-couples and potentiometer for measuring the temperature of the pipe.

30 The fittings closing the ends of the large pipe are welded on to prevent the oil leaking out. At one end the pipe is closed by a cast iron cap welded to the pipe and at the other a flange is welded on and the end is closed by a blank flange bolted to the other with a gasket between. This blank flange has screwed into it the stuffing box for the shaft of the circulating propeller, two insulated terminals for the electric leads to the heating coils inside and a nipple leading to an overflow pipe which receives the oil expelled by the expansion of the large volume of oil that the test pipe contains. The arrangment of the stirrer and the electric leads may be seen in Fig. 6, and the overflow and the refilling chamber for keeping the pipe full of oil are shown in Fig. 7.

31 To provide for the better circulation of the oil, which is necessary for uniform temperatures at different points along the pipe, a positive circulating arrangement was used. This consists of a $1\frac{1}{4}$-in. pipe located in the center of the test pipe and running its entire length, except for about 2 in. at each end. In one end of this was placed a screw propeller driven by a motor placed outside of the test pipe, and by means of this propeller oil was forced through the small pipe and it had to flow back to the propeller through the

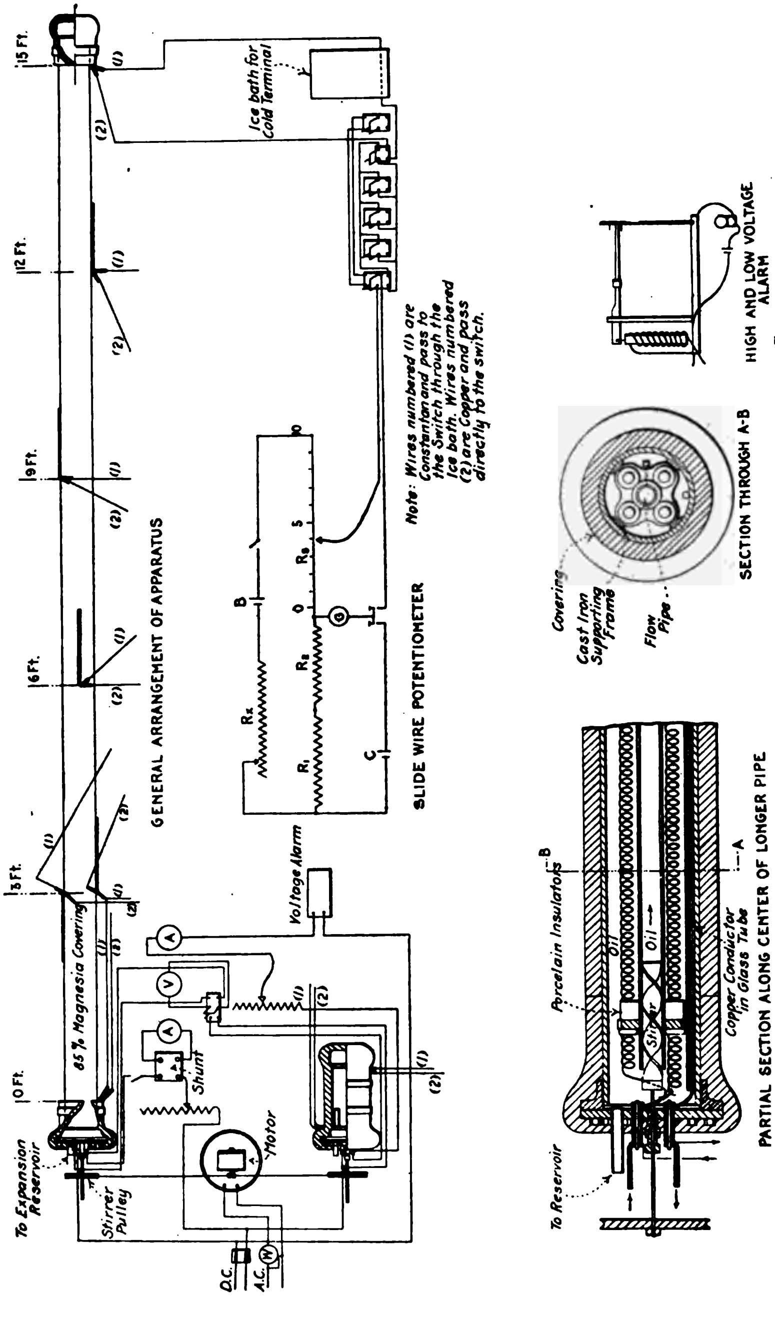


FIG. 6 APPARATUS FOR OBTAINING HEAT LOSSES THROUGH PIPE COVERINGS

annular space between the two pipes where it came in contact with the heating coils. These heating coils, made of nichrome resistance wire, are four in number and are arranged symmetrically around the small pipe mentioned above, the whole being supported by a cast iron frame which carries porcelain insulators for the coils. All this is shown in Fig. 6. The four coils are connected in parallel, one end of each being connected to one of the terminals in the front end of the pipe and the other to a copper cable which runs from the other terminal, through a glass tube as an insulator, to the back of the pipe.

32 About 5 in. at each end of the test pipe is covered with sectional 85 per cent magnesia 1 in. thick, leaving exactly 15 ft. of the central portion of the pipe as the test section, which will accommodate 5 lengths of standard pipe covering. The remaining surface of the ends is covered with plastic 85 per cent magnesia to a depth of about an inch. The pipe is suspended in a horizontal position by wires from the ceiling attached to steel bands placed around the short end sections just described. These remained in place throughout the entire series of tests, and the only covering changed was the five lengths on the 15-ft. test section.

33 Since the covering of the ends was the same for all the tests of one inch coverings, some means of correcting for the losses through these end portions had to be devised. The device used in measuring the amount of this correction is the short pipe shown at the lower left hand corner of the upper diagram, Fig. 6. It is an exact reproduction of the permanently covered ends of the test pipe. If the 15-ft. test section already mentioned were cut out and the two ends of the test pipe brought together, an exact duplicate of the short pipe would result, in so far as length, area exposed, and character of covering are concerned. This similarity is very noticeable in Fig. 7. Certain changes had necessarily to be made in the heating coils in the short pipe and the circulating pipe was omitted, but outside the likeness was exact. All parts tending to increase the heat losses from the ends of the test pipe, such as the stuffing box for the propeller, the insulated terminals for the electric leads, the overflow pipe, the refilling reservoir, etc., were reproduced in the short pipe as nearly like those on the test pipe as it was readily possible to make them. Therefore, the difference between the loss from the test pipe and that from the short pipe represents the exact loss from the 15-ft. section covered with five standard lengths of commercial covering.

34 For the thickness tests, where thicknesses up to 3 in. were ted, the ends of the test pipe were covered with double standard ckness of 85 per cent magnesia covering, and the short pipe was ered in the same way. A test was made on the short pipe under se conditions and the new end correction determined.

35 The power required to drive the circulating propeller was remely small, as compared with the energy supplied to the heater s. This fact would justify neglecting it entirely, but it was med better to take account of it. Fig. 8 shows the results of eral tests on the power consumed by the oil circulating device various temperatures of the oil. The points marked *o* are those power to run the propeller on the test pipe and those marked *x*

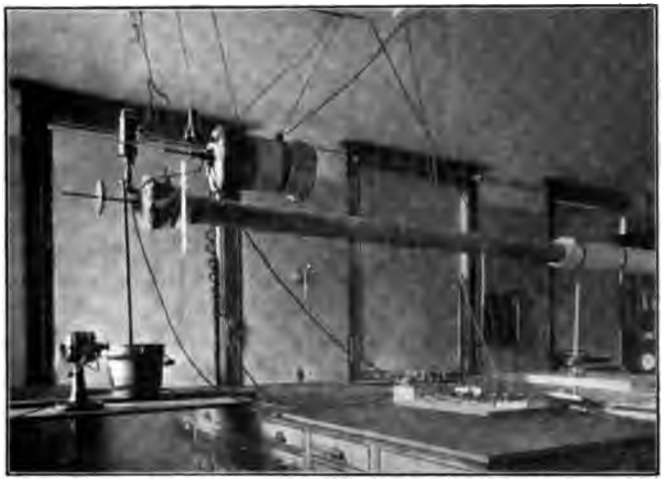


FIG. 7 VIEW OF SHORT PIPE AND FRONT END OF TEST PIPE

for that on the short pipe. It is noticeable that the power sumed in either case is the same within the limits of error of asurement; therefore, the error which would be introduced by lecting the heating effect of stirring the oil entirely disappears in net result, which is the difference between the losses from long l short pipes.

36 The thermo-couples for measuring the temperature of the e are similar to those described in a previous section, being made copper and constantan or advance wire. The calibration curve, . 2, applies to these couples as well as to those used in making temperature gradient determinations. There are seven of these

couples on the test pipe and two on the short pipe. They are placed on the top, side, and bottom of the pipe as shown in Fig. 6, and with this arrangement the average temperature of the outside of the pipe may be obtained with great accuracy. Each couple is imbedded in the metal of the pipe. A shallow hole, just large enough to admit the point forming the junction of the two metals of the couple, was drilled into the pipe, and after insertion the steel was forced down against this point by means of a center punch. The absolutely sure metal to metal contact secured in this way reduced to a minimum the chances for a drop in temperature from the metal of the pipe to the couple. The couple leads are made of wires insulated with a double covering of silk, and the portions under the pipe covering where the silk would be damaged by the heat are further protected by a wrapping of asbestos cord.

37 The potentiometer used for measuring the e.m.f. of the couples is shown diagrammatically in Fig. 6 and in the foreground of Fig. 9. No detailed explanation of its working will be given here since this may be found in any advanced text book on physics.

38 Direct current at 110 volts was used to supply the energy for the tests. It was obtained from a 25-kw. Curtis turbo-generator in the Steam Laboratory of the University of Wisconsin, and as the electric heater and the control rheostats were the only load on the machine, a very steady voltage was maintained.

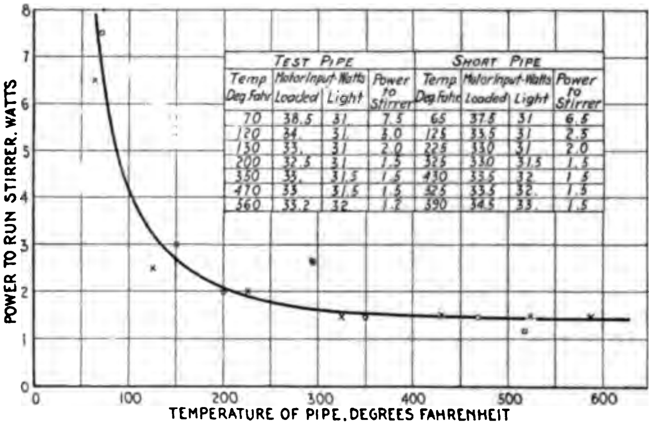
39 In Fig. 6 is shown an instrument which gave warning of fluctuations of voltage. It consists of a solenoid in series with the heater coils, and a soft iron core hanging inside the solenoid at the short end of a balanced lever. If the voltage goes up the current increases, and the solenoid becoming a stronger magnet draws the core downward. A greatly multiplied upward movement takes place at the tip of the long end of the lever, where contacts are provided for the ringing of an electric bell. In opposite manner if the voltage decreases the strength of the solenoid decreases, the core is lifted by the weight of the lever and the tip moves downward until it strikes another contact and rings the bell. This instrument is so sensitive that it may be relied upon to give warning of a change of voltage of 0.1 volt in either direction.

40 For regulating the current, there is a wound wire rheostat with three coils which may be used either all in parallel with a capacity of 75 amp. or all in series for currents less than 25 amp. The finer adjustments are made by means of a small wire rheostat in series with a lamp bank and both in parallel with the large rheostat.

41 The instruments for measuring the energy to the pipe are a Weston D.C. voltmeter and a Weston D.C. millivoltmeter and shunt, the latter combination serving as an ammeter. These instruments were calibrated at the Standards Laboratory of the University of Wisconsin and found to be very accurate in their readings. Fig. 9 shows all the control apparatus very well.

Method of Performing the Tests

42 The coverings, before being tested for their heat insulating qualities, were placed on the steam pipe, Fig. 3, and allowed to dry for a week. Steam in the pipe was kept at about 130 lb. pres-



Test Pipe				Short Pipe			
Temp Deg Fahr	Motor Input-Watts Loaded	Light	Power to Stirrer	Temp Deg Fahr	Motor Input-Watts Loaded	Light	Power to Stirrer
70	38.5	31.	7.5	65	37.5	31	6.5
120	34.	31.	3.0	125	33.5	31	2.5
130	33.	31	2.0	225	33.0	31	2.0
200	32.5	31	1.5	325	33.0	31.5	1.5
350	33.	31.5	1.5	430	33.5	32	1.5
470	33	31.5	1.5	525	33.5	32.	1.5
560	33.2	32	1.2	590	34.5	33.	1.5

Fig. 8 Power Required to Circulate the Oil in Pipe

sure throughout that time, and as a result the coverings were thoroughly dry when placed on the test pipe. Before being placed each was weighed and its thickness at each end was measured. The thickness so measured is called the actual thickness in the tables. The sections were fitted carefully so that all joints were snug and the canvas was pasted over the seams in the same manner as in power plant practice, the paste being allowed to dry thoroughly before the test began. The circumferences of the bare pipe and of outside of all coverings were measured and from these data were calculated the apparent thicknesses of the coverings, that is the distances from outside of pipe to outside of coverings. Where the covering was made up of soft sheets, as of felt, etc., pinned together, only the apparent thickness was measured.

43 A comparatively high current was passed through the pipe

until it was heated to near the desired temperature, and then the current was lowered to such a value as would just hold the temperature of the outside of the pipe constant. A little experimenting was sufficient to show approximately what power input was necessary to maintain this temperature. The current was then adjusted to such a value as would make the power input just a little greater than the estimated value, and was maintained at exactly this value for a period of several hours. If the first estimate was nearly correct, the temperature would rise slowly for a while and then become constant at some point at which the losses were exactly equal to the electrical input. This usually required from 4 to 8 hr. for one-inch

Fig. 9 View of Instruments and Electrical Control Apparatus

covering, according to the correctness of the preliminary estimate. For greater thicknesses longer time was required, up to 24 hr. for 3-in. thickness. When the temperature had remained practically constant for an hour, readings were taken of pipe temperature, room temperature, current in heater, and voltage across heater terminals. Then the current was diminished by about half an ampere and the temperature allowed to fall until it reached a value where the losses were just equal to this smaller amount of energy supplied, and readings were taken as before. The product of volts and amperes gave the energy in watts supplied to the heater. To maintain that current and voltage for one hour required an equal number of watt-hours, and the values of watt-hours were transformed into B.t.u.

by multiplying by 3.413. The resulting losses in B.t.u. per hour were plotted against difference between temperature of outside of pipe and room temperature. Such a curve is shown in Fig. 10.

44 The advantage of approaching the desired temperature from both below and above was that the error due to not waiting long enough for constant conditions would cause too high a value

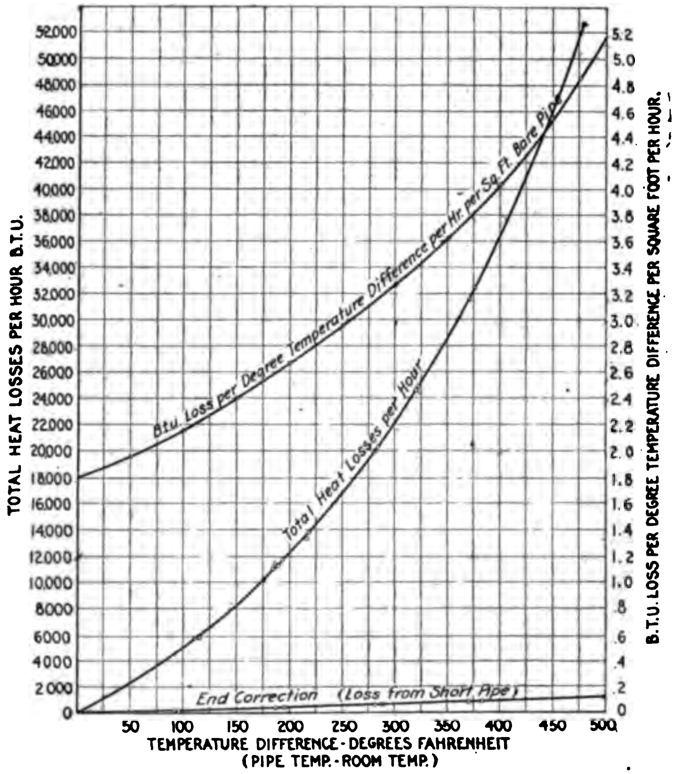


FIG. 10 TEST OF BARE PIPE

for loss at the given temperature in the first case and too low a value in the second. If these values checked, that was an indication that conditions were constant when the readings were taken; if they did not, then the correct value lay somewhere between the two. These duplicate determinations usually checked within one per cent.

45 If the preliminary estimate had not been nearly right the temperature might have either risen or fallen rapidly. In that case the current would have been regulated down or up as the case re-

quired until a point was reached where the temperature changed very slowly, and here the current was maintained until the temperature became constant and readings were taken as before. If the temperature fell slowly at the start, the point approached from above was sometimes taken first and then the current increased by about half an ampere and the other point determined as already described.

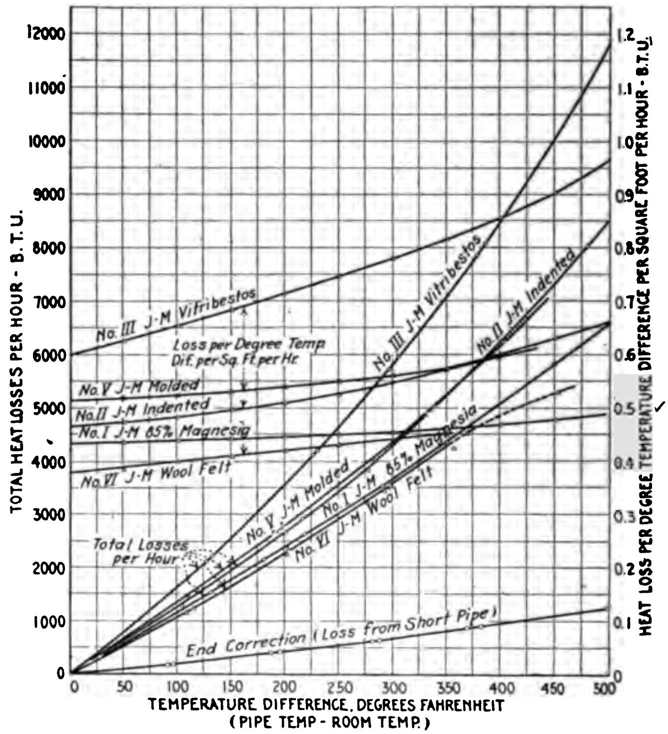


FIG. 11 TESTS OF COVERINGS I, II, III, V, & VI

46 The room temperature was taken as the average of the readings of five thermometers at different points in the room and all about equidistant from the pipe. These were shielded from direct radiation from the pipe by paper screens placed between them and the pipe, so that they registered the correct temperature of the air in the room. All doors and windows of the room were kept closed during the tests to avoid air currents other than those produced by the heated pipe itself.

47 It has already been pointed out that the scientific method

of conducting a research is to first ascertain what variables are likely to effect the results, and then, keeping all constant but one, to vary that one between wide limits to observe its effect. Each of the others in its turn is treated in the same way.

48 The three factors most important in determining the amount of heat that will be transmitted by a given insulating material are (1) the character of the material or, in other words, its conductivity; (2) the temperature difference between its two boundaries, and (3) the thickness of the layer of material. Two others of lesser importance are the character of the surface and the velocity of air

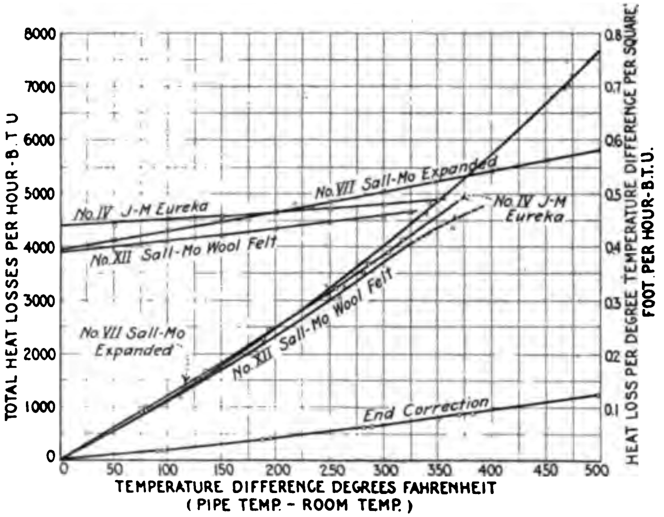


FIG. 12 TESTS OF COVERINGS IV, VII, & XII

fanning that surface. These last two were kept constant during all tests, but the effect of varying them with all other conditions constant was not determined. This, however, is contemplated for the near future.

49 The character of materials was varied while temperature range and thickness were kept constant by making tests of 17 different pipe coverings of approximately the same thickness and subjected to the same temperatures. The effect of the character of the material is shown in Fig. 15.

50 Tests were made at pipe temperatures ranging from 175 to 575 deg. fahr., and at least eight or ten tests were made on each material. This was varying the temperature while character of

material and thickness remained the same. The effect of temperature in the losses is shown in Figs. 11 to 15 and Table 2. The excellent agreement of so many tests on the same covering is the best indication of the accuracy of the work.

51 For two different materials the thickness was varied from 0 to 3 in., while in each case material and temperature range were constant. The effect of thickness is shown in Figs. 18 to 21.

DESCRIPTION OF COVERINGS TESTED

52 All the coverings tested were bought on the open market and the dealers were given no intimation that the coverings ordered were

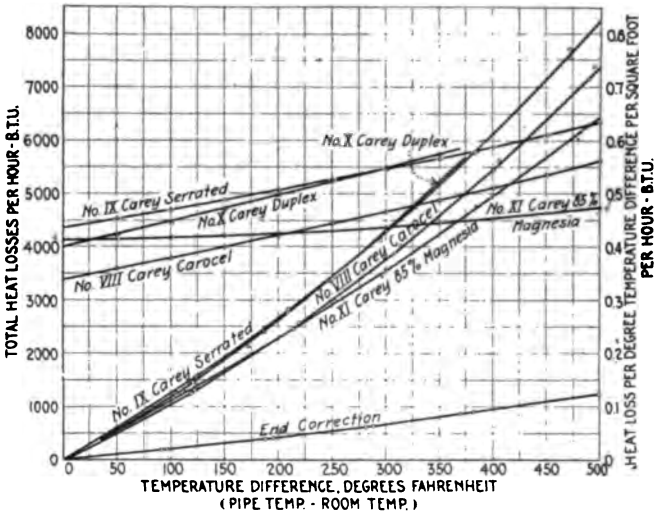


FIG. 13 TESTS OF COVERINGS VIII, IX, X, & XI

to be tested. In every case shipment was so prompt as to remove any suspicion that the order was filled from other than stock material.

53 A brief description of each of the 26 coverings tested is given below. The statements as to whether the covering was recommended for high or low pressure or superheated steam pipes were furnished by the manufacturers and are not conclusions drawn from the tests. The weight per foot in each case is understood to mean the average weight per lineal foot of 5 in. covering, and the thickness given is the average thickness.

1 *J-M 85 Per Cent Magnesia.* A molded sectional covering for

use on high pressure steam pipes. Contains 85 per cent by weight of magnesium carbonate and the remainder is principally asbestos fiber. Weight per foot is 2.92 lb. and the thickness 1.08 in.

II *J-M Indented.* Made up of layers of asbestos felt which has in it indentations, about 1¼ in. in diameter and ⅛ in. deep, spaced very close to each other in staggered rows. Suitable for use on pipes containing high pressure steam. Weight per foot 3.46 lb. and thickness 1.12 in.

III *J-M Vitribestos.* An asbestos air cell covering made of alternate layers of smooth and corrugated vitrified asbestos sheets.

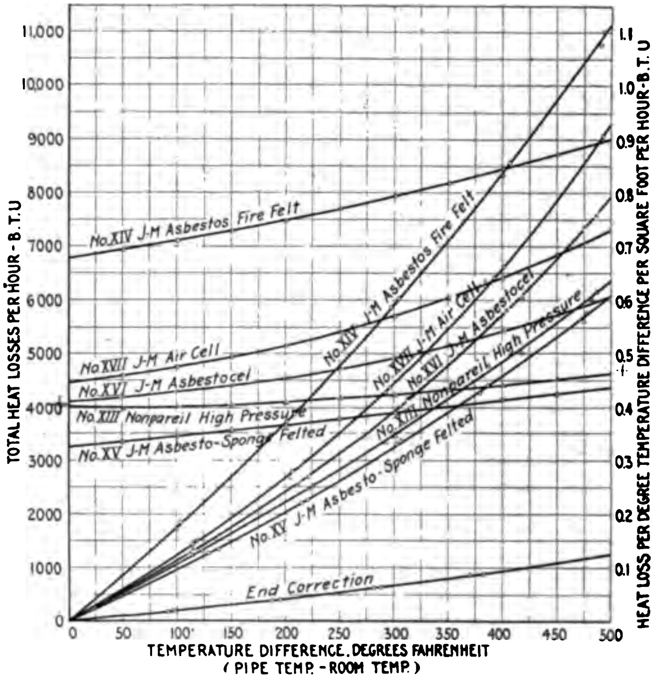


FIG. 14 TESTS OF COVERINGS XIII, XIV, XV, XVI, & XVII

Corrugations are about ¼ in. deep and run lengthwise of the pipe. Recommended for use on high pressure and superheated steam pipes and for stack linings, etc. Weight per foot 4.05 lb. and thickness 0.96 in.

IV *J-M Eureka.* For use on low pressure steam and hot water pipes. Made of ¼-in. of asbestos felt on the inside of the section and the balance of alternate layers of asbestos and wool felt. Weigh 4.60 lb. per ft. and is 1.04 in. thick.

V *J-M Molded Asbestos.* A molded sectional covering for use on low and medium pressure steam pipes. Made of asbestos fiber and other fireproof material. Weight per ft. 5.53 lb. and thickness is 1.25 in.

VI *J-M Wool Felt.* A sectional covering made of layers of

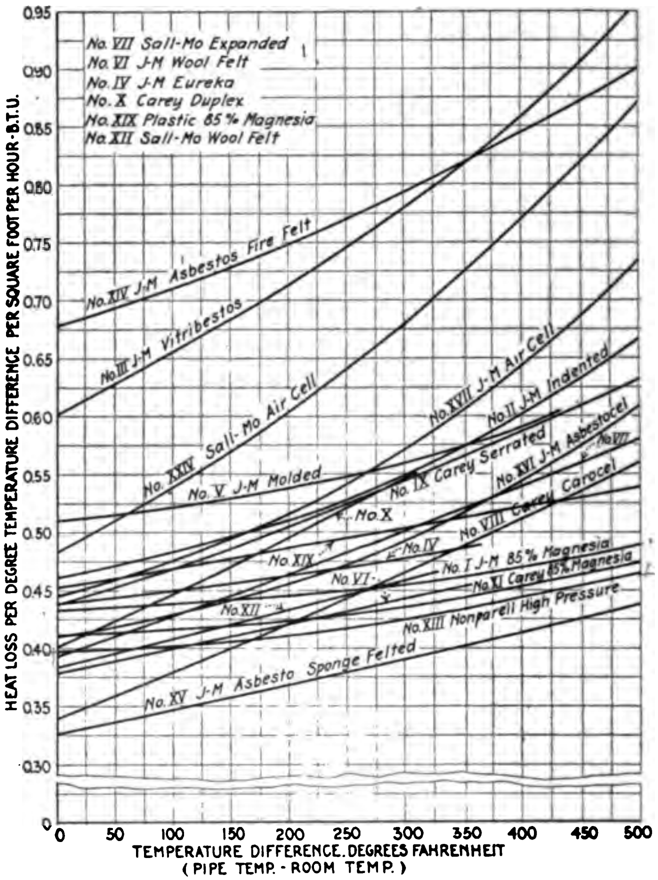


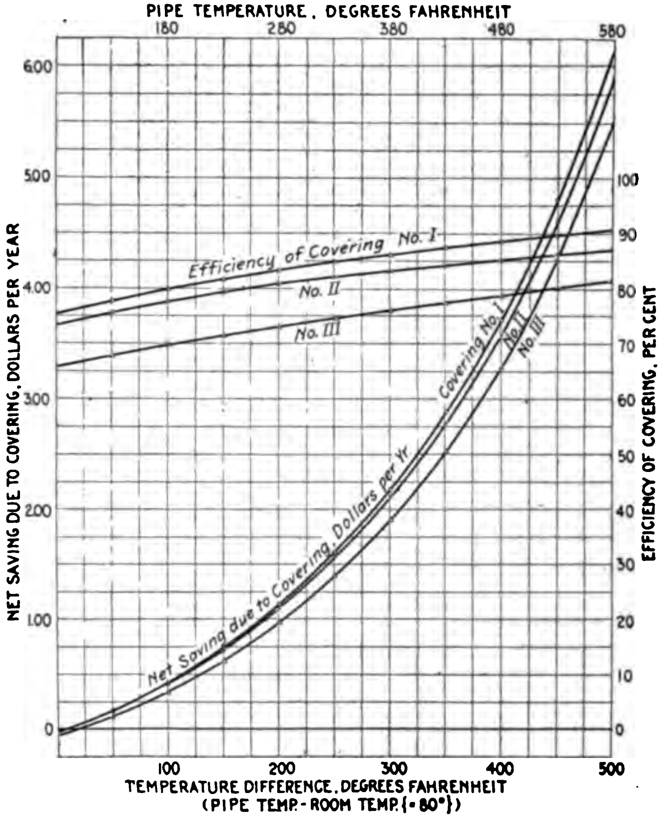
FIG. 15 SUMMARY OF RESULTS ON SINGLE THICKNESS COVERINGS

wool felt with an interlining of two layers of asbestos paper. M be used on low pressure steam and hot water pipes. Weight p ft. 2.59 lb. and thickness 1.10 in.

VII *Sall-Mo Expanded.* A covering for use on high and l pressure steam pipes. Made of eight layers of material, each co sisting of a smooth and a corrugated piece of asbestos paper, t

:ugations being so crushed down to form small longitudinal air ces. Weight 3.47 lb. per ft., and thickness 1.07 in.

VIII *Carey Carocel.* Composed of plain and corrugated as- tos paper firmly bound together. Corrugations are approxi- :ely ⅛ in. deep and run lengthwise of the pipe. For use on



16 EFFICIENCY AND NET SAVINGS CURVES FOR COVERINGS I, II, & III

lium and low pressure steam pipes. Weight 3.06 lb. per ft. and :kness 0.99 in.

IX *Carey Serrated.* A covering for use on high pressure steam es. Composed of successive layers of heavy asbestos felt having ely spaced indentations in it. Weight 5.66 lb. per ft., and :kness 1.00 in.

X *Carey Duplex.* For use on low pressure steam and hot water es. Made of alternate layers of plain wool felt and corrugated asbestos paper firmly bound together. Corrugations run lengthwise of the pipe and make air cells approximately ¼ in. deep. Weighs 1.79 lb. per ft. and 0.96 in. thick.

XI *Carey 85 Per Cent Magnesia.* A covering for high pressure steam and similar in composition to No. I. Weight per foot 2.74 lb. and thickness is 1.10 in.

XII *Sall-Mo Wool Felt.* Similar to No. VI except that it has no interlining of asbestos paper. For use on low pressure steam and hot water pipes. Weight per foot 3.73 lb. and thickness is 1.01 in.

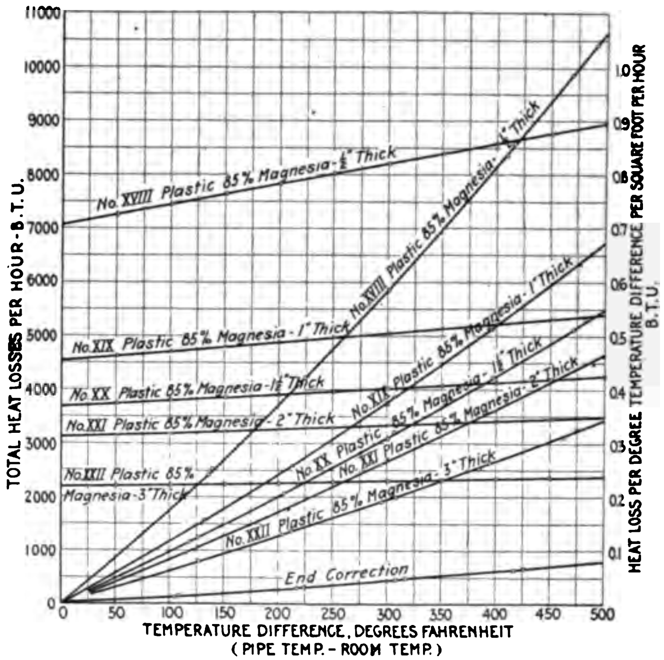
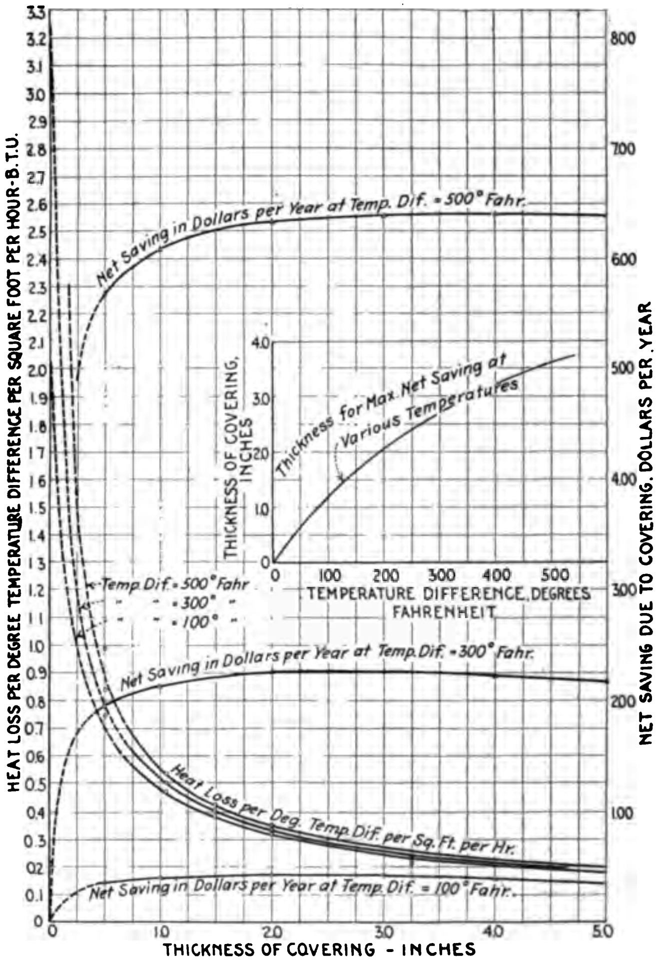


FIG. 17 TESTS OF COVERINGS XVIII TO XXII

XIII *Nonpareil High Pressure.* A molded sectional covering consisting mainly of silica in the form of diatomaceous earth—the skeletons of microscopic organisms. For use on high pressure and superheated steam pipes. Weighs 2.96 lb. per ft., and is 1.16 in. thick.

XIV *J-M Asbestos Fire Felt.* Consists of asbestos fiber loosely felted together, forming a large number of small air spaces. For use on high pressure and superheated steam pipes. Weight per ft. is 3.75 lb., and thickness 0.99 in.

:V *J-M Asbestos Sponge Felted.* Covering is made from a felt of asbestos fiber and finely ground sponge forming a very .ar fabric. Made up of 41 of these sheets per in. thickness and paces are formed between the sheets in addition to those in the



18 THICKNESS-SAVINGS CHART FOR 85 PER CENT MAGNESIA COVERINGS

itself. Specially recommended for high pressure and super-.ted steam pipes. Weight per ft. 4.04 lb. and thickness 1.16 in.

XVI *J-M Asbestocel.* For use on medium pressure steam and .ting pipes. Consists of alternate sheets of corrugated and plain asbestos paper forming air cells about ⅛ in. deep that run around the pipe. Weight per ft. 1.94 lb., and thickness 1.10 in.

XVII *J-M Air Cell.* Made of corrugated and plain sheets of asbestos paper arranged alternately so as to form air cells about ¼ in. deep running lengthwise of the pipe. For use on medium

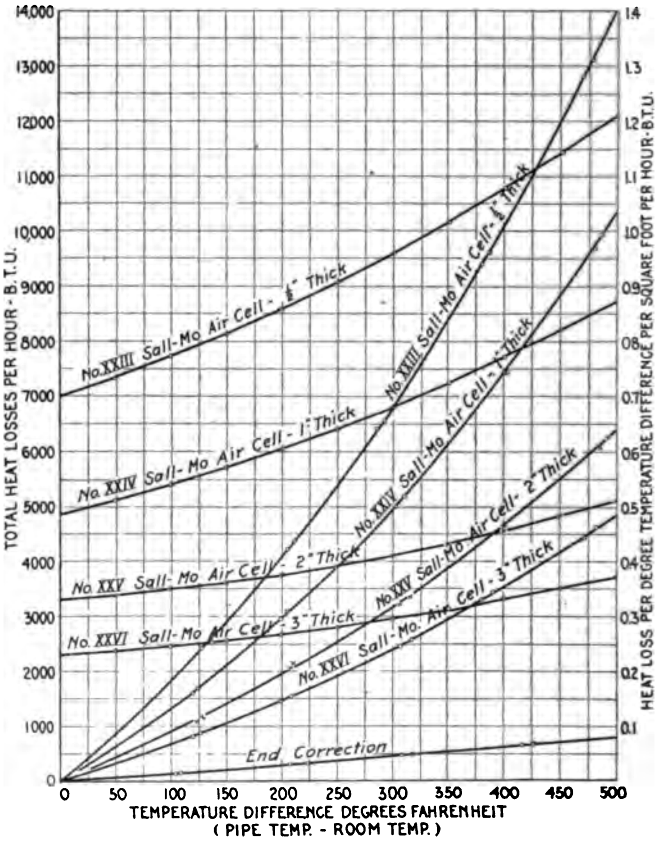


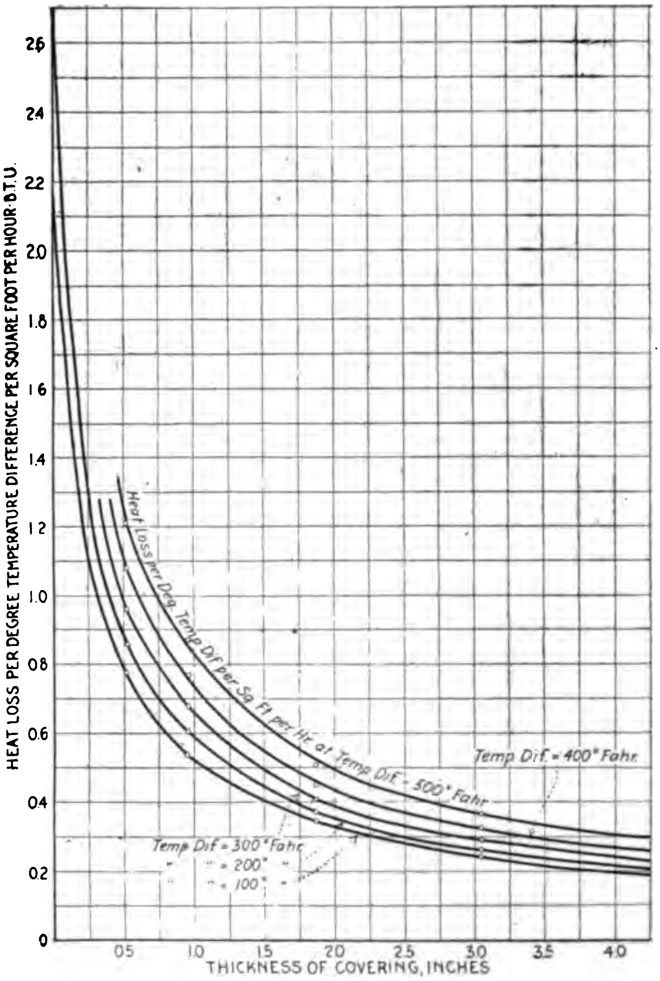
FIG. 19 TESTS OF COVERINGS XXIII TO XXVI

pressure steam and heating pipes. Its weight per ft. is 1.55 and thickness is 1.00 in.

XVIII ½-*In. J-M Plastic* 85 *Per Cent Magnesia.* For use on fittings, valves, irregular surfaces, boiler coverings, etc. Similar in composition to the sectional 85 per cent magnesia, but applied in the form of a cement or plaster. Thickness was 0.51 in. for the first test, and weight per ft. was 1.51 lb.

XIX 1-*In. J-M Plastic 85 Per Cent Magnesia.* Thickness in., weight per ft. 3.33 lb.

XX 1½-*In. J-M Plastic 85 Per Cent Magnesia.* Thickness in., weight per ft. 5.23 lb.



. 20 THICKNESS HEAT LOSSES CURVES FOR SALL-MO AIR CELL COVERINGS

XXI 2-*In. J M Plastic 85 Per Cent Magnesia.* Thickness 9 in., weight per ft., 7.46 lb.

XXII 3-*In. J-M 85 Per Cent Magnesia.* Consisted of the two hes of plastic covering of No. XXI and one standard thickness layer of sectional covering outside of that. Thickness 3.24 in., weight per ft. 11.67 lb.

XXIII *½-In. Sall-Mo Air Cell.* This covering is similar in composition and uses to No. XVII. Its thickness is 0.51 in. and its weight per ft. is 0.99 lb.

XXIV *1-In. Sall-Mo Air Cell.* Thickness 0.95 in., weight per ft. 1.57 lb.

XXV *2-In. Sall-Mo Air Cell.* Thickness 1.86 in., weight per ft. 3.58 lb.

XXVI *3-In. Air Cell.* Consisted of two inches of *Sall-Mo* and one inch of *J-M Air Cell.* Thickness 3.04 in., weight per ft., 6.66 lb.

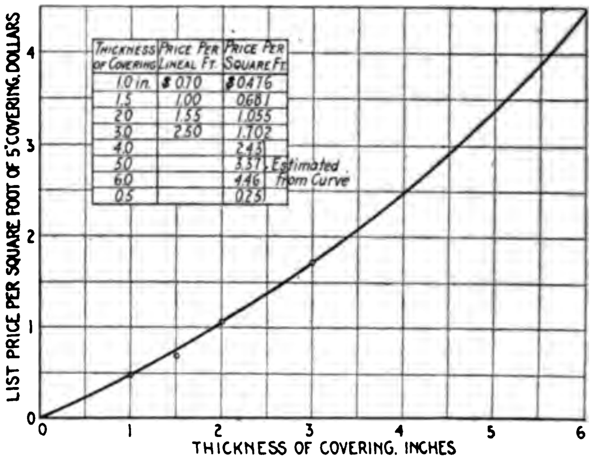


Fig. 21 Standard List Prices of 5-inch Pipe Coverings

DISCUSSION OF RESULTS

54 The results of the test on bare pipe are shown in Fig. 10. The total loss curve is plotted directly from the data obtained during the test. The ordinate of any point is the total heat loss per hour, which is the equivalent of the electrical energy required to maintain the pipe at the given temperature, and the abscissa is the difference between pipe temperature and room temperature. On the same sheet is plotted a curve of heat losses per hour from the short pipe at various temperature differences; this curve is called "end correction." The difference of ordinates between the two curves at any value of temperature difference gives the net heat loss per hour from the 15-ft. length of bare pipe. This net loss divided by the tem-

perature difference and the area of test section (22.03 sq. ft.) gives the heat loss per degree temperature difference per sq. ft. per hour.

55 The curve of net heat losses per degree temperature difference per square foot per hour is shown on the same sheet as the total loss curves but to a much larger scale. This curve shows that the heat loss per degree temperature difference is far from being a constant at all temperatures as has been assumed or implied by most former investigators. Most of their tests were made at temperatures nearly corresponding to the saturation temperature at 100 lb. per sq. in. gage, and the results plotted to the same scale would give points near the curve of Fig. 10. But it is at once apparent that an error of about 100 per cent would be introduced by using the value determined at 300 deg. fahr. for 500 deg. fahr. where the rate of loss per degree is almost twice as great.

56 The heat losses in this case are made up of two components, each of which tends to cause the losses to increase with temperature at a much more rapid rate than that at which the temperature itself increases. These are radiation and convection. According to Stefan's Law, the radiation varies as the fourth power of the absolute temperature. The velocity of convection currents also increases much more rapidly than the temperature; therefore, a combination of these two effects will cause the resultant losses to increase with temperature much more rapidly than does temperature itself.

57 Figs. 11 to 14 inclusive show curves from tests of 17 different coverings of single thickness, i.e., about one inch. The data and computations for each of these tests are of the same form as those for the bare pipe tests. The ordinate, heat loss per degree temperature difference per square foot per hour, in each of the figures mentioned above refers to square feet of *pipe surface* and not of covering surface. This is true of all other figures in which this ordinate appears unless it is specifically stated to be otherwise.

58 If the transfer of heat from the pipe to air through the intervening layer of insulating material were purely a case of heat *conduction*, the transfer of heat would be exactly proportional to the first power of the temperature difference and the rate curves would be straight lines parallel to the temperature axis. But they slope upward as the temperature difference increases, being in that respect similar to the rate curve for bare pipe, though they do not bend nearly so rapidly. It is noticeable, further, that some are nearly parallel to the temperature axis and that others slope much more rapidly. The reason for the increase in the rate of heat flow from

bare pipe has already been explained, and in a measure the same explanation applies to the case of covered pipe. The radiation and convection from the surface of the covering increases at a more rapid rate than the temperature as the pipe is heated up. But the principal reason for the increase in the rate of losses in the case of covered pipe is within the covering itself. Where the air spaces in the coverings are large, heat is transmitted through them by radiation and convection in addition to the heat that is *conducted* by the solid material, and as already explained this effect of radiation and

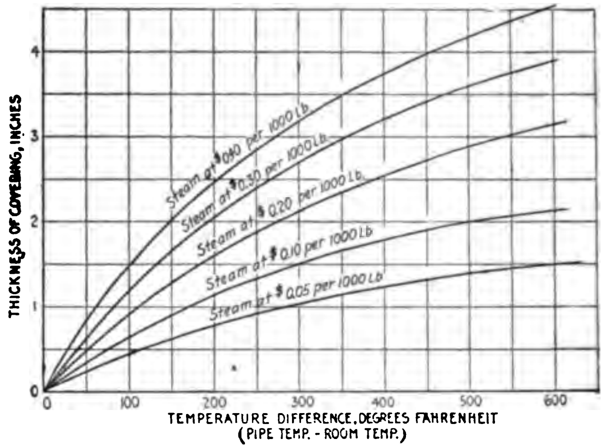


Fig. 22 Chart for Determining Proper Thickness of 85% Magnesia for Maximum Net Savings at Various Temperatures and Prices of Steam

convection which now is within the covering itself increases at a more rapid rate than does the temperature. This accounts for the greater slope of the rate curves for air cell and similar coverings than for magnesia, diatomaceous earth, etc. In the case of the latter coverings, the increase in rate of loss is probably due only to the increased radiation and convection from the *surface*.

59 According to Nusselt[1] the conductivity of many poorly conducting substances increases nearly as the absolute temperature instead of remaining constant. The tests described in the present paper establish the fact that the conductivity does not remain constant, but in no case except in that of the bare pipe did it increase quite as rapidly as the absolute temperature. Conductivity in this

[1]Zeit. d. Ver. deut. Ing., vol. 52, p. 906.

sense is rate of heat conduction per degree temperature difference per square foot per inch thickness per hour.

60 The net heat loss curves from all the tests of coverings 1 in. thick are assembled in the form of a general summary in Fig. 15. From the curves shown in this figure, one can tell at a glance what coverings are the more efficient at any temperature. Of all the coverings tested, the four best, at temperatures of 200 to 600 deg. fahr., are J-M Asbestos-Sponge Felted, Nonpareil High Pressure, Carey 85 Per Cent Magnesia and J-M 85 Per Cent Magnesia, ranking in the order named with the first well ahead of all the others. Those losing the greatest amount of heat were J-M Vitribestos and J-M Firefelt. The first of these latter is little used as a pipe covering, being employed mostly for stack linings, etc., while the virtue of the second is in its being a heat-proof material suitable for use as the layer in contact with a pipe carrying superheated steam, where the better insulating material used for the outer layers could not stand the temperature of superheat.

61 The saving in dollars per year due to use of covering has been calculated for each of the coverings tested and the results appear in Table 2. Also, the first cost of the covering is taken account of and values of net saving and per cent saving on investment are given for values of temperature differences of from 1 to 500 deg. fahr. The saving in B.t.u. per degree temperature difference per square foot per hour was first found for each temperature by subtracting from the bare pipe loss at that temperature the loss from covered pipe at same temperature. Then the total saving per square foot for a year of 365 twenty-four hour days was found by multiplying the saving per degree per hour per square foot by 8760 and by the temperature. The cost of heat was taken at $0.30 per million B.t.u., which is nearly equivalent to $0.30 per 1000 lb. of steam. Using this value, the saving in dollars per square foot per year was computed. The first cost of covering was ascertained from the manufacturers, and 10 per cent of list price was added for erecting and 10 per cent more for painting. The cost of covering *per year* was taken as 14 per cent of the total first cost, the 14 per cent including interest, depreciation, repairs, insurance, etc. The difference between the total saving per year and the cost of covering per year is the net saving per year.

62 In the above computations the period of effective service was assumed to be the same for all the coverings. This is not strictly in accordance with the facts, but for the single thickness coverings,

for which those computations were made, the saving due to the use of covering is so great in comparison with the cost of the material that the error in net saving, due to such an assumption as the above, is negligibly small. However, where the first cost is greater in comparison with the annual saving, as in the case of thicker coverings than those mentioned above, the durability becomes a very important factor. In every case where comparative costs of different coverings are desired the cost per year rather than the first cost should be considered.

63 Efficiencies of all the coverings tested, i.e., loss from covered

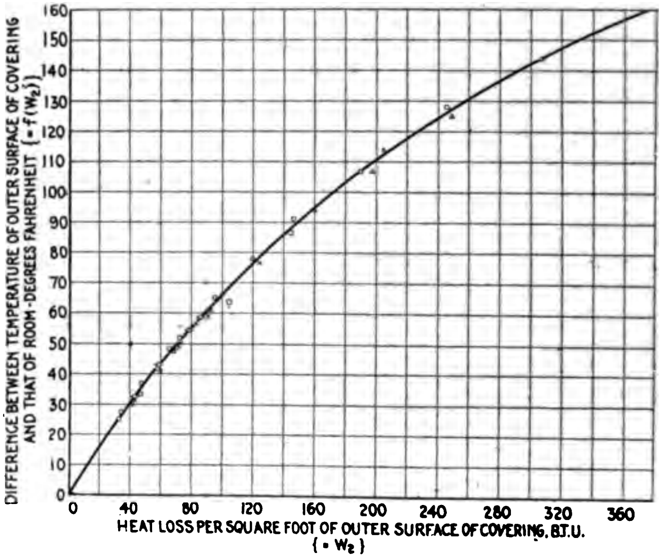


FIG. 23 CURVE SHOWING RELATION OF HEAT LOSSES TO TEMPERATURE DIFFERENCE BETWEEN COVERING SURFACE AND SURROUNDING AIR

pipe divided by loss from bare pipe of same area and temperature, are also shown in Table 2.

64 The data in Table 2 have been plotted in the form of curves, shown in Fig. 16, for the first three coverings tested. The reasons for not plotting the data for all the coverings was that these three tests serve to show the manner in which saving and efficiency vary with temperature, and little would be gained by plotting curves for all the tests unless all could be put on the same sheet. This would not be satisfactory without the use of a very large scale, for many of the curves fall very near each other. However, all the data have been

computed and any curve desired may be plotted in a very short time, by anyone wishing it.

65 The facts brought out by these curves are that the efficiency and saving increase with the temperature on account of more rapid increase, with temperature, of losses from bare pipe than of losses from covered pipe.

66 The results of tests on ½-in., 1-in., 2-in., and 3-in. thicknesses of plastic 85 per cent magnesia are given in Fig. 17. Fig. 18 shows the variation of losses with thickness at temperature difference of 100, 300 and 500 deg. fahr. and for thicknesses from 0 to 5 in. These curves were plotted from data calculated from the theoretical equation using constants determined in the tests herein described. The points marked *o* are values from the actual experiments and they fall on the calculated curves with remarkable regularity except at the ½-in. and 3-in. points. However, there are good reasons why these should fall at the places they do with reference to the curve. The ½-in. thickness cracked and checked considerably when heated up and this naturally decreased its insulating value, so that the values of losses would be greater than those calculated from the equation which made use of conductivities obtained at greater thicknesses where conditions were more nearly uniform. The outer one inch thickness on the test of 3-in. covering was not plastic, but was sectional magnesia covering which has a lower conductivity than the plastic. This accounts in a measure for the experimentally determined points falling below the theoretical curve. In view of their close agreement with all the other points and of the good reasons why they do not agree with those for ½ and 3-in. the theoretical curves may be considered more reliable than curves joining all of the points would be.

67 Figs. 19 and 20 show results of similar tests on different thicknesses of air cell covering. These two coverings, magnesia and air cell, were chosen because of their being representative of the two different classes of materials, i.e., those with very small air spaces and those with rather large air spaces. The mathematical derivation of the curves for variation of losses with thickness could not be used in this case; for the conductivity of the material decreased with the thickness on account of the better enclosing of the air spaces by the successive layers of covering. Therefore, the curves were drawn through the average of the points determined by experiment.

68 The net saving in dollars per year were calculated for various

TABLE 2 DATA ON EFFICIENCIES AND SAVINGS FOR SINGLE THICKNESS COVERINGS

Covering No.	Kind of Covering	Temperature Difference (Pipe and Room)	Actual Temperature (Room = 80 deg. fahr.)	B.t.u. Loss/Sq.ft./Deg Temperature Difference/Hr.		B.t.u. Saving Due to Covering/Deg./Sq. ft./Hr.	Efficiency of Covering—Per Cent	Saving Due to Covering in B.t.u./Sq. ft./Yr.	Saving in $/Sq. ft./Yr.	First Cost of Covering/Sq. ft.	Cost of Covering/Sq. ft./Yr.	Net Saving in $/Sq. ft./Yr.	Interest on Investment
				Bare Pipe	Covered Pipe								
I	J-M 85% Magnesia	50	130	1.950	0.435	1.515	77.7	664,000	0.199	0.238	0.033	0.166	69.7
		100	180	2.152	0.438	1.714	79.6	1,502,000	0.451			0.418	175.5
		200	280	2.665	0.446	2.219	83.3	3,887,000	1.165			1.132	475
		300	380	3.260	0.455	2.805	86.1	7,370,000	2.211			2.178	914
		400	480	4.035	0.469	3.566	88.4	12,490,000	3.750			3.717	1560
		500	580	5.180	0.488	4.692	90.6	20,560,000	6.165			6.132	2575
II	J-M Indented	50	130	1.950	0.472	1.478	75.6	647,300	0.194	0.214	0.030	0.164	76.6
		100	180	2.152	0.483	1.669	77.6	1,462,000	0.438			0.408	190.5
		200	280	2.665	0.509	2.156	80.9	3,777,000	1.133			1.103	515
		300	380	3.260	0.549	2.711	83.2	7,120,000	2.136			2.106	983
		400	480	4.035	0.603	3.432	85.1	12,020,000	3.608			3.578	1676
		500	580	5.180	0.666	4.514	87.1	19,780,000	5.935			5.905	2760
III	J-M Vitribestos	50	130	1.950	0.626	1.324	67.9	580,000	0.174	0.381	0.053	0.121	31.8
		100	180	2.152	0.654	1.498	69.6	1,312,000	0.394			0.341	89.6
		200	280	2.665	0.715	1.950	73.2	3,417,000	1.025			0.972	255.0
		300	380	3.260	0.781	2.481	76.0	6,519,000	1.955			1.902	500.0
		400	480	4.035	0.858	3.177	78.8	11,130,000	3.340			3.287	865.0
		500	580	5.180	0.967	4.213	81.4	18,450,000	5.540			5.487	1442.0
IV	J-M Eureka	50	130	1.950	0.440	1.510	77.4	661,000	0.198	0.262	0.037	0.161	61.5
		100	180	2.152	0.451	1.701	79.0	1,490,000	0.447			0.410	156.6
		200	280	2.665	0.464	2.201	82.6	3,860,000	1.158			1.121	428.0
		300	380	3.260	0.478	2.782	85.4	7,310,000	2.192			2.155	824.0
		350	430	3.627	0.487	3.140	86.6	9,620,000	2.888			2.851	1090.0
V	J-M Molded	50	180	1.950	0.517	1.433	73.4	627,600	0.188	0.190	0.027	0.161	55.5
		100	180	2.152	0.522	1.630	75.8	1,428,000	0.428			0.401	211.0
		200	280	2.665	0.539	2.126	79.8	3,725,000	1.117			1.090	574.0
		300	380	3.260	0.561	2.699	82.8	7,088,000	2.126			2.099	1105.0
		400	480	4.035	0.596	3.439	85.2	12,050,000	3.615			3.588	1888.0
VI	J-M Wool-Felt	50	130	1.950	0.386	1.564	80.2	684,700	0.205	0.214	0.030	0.175	81.8
		100	180	2.152	0.400	1.752	81.4	1,535,000	0.460			0.430	201.0
		200	280	2.665	0.421	2.244	84.2	3,930,000	1.179			1.149	537.0
		300	380	3.260	0.442	2.818	86.4	7,400,000	2.220			2.190	1023.0
		350	430	3 627	0.453	3.174	87.6	9,730,000	2.919			2.889	1349.0
VII	Sall-Mo Expanded	50	130	1.950	0.409	1.541	79.0	674,600	0.202	0.248	0.035	0.167	67.3
		100	180	2.152	0.427	1.725	80.2	1,511,000	0.453			0.418	168.5
		200	280	2.665	0.464	2.201	82.6	3,856,000	1.156			1.121	452.0
		300	380	3.260	0.503	2.757	84.6	7,240,000	2.172			2.137	852.0
		400	480	4.035	0.541	3.494	86.6	12,240,000	3.672			3.637	1466.0
		500	580	5.180	0.581	4.599	88.8	20,140,000	6.042			6.007	2421.0
VIII	Carey Carocel	50	130	1.950	0.358	1.592	81.6	697,000	0.209	0.195	0.027	0.182	93.4
		100	180	2.152	0.378	1.774	82.4	1,555,000	0.467			0.440	226.0
		200	280	2.665	0.421	2.244	84.2	3,932,000	1.180			1.153	592.0
		300	380	3.260	0.466	2.794	85.7	7,340,000	2.202			2.175	1116.0
		400	480	4.035	0.510	3.525	87.4	12,350,000	3.705			3.678	1886.0
		500	580	5.180	0.562	4.618	89.2	20,220,000	6.066			6.039	3100.0
IX	Carey Serrated	50	130	1.950	0.454	1.496	76.7	655,000	0.197	0.214	0.030	0.167	78.0
		100	180	2.152	0.468	1.684	78.2	1,475,000	0.443			0.413	193.0
		200	280	2.665	0.506	2.159	81.0	3,782,000	1.135			1.105	517.0
		300	380	3.260	0.546	2.714	83.3	7,132,000	2.140			2.110	986.0
		400	480	4.035	0.587	3.448	85.4	12,075,000	3.622			3.592	1680.0
		500	580	5.180	0.634	4.546	87.8	19,900,000	5.970			5.940	2780.0

TABLE 2 DATA ON EFFICIENCES AND SAVINGS FOR SINGLE THICKNESS COVERINGS—CONCLUDED

Covering No.	Kind of Covering	Temperature Difference (Pipe and Room)	Actual Temperature (Room = 80 deg. fahr.)	B.t.u. Loss /Sq.ft./Deg Temperature Difference/Hr. Bare Pipe	Covered Pipe	B.t.u. Saving Due to Covering/Deg./Sq. ft./Hr.	Efficiency of Covering—Per Cent	Saving Due to Covering in B.t.u./Sq. ft./Yr.	Saving in $/Sq. ft./Yr.	First Cost of Covering/Sq. ft.	Cost of Covering/Sq. ft./Yr.	Net Saving in $/Sq. ft./Yr.	Interest on Investment
X	Carey Duplex	50	130	1.950	0.423	1.527	78.3	669,000	0.201	0.162	0.023	0.178	110.0
		100	180	2.152	0.447	1.705	79.2	1,494,000	0.448			0.425	252.0
		200	280	2.665	0.498	2.167	81.3	3,798,000	1.139			0.116	689.0
		300	380	3.260	0.548	2.712	83.2	7,128,000	2.140			0.117	1307.0
		350	430	3.627	0.574	3.053	84.2	9,360,000	2.808			2.785	1720.0
XI	Carey 85% Magnesia	50	130	1.950	0.413	1.537	78.8	673,000	0.202	0.200	0.028	0.174	87.0
		100	180	2.152	0.418	1.734	80.5	1,519,000	0.456			0.428	214.0
		200	280	2.665	0.424	2.241	84.1	3,929,000	1.179			1.151	576.0
		300	380	3.260	0.436	2.824	86.6	7,420,000	2.226			2.198	1099.0
		400	480	4.035	0.454	3.581	88.8	12,550,000	3.765			3.737	1869.0
		500	580	5.180	0.472	4.708	90.9	20,610,000	6.183			6.155	3078.0
XII	Sall-Mo Wool-Felt	50	130	1.950	0.395	1.555	79.8	681,000	0.204	0.190	0.027	0.177	93.3
		100	180	2.152	0.401	1.751	81.4	1,535,000	0.461			0.434	228.0
		150	230	2.400	0.421	1.979	82.5	2,600,000	0.780			0.753	396.5
		200	280	2.665	0.433	2.232	83.8	3,910,000	1.173			1.146	604.0
		250	330	2.951	0.445	2.506	84.9	5,484,000	1.645			1.618	852.0
		300	380	3.260	0.459	2.801	85.9	7,360,000	2.208			2.181	1150.0
XIII	Nonpareil High Pressure	50	130	1.950	0.399	1.551	79.5	679,000	0.204	0.224	0.031	0.173	77.2
		100	180	2.152	0.402	1.750	81.3	1,533,000	0.460			0.429	192.0
		200	280	2.665	0.412	2.253	84.6	3,950,000	1.185			1.154	516.0
		300	380	3.260	0.426	2.834	86.9	7,448,000	2.234			2.203	985.0
		400	480	4.035	0.444	3.591	89.0	12,580,000	3.774			3.745	1673.0
		500	580	5.180	0.465	4.715	91.0	20,640,000	6.190			6.159	2752.0
XIV	J-M Fire Felt	50	130	1.950	0.694	1.256	64.4	550,000	0.165			0.118	35.4
		100	180	2.152	0.711	1.441	67.0	1,262,000	0.379	0.333	0.047	0.332	99.7
		200	280	2.665	0.749	1.916	71.9	3,360,000	1.008			0.961	288.6
		300	380	3.260	0.795	2.465	75.6	6,480,000	1.944			1.897	570.0
		400	480	4.035	0.845	3.190	79.0	11,175,000	3.353			3.306	993.0
		500	580	5.180	0.901	4.279	82.6	18,740,000	5.620			5.573	1675.0
XV	J-M Sponge Felted	50	130	1.950	0.336	1.614	82.7	707,000	0.212	0.333	0.047	0.165	49.5
		100	180	2.152	0.347	1.805	83.8	1,581,000	0.474			0.427	128.0
		200	280	2.665	0.369	2.296	86.2	4,035,000	1.211			1.164	350.0
		300	380	3.260	0.391	2.869	88.0	7,540,000	2.262			2.215	665.0
		400	480	4.035	0.414	3.621	89.8	12,690,000	3.809			3.762	1132.0
		500	580	5.180	0.439	4.741	91.5	20,770,000	6.230			6.183	1860.0
XVI	J-M Asbestocel	50	130	1.950	0.418	1.532	78.5	671,000	0.201	0.262	0.037	0.164	62.7
		100	180	2.152	0.429	1.723	80.0	1,510,000	0.453			0.416	159.0
		200	280	2.665	0.454	2.211	83.0	3,876,000	1.163			1.126	430.0
		300	380	3.260	0.493	2.767	84.8	7,272,000	2.181			2.144	820.0
		400	480	4.035	0.544	3.491	86.5	12,230,000	3.670			3.633	1388.0
		500	580	5.180	0.609	4.571	88.2	20,020,000	6.006			5.969	2280.0
XVII	J-M Air Cell	50	130	1.950	0.459	1.491	76.4	653,000	0.196	0.190	0.027	0.169	89.0
		100	180	2.152	0.475	1.677	77.9	1,469,000	0.441			0.414	218.0
		200	280	2.665	0.515	2.150	80.7	3,769,000	1.130			1.103	581.0
		300	380	3.260	0.571	2.689	82.7	7,066,000	2.120			2.093	1101.0
		400	480	4.035	0.643	3.392	84.1	11,885,000	3.568			3.541	1865.0
		500	580	5.180	0.733	4.447	85.8	19,475,000	5.841			5.814	3060.0

thicknesses of J-M Sectional 85 Per Cent Magnesia at temperature differences of 100, 300 and 500 deg. fahr. This was done in exactly the same manner as already described for single thickness coverings. The variation of standard list prices with thickness is shown in Fig. 21. These values were used in computing the cost per year of the various thicknesses of covering and results are given in Table 3. The curves for net savings per square foot per year in Fig. 18 show rapid increase in savings as the thickness is increased up to a point of maximum savings after which there is a decrease owing to the rapid increase in cost of covering. Fig. 18 shows the proper thickness for the maximum net saving at any temperature difference from 0 to 500 deg. fahr.

69 The chart for proper thickness mentioned above applied only to the case where steam costs \$0.30 per 1000 lb. and is on 365 days in the year. Fig. 22 is a chart for proper thickness of magnesia covering to be used at any temperature, any price of steam and any number of hours service per year. The chart does not show values for length of service, but to use it for other periods than 365 days at 24 hours a day, multiply the price of steam by the number of hours per year the steam line considered is in service and divide by 8760 and, using the result as the price of steam on the chart, find the proper thickness.

70 For example, suppose that the steam pressure is 150 lb. per sq. in. gage, that it costs \$0.30 per 1000 lb. generated, and that the line is in use 12 hr. a day and 9 months out of the year. The number of hours per year that the steam is on is therefore 2920, or $\frac{1}{3}$ of the time. The price of steam to be used on the chart is 0.30 (2920/8760) = \$0.10. The temperature of the pipe containing steam at 150 lb. gage pressure will be about 365 deg. fahr., and assuming a room temperature of 80 deg., the temperature difference between pipe and room will be 285 deg. Now on the chart, using the curve for steam at \$0.10 per 1000 lb., the proper thickness corresponding to 285 deg. temperature difference is found to be 1.5 in. This then is the proper thickness for maximum net saving under the given conditions.

MATHEMATICAL TREATMENT OF HEAT FLOW IN INSULATING MATERIALS

71 The problem of insulating objects against the flow of heat is one which, when the necessary constants are known, is capable of very complete mathematical solution. In order to obtain experimental data on all thicknesses of coverings that might be used, a

very large number of tests would have to be made. Therefore, when mathematical relations have been established, and these proven by experiment to be correct, the making of an almost endless number of tests is entirely unnecessary. The mathematical treatment cannot take the place of all experimental work, since the conductivities of the materials must be determined by actual experiment; but once these are known, further tests are not required for the accurate determination of the losses from any thickness whatsoever.

72 In the case of flat surfaces, where the two boundaries of the

TABLE 3 DATA ON EFFICIENCIES AND SAVINGS OF VARIOUS THICKNESSES OF 85 PER CENT MAGNESIA COVERING

Temperature Difference	Thickness	B.t.u./Sq.ft./deg.dif./hr.			Saving	Efficiency	Total Saving/yr.		First Cost of Covering	Cost of Covering/yr.	Net Saving
		Bare Pipe	Plastic 85 per cent Magnesia	Sectional 85 per cent Magnesia			B.t.u.	Dollars (Steam at 30c/1000 lb.)			
100	0.5	2.152	0.735	0.691	1.461	67.8	1,280,000	0.384	0.125	0.018	0.366
100	1.0		0.492	0.462	1.690	78.4	1,481,000	0.444	0.238	0.033	0.411
100	2.0		0.319	0.300	1.852	85 5	1,622,000	0.486	0.528	0.074	0.412
100	3.0		0.248	0.233	1.919	89.1	1,681,000	0.504	0.851	0.119	0.385
100	4.0		0.209	0.196	1.956	90.8	1,714,000	0.514	1.215	0.170	0.344
100	5.0		0.185	0.174	1.978	91.9	1,733,000	0.520	1.685	0.236	0.284
300	0.5	3.260	0.805	0.757	2.503	76.8	6,579,000	1.975	0.125	0.018	1.957
300	1.0		0.524	0.493	2.767	84.9	7,270,000	2.181	0.238	0.033	2.148
300	2.0		0.335	0.315	2.945	90.4	7,740,000	2.322	0.528	0.074	2.248
300	3.0		0.260	0.244	3.016	92.5	7,922,000	2.377	0.851	0.119	2.258
300	4.0		0.219	0.206	3.054	93.7	8,026,000	2.408	1.215	0.170	2.238
300	5.0		0.192	0.181	3.079	94.4	8,090,000	2.427	1.685	0.236	2.191
500	0.5	5.180	0.895	0.842	4.338	83.7	19,000,000	5.700	0.125	0.018	5.682
500	1.0		0.557	0.524	0.656	89.9	20,410,000	6.125	0.238	0.033	6.092
500	2.0		0.350	0.329	4.851	93.6	21,260,000	6.380	0.528	0.074	6.306
500	3.0		0.273	0.257	4.923	95.0	21,570,000	6.470	0.851	0.119	6.351
500	4.0		0.229	0.215	4.965	95.8	21,750,000	6.525	1.215	0.170	6.355
500	5.0		0.199	0.187	4.993	96.4	21,880,000	6.560	1.685	0.236	6.324

insulating material are parallel planes, the application of the mathematical treatment is very simple. In this case the quantity of heat conducted is given by the equation

$$Q = k\frac{\theta_1 - \theta_2}{x} A\, t \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

in which Q is the quantity of heat conducted, θ_1 and θ_2 the temperatures of hotter and colder surfaces respectively, x is the thickness in in. of the layer of material, A is the area in sq. ft. of the surface considered, t is the time in hr. and k is the conductivity of the material in B.t.u. per degree temp. difference per sq. ft. per in. thickness per hr. The reason for strange mixture of feet and inches in the equation is the irrational definition, in the English units, of conductivity, since in it the square foot is made the unit of area and the inch the unit of thickness.

73 The tests of pipe coverings now under discussion involved

TABLE 4 DATA ON NET SAVINGS AT VARIOUS THICKNESSES OF 85 PER CENT MAGNESIA COVERING FOR STEAM AT \$0.10 AND \$0.20 PER 1000 LB.

Temperature Difference	Thickness	Total Saving in \$ (Steam at 10c./1000 lb.)	First Cost of Covering	Cost of Covering/yr.	Net Saving \$/yr.	Temperature Difference	Thickness	Total Saving in \$ (Steam at 20c./1000 lb.)	First Cost of Covering	Cost of Covering/yr.	Net Saving \$/yr.
100	0.5	0.128	0.125	0.018	0.110	100	0.5	0.256	0.125	0.018	0.236
100	1.0	0.148	0.238	0.033	0.115	100	1.0	0.296	0.238	0.033	0.263
100	2.0	0.162	0.528	0.074	0.088	100	2.0	0.324	0.528	0.074	0.250
100	3.0	0.168	0.851	0.119	0.049	100	3.0	0.336	0.851	0.119	0.217
100	4.0	0.171	1.215	0.170	0.001	100	4.0	0.342	1.215	0.170	0.172
100	5.0	0.173	1.685	0.236	0.063	100	5.0	0.346	1.685	0.236	0.110
300	0.5	0.658	0.125	0.018	0.640	300	0.5	1.316	0.125	0.018	1.298
300	1.0	0.727	0.238	0.033	0.694	300	1.0	1.454	0.238	0.033	1.421
300	2.0	0.774	0.528	0.074	0.700	300	2.0	1.548	0.528	0.074	1.474
300	3.0	0.792	0.851	0.119	0.673	300	3.0	1.584	0.851	0.119	1.465
300	4.0	0.803	1.215	0.170	0.633	300	4.0	1.606	1.215	0.170	1.436
300	5.0	0.809	1.685	0.236	0.573	300	5.0	1.618	1.685	0.236	1.382
500	0.5	1.900	0.125	0.018	1.882	500	0.5	3.800	0.125	0.018	3.782
500	1.0	2.040	0.238	0.033	2.007	500	1.0	4.080	0.238	0.033	4.047
500	2.0	2.126	0.528	0.074	2.052	500	2.0	4.252	0.528	0.074	4.178
500	3.0	2.157	0.851	0.119	2.038	500	3.0	4.314	0.851	0.119	4.195
500	4.0	2.175	1.215	0.170	2.005	500	4.0	4.350	1.215	0.170	4.180
500	5.0	2.188	1.685	0.236	1.952	500	5.0	4.376	1.685	0.236	4.140

cylindrical surfaces rather than flat ones, so that a more extensive explanation will be made of this class of problems. The rate of heat flow is given by the equation

$$W_1 = \frac{k\,(\theta_1 - \theta_2)}{r\,(\log_e r_2 - \log_e r_1)} \qquad [2]$$

in which W_1 is the rate of heat flow per sq. ft. of pipe per hr.; k is the conductivity; θ_1 and θ_2 are the temperatures; r_1 and r_2 are the radii of the inner and outer surfaces of the covering respectively, and r is the outside radius of the pipe. The derivation of this equation is not nearly so simple as that of equation [1], since it consists of the application of Fourier's conduction equation to the steady state with flow of heat in two directions; but the entire equation is rational and contains nothing of empirical nature except the conductivity, k, which must be determined by experiment.

74 The data obtained from the tests just described form an excellent basis for the calculation of these conductivities. The values of B.t.u. loss per degree temperature difference between pipe surface and air per square foot per hour cannot be taken at once as the conductivities because the thicknesses were not exactly 1 in., and in the case of conductivity the temperature difference considered is that between inner and outer surfaces of covering. The conductivity may then be calculated by solving equation [2] for k.

$$k=\frac{W_1 r\ (\log_e r_2-\log_e r_1)}{\theta_1-\theta_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

In this equation everything is known except θ_2. Now the loss per square foot of outside surface of the covering, W_2, is a function of the temperature difference between the outside surface of the covering and the air in the room.

That is

$$W_2=F\ (\theta_2-\theta_r) \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

or

$$\theta_2-\theta_r=f\ (W_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

and

$$\theta_2=\theta_r+f\ (W_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

$$k=\frac{W_1 r\ (\log_e r_2-\log_e r_1)}{\theta_1-\theta_r-f\ (W_2)} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

$$W_2=\frac{r_1}{r_2}W_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots [8]$$

75 The value of $f\ (W_2)$ cannot easily be expressed mathematically, and, as it stands, the above transcendental equation [7] cannot be solved. However, in order that $f\ (W_2)$ might be evaluated a series of tests was made in which the relation between losses per square foot of outer surface of covering per hour and the difference in temperature between that of outside of covering and that of room was ascertained. Six different coverings of the most diverse conductivities were tested, and the results from all six were essentially

the same, which shows that the rate of heat loss from a surface in contact with air depends upon the character of the surface and the temperature difference between the surface and the air, and not directly upon the conductivity of the material beneath the surface. But the temperature difference just mentioned depends upon the conductivity, since the heat that is lost must come through the covering; therefore the temperature of the surface will be maintained at some point just high enough above the room temperature to bring about the dissipation of the given amount of heat. The above must not be misconstrued to mean that the character of material has no effect on the amount of heat that will pass through the covering, because the temperature difference between the surface of the covering and the surrounding air will be higher for the ones losing the greater amounts of heat and vice versa. The data for these tests appear in Table 5 and the same are plotted in the form of a curve in Fig. 23.

76 Some of the above tests were made on a pipe heated by steam and some on one heated by electricity, and no difference whatsoever was observed in the results. This shows conclusively that what a pipe contains makes absolutely no difference in the amount of heat lost, provided the temperature of the pipe surface is the same in each case.

77 The curve in Fig. 23 is perhaps the most valuable of all the results contained in this paper. It not only removes the principal obstacle in the way of mathematical treatment of heat insulation problems, but it forms also a new basis for approximate tests on pipe coverings. In order to find out the losses from any pipe covering having its surface finished with white canvas, all that is necessary is to place a thermometer under the canvas and another in the air 4 or 5 ft. from the pipe; take the difference between the two temperature readings and on the curve find the corresponding loss. Such a test might give results as much as 5 per cent in error due to the chances of not getting the average temperature difference any closer than that, but at that it would be accurate enough for some purposes.

78 The investigation of the relation of losses from covering surfaces and temperature difference between such surfaces and the surrounding air throws some light on the question of the effect of finishing insulated surfaces with planished iron, etc., which question has been much discussed and little understood up to the present. The planished surface would radiate less heat at the same tem-

perature than would a dull canvas surface, but since more heat would come through the covering than it could radiate at that temperature the surface would heat up to a higher temperature than if it were canvas; and as it heated up the temperature difference between inner and outer surfaces of covering would be correspondingly decreased due to the heating up of the cooler one. Therefore, it is quite apparent from equation [2] that there would be a decrease in the amount of heat transmitted. The resulting conditions then would be a higher surface temperature and less heat transmitted in the case of the planished surface than for canvas surface.

79 Below is given an example of the computation of the conductivity, k. The material is J-M 85 Per Cent Magnesia; temperature difference between pipe surface and air is 300 deg. fahr.; thickness of covering, 1.13 in.; outside diameter of pipe, 5.6 in. The rate of heat loss per degree temperature difference per square foot per hour is found from Fig. 15 to be 0.455 B.t.u. Therefore, W_1, the total loss per sq. ft. per hr., is equal to $300\times0.455=136.5$ $W_2=136.5\times2.8\div(2.8+1.13)=97.2$ B.t.u.

80 From the curve, Fig. 23, the temperature difference between outer covering surface and air corresponding to a loss of 97.2 B.t.u. is 65 deg. Therefore, the temperature difference between inner and outer covering surfaces is $(300—65)=235$ deg.

$$k=\frac{W_1\, r\ (\log_e r_1-\log_e r_2)}{\theta_1-\theta_2}$$

$$=\frac{136.5\times2.8\ (\log_e 3.93-\log_e 2.8}{235}$$

$$=0.551$$

The conductivity of J-M 85 Per Cent Magnesia at 300 deg. temperature difference is therefore 0.551 B.t.u. per deg. temp. difference per sq. ft. per in. thickness per hr.

81 Conductivities have been calculated at 300 deg. temp. difference (pipe temperature—room temperature) for all of the coverings tested. The values are given in Table 6.

82 Table 6 is the best basis on which to compare coverings, because here all differences due to different thicknesses are done away with and the coverings may be compared under exactly the same conditions. If conductivities at other temperatures are desired, they may be calculated from the data in Fig. 15 and using the curve Fig. 23.

83 For finding the heat loss through any thickness of any material of which the conductivity is known, and at any tempera-

TABLE 5 DATA ON RELATION OF HEAT LOSSES TO TEMPERATURE DIFFERENCE BETWEEN COVERING SURFACE AND SURROUNDING AIR

	Date 1915	Thermo-Couple Potential—Mv.	Temperatures of Outside of Pipe	Temperatures of Room	Temperatures of Difference Pipe and Room	Loss/Deg. Dif./Sq. ft./Hr.-B.t.u.	Total Loss/Sq. ft. of Pipe Hr.	Total Loss/Sq. ft. Outer Surface of Covering	Temperature of Outside of Covering	Temp. Difference Covering and Room	Means of Heating
J-M 85% Magnesia	Aug. 2	4.53	224	86.7	137.3	0.441	60.6	43.2	119	32.3	Steam
	Aug. 2	5.69	268	84	184	0.444	81.7	58.6	126.7	42.7	Steam
	Aug. 2	6.58	301	83	218	0.447	97	69.1	131.5	48.7	Steam
	Aug. 3	7.28	326.5	83.2	243.3	0.449	102.3	77.9	137.5	54.3	Steam
	Aug. 3	8.05	354.5	88.2	266.3	0.451	120	85.5	147	58.8	Steam
J-M Asbestos Sponge Felted	Aug. 2	4.53	224	86.7	137.3	0.355	48.7	34.5	113.7	27	Steam
	Aug. 2	5.69	268	84	184	0.365	67.1	47.5	120.2	36.6	Steam
	Aug. 2	6.58	301	83	218	0.372	81.1	57.4	124.6	41.6	Steam
	Aug. 3	7.28	326.5	83.2	243.3	0.378	92	65.1	131.2	48	Steam
	Aug. 3	8.05	354.5	88.2	266.3	0.383	102	72.2	139.7	57.5	Steam
	Aug. 6	4.21	211	79	132	0.353	46.6	32.9	108	25	Electric current
	Aug. 7	9.85	416	80	336	0.398	133.7	94.6	145	65	Electric current
	Aug. 7	12.12	490	84	406	0.415	168.5	119.2	162	78	Electric current
	Aug. 7	14.46	562	84	478	0.432	206.5	146	175	91	Electric current
J-M Asbestocel	Aug. 2	4.53	224	86.7	137.3	0.438	60.2	43.2	118.3	31.6	Steam
	Aug. 2	5.69	268	84	184	0.451	83.0	59.6	125	41	Steam
	Aug. 2	6.58	301	83	218	0.461	100.5	72.1	133	53	Steam
	Aug. 3	7.28	326.5	83.2	243.3	0.470	114.4	82.1	139.5	56.3	Steam
	Aug. 3	8.05	354.5	88.2	266.3	0.479	127.5	91.5	149.2	61	Steam
	Aug. 6	4.21	211	79	132	0.437	57.6	41.4	109	30	Electric current
	Aug. 6	7.73	343	80	263	0.478	125.7	90.2	139	59	Electric current
	Aug. 7	9.85	416	80	336	0.510	171.4	123	156.5	76.5	Electric current
	Aug. 7	12.12	490	84	406	0.548	222.5	159.8	178	94	Electric current
	Aug. 7	14.46	562	84	478	0.593	233.5	203.5	198	114	Electric current
J-M Fire Felt	Aug. 6	4.21	211	79	132	0.722	95.3	68.5	127	48	Electric current
	Aug. 6	7.73	343	80	263	0.777	204.3	147	167	87	Electric current
	Aug. 7	9.85	416	80	336	0.812	273	196.4	187	107	Electric current
	Aug. 7	12.12	490	84	406	0.845	344	247.5	209	125	Electric current
	Aug. 7	14.46	562	84	478	0.889	425	305.8	228	144	Electric current
J-M Air-Cell	Aug. 6	4.21	211	79	132	0.487	64.2	46	112.5	33.5	Electric current
	Aug. 6	7.73	343	80	263	0.548	144	103	144	64	Electric current
	Aug. 7	9.85	416	80	336	0 596	200	143.2	166.5	86.5	Electric current
	Aug. 7	12.12	490	84	406	0.648	263	188.2	191	107	Electric current
	Aug. 7	14.46	562	84	478	0.713	341	244	212	128	Electric current

ture difference between pipe and room up to 500 deg. fahr., a modification of equation [2] may be used.

$$W_1 = \frac{k[\theta_1 - \theta_r - f\,(W_2)]}{r_1\;(\log_e\,r_2 - \log_e\,r_1)} \dots\dots\dots\dots [9]$$

This equation involves two unknown quantities, and while it may be expressed entirely in terms of one of them,

$$W_2 = \frac{k\,[\theta_1 - \theta_r - f\,(W_2)]}{r_2\;(\log_e\,r_2 - \log_e\,r_1)} \dots\dots\dots\dots [10]$$

the expression is still not capable of solution by the ordinary mathematical operations, since the function of W_2 is one not easily expressed in mathematical terms. However, equation [10] may be solved readily by trial, using the curve in Fig. 23 as the means of evaluating $f\,(W_2)$.

84 For example, let it be required to find the heat loss per square foot of pipe surface per degree temperature difference, between pipe surface and air, per hour if a pipe of 5.6 in. outside diameter is covered with 3 in. thickness of a material whose conductivity is 0.587, and is maintained at a temperature of 380 deg. fahr. when room temperature is 80 deg. Then from equation [10]

$$W_2 = \frac{0.587\;\;[380 - 80 - f\,(W_2)]}{(2.8+3)\,(\log_e\,5.8 - \log_e\,2.8)}$$

$$= \frac{0.587}{4.225}\,[300 - f\,(W_2)]$$

Now assume that $f\,(W_2) = 20$ deg. Then W_2 from Fig. 23 = 25.5 B.t.u. But W_2 from equation [10] = 0.139 (300—20) = 38.9 B.t.u. The lack of agreement between these values of W_2 shows that $f\,(W_2)$ should have been taken at a value greater than 20 deg.

85 $f\,(W_2) = 30$ gives, from Fig. 23, $W_2 = 39.5$ B.t.u. and from equation [10], $W_2 = 0.139$ (300—30) = 37.5 B.t.u. This time $f\,(W_2)$ was taken a little too large, and a few more trials show the correct value to be 28.8 deg. Then $W_2 = 0.139$ (300—28.8) = 37.7 B.t.u. and from equation [8] $W_1 = (5.8 \div 2.8) \times 37.7 = 78.1$ B.t.u Loss per square foot per degree temperature difference between pipe surface and air in room, per hour = 78.1 ÷ 300 = 0.260 B.t.u.

86 The trial solutions as outlined above may be made very quickly with the aid of a slide rule, and the results obtained are quite as accurate as if the solutions were made by use of the ordinary processes of mathematics.

CONCLUSION

87 In conclusion it may be said that in most cases it pays to

use the best commercial pipe covering obtainable; because where the material is paid for many times over during the first year by the saving effected by its use, the first cost loses much of its weight as a determining factor in the selection of type of covering to be used. In view of the results of the thickness tests, it is a deplorable fact that few steam lines at the present time are provided with thick enough a covering for the greatest net saving. However, where fuel is cheap and the lines are in use only a small percentage of the time, the cheaper coverings have their advantages. Also there are places,

TABLE 6 CONDUCTIVITIES OF PIPE COVERINGS AT 300 DEGREES TEMPERATURE DIFFERENCE BETWEEN PIPE SURFACE AND ROOM

No.	Kind of Covering	Conductivity
I	J-M 85 Per Cent Magnesia	0.551
II	J-M Indented	0.686
III	J-M Vitribestos	1.087
IV	J-M Eureka	0.549
V	J-M Molded Asbestos	0.778
VI	J-M Wool Felt	0.521
VII	Sall-Mo Expanded Asbestos	0.598
VIII	Carey Carocel	0.540
IX	Carey Serrated	0.682
X	Carey Duplex	0.636
XI	Carey 85 Per Cent Magnesia	0.546
XII	Sall-Mo Wool Felt	0.510
XIII	Nonpareil High Pressure	0.543
XIV	J-M Asbestos Fire Felt	1.093
XV	J-M Asbestos Sponge Felted	0.468
XVI	J-M Asbestos Cell	0.596
XVII	J-M Air Cell	0.718
XVIII to XXII	Plastic 85 Per Cent Magnesia	0.587
XXIV	Sall-Mo Air Cell	0.802

as on some heating systems, where the heat lost through the coverings is not wasted and the object of covering the pipes at all is to keep tunnels, etc., cool enough that men may work in them. Therefore, a careful study of conditions is necessary before a certain type of covering can be recommended for a given piece of work. However, it is hoped that the data given in this paper will be of assistance to engineers in deciding upon the material to be used under certain conditions, and in calculating heat losses on installations already in use.

88 The durability of materials used for pipe coverings is a very important factor in determining the most economical covering for a given set of conditions. It has already been pointed out that the

proper basis for comparing costs was the cost per year and not the first cost of the material. This is true because the covering giving the greatest length of service for a given first cost and efficiency is obviously the one to select. In general, fibrous coverings are more durable than the molded forms; since the latter tend to revert to their original powdered state due to vibration and rough usage, while those made of fibrous material firmly felted together show no such tendency.

89 The author wishes to acknowledge the valuable assistance in the way of advice and suggestions of Prof. William Black, Prof. L. R. Ingersoll, and Prof. E. M. Terry of the University of Wisconsin, and Prof. A. G. Christie of Johns Hopkins University.

DISCUSSION

Leonard Waldo. I doubt whether even the author is aware of the great importance of his timely discussion of this subject. Of late years there have developed new uses for steam under pressure developing mechanical power at distances from the point of steam generation; for instance, the application of steam to the atomization of oil in open hearth furnaces, where it is necessary to retain the full pressure power of the steam, and where it has usually to be conducted through transmission lines over long distances. In this use it is essential that no loss of terminal pressure power takes place from heat losses in the steam with consequent pressure lowering at the colder steam exits.

In this paper the very high insulating materials are not referred to. Infusorial earth, used at times in the insulation of large furnaces, has a transmission power of $\frac{1}{4}$ that of the magnesia used in the tests described. There are better insulating materials considered from the insulating standpoint alone than those mentioned in the paper, but the ability to transfer from the data given here to the conditions of practice in individual cases is of great value.

F. M. Farmer. At the Electrical Testing Laboratories we recently completed a series of tests of this character on about 30 pipe coverings. We used the Stott method, which I think the author will agree is of the same accuracy as the method he used and much simpler, although the matter of making extended tests is a question of having the necessary large current available. A 2-in. bare pipe heated to a temperature of only about 350 deg.

required approximately 1500 amperes, showing that a large current source is one of the obstacles to the use of this method for the larger sizes of pipe.

Only one of our coverings can be assumed identical with one of those tested by Mr. McMillan. In that one case we obtained the same so-called efficiencies within about one per cent. However, our actual values on both the bare pipe and the covered pipe were of the order of 10 or 12 per cent higher, a difference which we ascribed to the size of pipe used. We made the bare test on a 2-in. pipe and the covered test on a 3-in. pipe.

Dr. Kennelly, in some of his work on heat losses from very small rods, quoted Professor Boussinesq as concluding that the connection losses varied inversely with the square root of the diameter. If that law holds for large diameters, it would fully account for the difference between the author's results and ours.

Herbert N. Dawes. This paper indicates much careful work and observation and contributes some very valuable data.

The question of the permanency of the various coverings is touched upon but lightly. Apparently no tests have been made upon coverings after they have been in actual service for some time. Some tests of this sort were made by Prof. C. L. Norton several years ago at the Massachusetts Institute of Technology. Sôme sections of 85 per cent magnesia covering which had been on a steam line in a tunnel for eight years were tested and found to be a fraction of 1 per cent more efficient than the new covering of the same make and thickness. This difference was probably due to the extreme dryness of the sample which had been so long in use. Other coverings which had been in service for a number of years were tested and their efficiency had dropped varying amounts.

In twenty years' experience with insulating materials of various kinds, I have seen some coverings which have deteriorated very much after a few years' service, in some cases so much so that there was not enough material left upon which to make any test.

The temperatures frequently used today in superheated steam lines are sufficient to drive the water of combination out of asbestos fiber, and of course make the use of wool fiber impossible. The effect of these temperatures on the asbestos fiber is to cause a breaking up and powdering, particularly in the presence of pipe vibration. Another practical point that should be considered is the effect of moisture from steam leaks or submersion. Along this line

I have found that coverings of molded form, such as 85 per cent magnesia, have been less damaged at least as regards future efficiency, than those made up of wool or asbestos fiber. I have therefore to take issue with Mr. McMillan on his statement that fibrous coverings are the more permanent. They certainly do not compare favorably with 85 per cent magnesia covering.

Another point of great importance to be considered in selecting a pipe covering is the possibility of re-use, either for application in the same sectional form or in the form of plastic. Fibrous coverings, for instance, if once badly damaged, cannot be used again. On the contrary, some of the molded coverings such as 85 per cent magnesia can be pounded up and reapplied as plastic, and are thus more or less permanent.

A factor affecting the efficiency of a pipe covering is the care of application. It has been found that a covering carefully applied and the same covering applied in a casual manner may show a difference of over one per cent in efficiency.

The proportion of surfaces to be covered which must be treated with a plastic rather than a sectional covering is a factor of interest. With 85 per cent magnesia covering, the plastic is of the same composition as the sectional, and hence of practially equal insulating value. When other types of covering are used, particularly those of a fibrous nature, the plastic has to be of a different composition, and this affects the efficiency of the entire insulation equipment in a substantial manner.

As the difference in nonconducting efficiency of several of the better coverings as shown by Mr. McMillan's figures is very slight, it seems some of the practical points I have spoken of are really of more importance in determining the kind of covering to use on most steam installations.

A large number of the coverings tested by the author seem to be the manufacture of one concern. There are at least five different makes of 85 per cent magnesia covering on the market, and in these tests but two were measured.

It would be interesting to know whether the author has made any investigation of the temperature drop in superheated steam lines at different velocities. Some reliable information on this subject would be most useful in solving insulation problems in connection with superheated steam insulation.

Arthur M. Greene, Jr. (written). Last year we performed experiments on 85 per cent magnesia and Nonpareil High Pressure covering and found a slight advantage for the latter, as the author has found, but for our range from about 220 to 320 deg. fahr., we found a much smaller change in the value of k. Our results gave the value of k as a constant.

It must be remembered that if k varies with the temperature difference, the expression

$$Q = k\frac{\theta_1 - \theta_2}{x} A\ t$$

gives on solution values of k which is now not the coefficient of conduction but the value of $Q\ x \div (\theta_1 - \theta_2)\ A\ t$. This quantity may be used in computing heat losses if its value is known for different values of temperature differences. It is of distinct value, but it is not a true constant of conduction. The value of the constant might be found by using an expression of the form

$$Q = k_1\ f\left(\frac{d\theta}{dx}\right)\frac{d\theta}{dx}\ A\ t$$

If this could be investigated, valuable information might be obtained.

Equation [2] is apparently derived from

$$Q = k\ \frac{d\theta}{dr} 2\pi r$$

but it might be well to state the origin of the expression. I think the term r outside the bracket should have a subscript and should be r_2, and that W_1 should be W_2, since it corresponds to the heat through the area at the outside. If it is the heat at the inside, r should be r_1.

The writer has not checked over the cost problems and monetary efficiencies, but would like to point out the importance of remembering that these results only hold for the assumed cost data and assumed rates of interest, depreciation, taxes and insurance.

It is hoped that the author may carry his investigations one step farther and use one and the same covering on from three to five different diameters of pipe, with say three different thicknesses of covering on each. There may be certain relations between the inner and outer surfaces at different radii which may affect the result. If this is not necessary, then the simpler methods used by Knoblauch or by Nusselt with flat discs of materials would be the better way to determine values of the quantity k.

L. R. INGERSOLL[1] (written). I am very glad to note that the author has applied the mathematical formulæ for heat conduction in his paper, for by the judicious use of theory in connection with experimental results, a piece of work of this sort is made vastly more effective than otherwise. It is not practicable to cover by experiment all the possible variations of size, thickness, etc., which enter into a problem of this kind, and the use of such well-grounded formulæ as those of head conduction to fill the gaps left by experiment and to extend the results is eminently desirable.

As I have pointed out in a former paper[2] it is a little unfortunate that the engineers have adopted a unit for the measurement of heat conductivity which—because of its inconsistency in using two different units of length, i.e. the inch and (sq.) foot—makes it difficult to apply heat conductivity formulæ to any but the simpler cases. With the general adoption of the metric units will come an incentive to a wider application of theory to experiment along these lines than has existed heretofore.

C. M. SAMES.[3] The values in the curve of Fig. 23 may be obtained from the empirical equation

$$f\,(W_2) = \frac{328\,W_2 - 220}{W_2 + 390}$$

Substituting this value of $f(W_2)$ in the equation of Par. 84, W_2 may be obtained directly by solving the resulting quadratic, instead of by the trial-and-error method of Par. 85.

THE AUTHOR. Replying to Dr. Waldo's remarks, the author believes that it would be highly desirable to have tests made on infusorial earth and other materials that are used in preventing loss of heat from hot blast stoves. However, it would be very remarkable if any of these were found to have four times the insulating value of 85 per cent magnesia.

As Mr. Farmer says, the difference in size of pipe as tested by him from that used in the tests described might account entirely for the larger values of losses he obtained. This can be demonstrated mathematically. With 1 in. of covering the 5-in. pipe would lose only 85 per cent as much heat as the 2-in. and only 92 per cent as much as the 3-in., other conditions being the same. Therefore, his results are in almost perfect agreement with those in the paper.

[1]Assoc. Prof. of Physics., Univ. of Wisconsin, Madison, Wis.
[2]Eng. News, October 30, 1913.
[3]542 Bramhall Ave., Jersey City, N. J.

In reply to Mr. Dawes, at the present time it is contemplated to collect data on the durability of various coverings, by correspondence with those who have had them in use for a number of years.

The reason for not testing more magnesia coverings was the very close agreement of the two which were tested. It was decided then that magnesia covering made up properly would be about alike for all manufacturers, and the author considered that it would be a waste of time and money to test all.

In reference to the temperature drop in superheated steam pipes at different velocities, the author can only refer Mr. Dawes to the paper by Eberle, the reference to which is given in this paper. Eberle's results are probably the best yet published on the subject, but the conditions under which they were obtained were not varied enough to make them generally applicable.

Professor Greene's discussion referring to the factor k is of interest in a mathematical discussion. However, the fact that k has different values at different temperatures is not sufficient justification for saying that it is not the conductivity of the material in question. Conductivity is defined as being the rate of heat flow per degree temperature difference per unit area per unit thickness per unit time. k, as used in the paper, conforms to this definition, and the fact that it is not constant but varies with the temperature has already been demonstrated by Nusselt as explained in the paper.

Professor Greene is correct as to the origin of the equation giving W_1. However, since it applies to unit area, the term $2\pi r$ does not appear and the fundamental equation is

$$\frac{Q}{2\pi r} = W = k \frac{d\theta}{dr}$$

The subscripts are correct according to the definitions given to the various factors.

The objection to the methods used by Knoblauch and Nusselt with flat discs instead of pipes as the covered surfaces is that such methods do not permit of the testing of commercial pipe coverings. But where the conductivities of the materials used have been accurately determined by those methods, the results may be made applicable to actual pipe covering conditions, as explained in the mathematical treatment of the subject.

The great truth of Professor Ingersoll's remarks on the advantages of the metric units over the more unwieldy ones now so commonly used by engineers will be appreciated by all those who have attempted to apply theory to the more complicated physical phenomena.

No. 1515

PERFORMANCE AND DESIGN OF HIGH VACUUM SURFACE CONDENSERS

BY

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AND

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The coefficient of heat transmission in an experimental condenser filled with pure steam appears from recent investigations to be a determinate quantity dependent upon steam temperature, water velocity, diameter of tube and mean water temperature and its value is of the order of 500 to 1500 B.t.u. per sq. ft. per hr. per deg. fahr. difference. In the commercial condenser the high rate of transmission undoubtedly existing for certain parts of the surface is masked by the large amount of relatively idle surface, so that the average coefficient for all the surface figures out much lower—sometimes as low as 50 B.t.u. The size and cost of a condenser to maintain a certain vacuum depend primarily on the extent of the zone of active condensation, that is, on how much of the surface does work and how much is idle. As will be shown, the problem is one of hydrodynamics as well as of heat transmission.

2 These variations being commonly attributed to the presence of air and to imperfect steam distribution and penetration, modern high vacuum surface condensers are equipped with large capacity air pumps and are designed with liberal areas for the flow of steam through the tube bank, with the minimum depth of tube bank between inlet and outlet. When careful attention is also given to excluding air in leakage, high vacuums are successfully maintained. However, a condenser, which will give a high coefficient of heat

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

transfer when the test readings are taken under summer conditions with circulating water at temperatures or 70 to 80 deg. fahr., will also give relatively low values in winter with cold water.

COMPARISON OF SUMMER AND WINTER PERFORMANCES

3 A 25,000 sq. ft. two-pass condenser with reciprocating dry air pump showed the following values of the average heat transmission coefficient.[1]

Mean Water Temperature	Vacuum Referred to 30-in. Barometer	Coefficient of Transmission
41	28.85	164
80	28.00	392

4 Tests of two large surface condensers with rotary hurling water air pumps showed heat transmissions as follows:[2]

Mean Water Temperature	Vacuum Referred to 30-in. Barometer	Coefficient of Transmission
46	29.25	284
73	28.54	580

5 Observations were recently taken on a surface condenser with turbo air pump, with constant load and variable condensing water temperature and vacuum. The differences of steam and mean water temperature are plotted against mean water temperature in Fig. 1. The warmer the water and the lower the vacuum, the smaller the mean temperature head required to transmit practically constant amount of heat through the surface.

DEPRESSION OF AIR PUMP SUCTION TEMPERATURE AS AN INDEX OF SURFACE EFFICIENCY

6 Examination of the results of a number of tests on commercial condensers, a vacuum evaporator, and a laboratory condenser indicates a definite relation, Figs. 2 to 9, between the coefficient of heat transfer and the difference between the steam temperature at the condenser inlet and the air temperature at the outlet.

[1]Tests on exhaust steam turbine at 59th St. Power Station of Interborough Rapid Transit Co., New York City, H. G. Stott and R. J. S. Pigott, Transactions Am. Soc. M. E., vol. 32, p. 69.

[2]Condensers and Their Auxiliaries, R. N. Ehrhart, Assoc. Iron and Steel Elect. Engrs., 1914.

7 The water velocity, mean water temperature and tube diam-
ter affect the rate of transmission from the tube wall to the water,

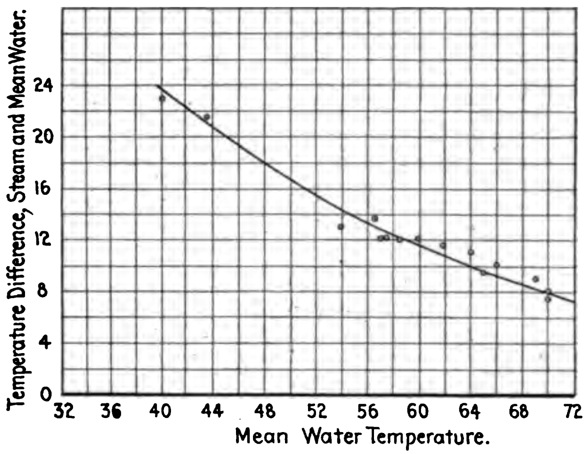


FIG. 1 CHART INDICATING INCREASE OF TEMPERATURE DIFFERENCE AND CORRESPONDING DECREASE OF COEFFICIENT OF HEAT TRANSFER WITH DECREASE IN TEMPERATURE OF CIRCULATING WATER

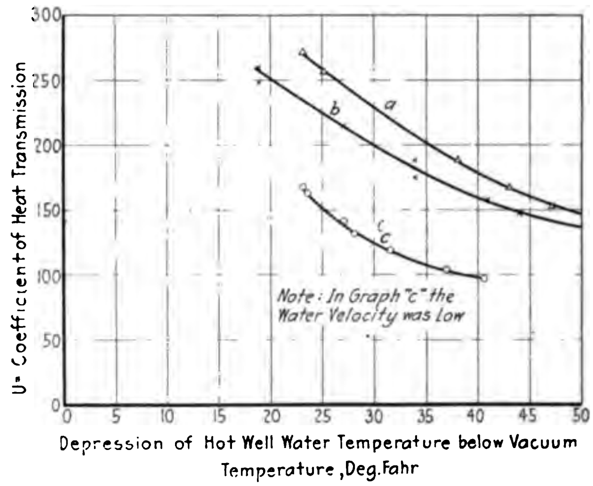


FIG. 2 RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION, CONDENSERS *a*, *b*, *c*

hat is, the transmission on the *water side* of the tube, and their in-
dividual influences have been determined in experimental single-
tube condensers, so that corrections for these variables can be ap-

plied with a reasonable degree of certainty. Geo. A. Orrok's results, Figs. 21 and 22, have been used for the present purpose so far as possible.

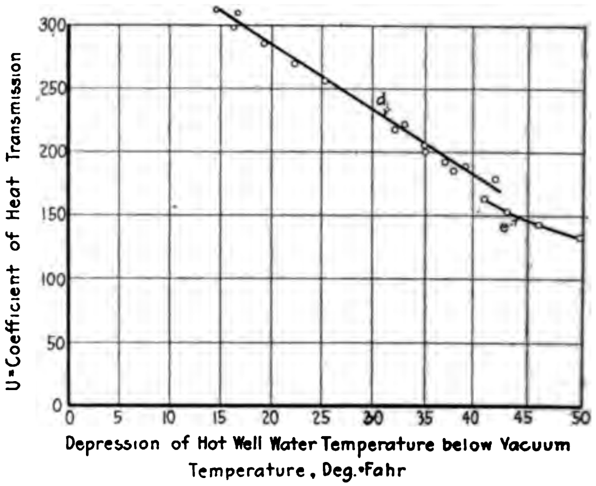


FIG. 3 RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION, CONDENSERS *d* AND *e*

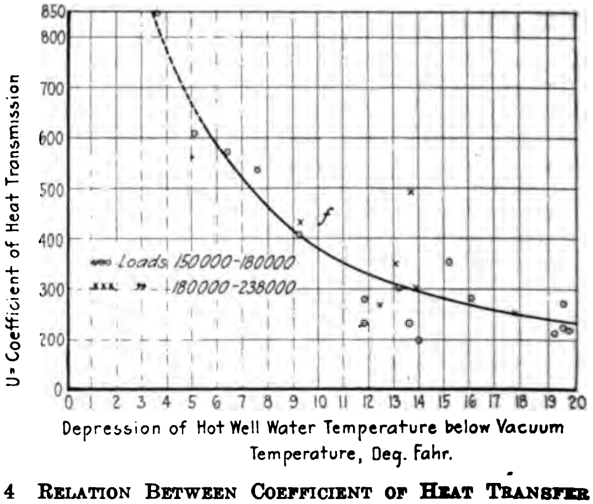


FIG. 4 RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION, CONDENSER *f*

8 The variables affecting heat transmission on the ***steam side of the tube in an actual condenser*** are complex and difficult of isola-

tion; yet as an observed fact, their influence really dominates performance, since the coefficients of heat transfer still vary several

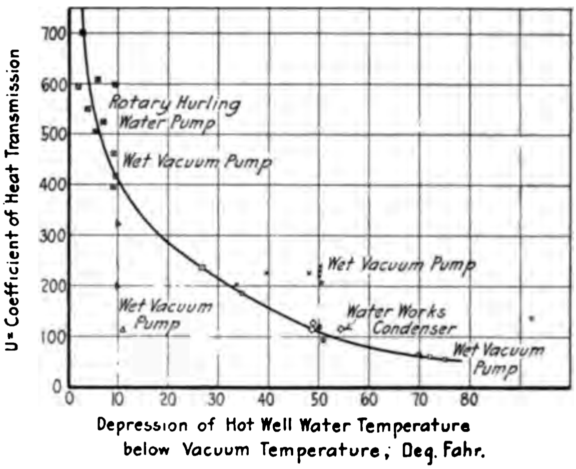


Fig. 5 Relation Between Coefficient of Heat Transfer and Temperature Depression

Graph *g* for a number of Condensers with Different Air Pumps

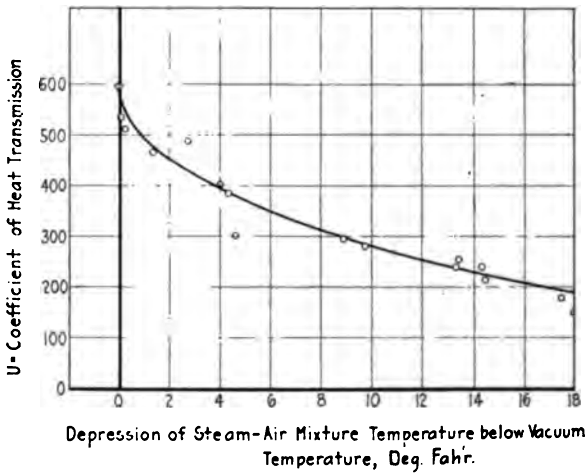


Fig. 6 Relation Between Coefficient of Heat Transfer and Temperature Depression

Graph *h* for Small Vacuum Evaporator

hundred per cent after all variables affecting heat transfer on the water side have been allowed for.

9 In Figs. 2 and 3 the coefficient of transmission for a number of large surface condensers served by reciprocating dry air pumps is plotted against difference between steam and condensate temperatures. Actual air pump suction temperatures were not available in all cases, and were also inaccurate due to difficulty in measurements. These tests on condensers were made by Mr. Orrok, and through his courtesy the authors have been supplied with the figures for plotting the graphs. Different amounts of air were admitted to each condenser under conditions of approximately constant load, water temperature and velocity and air pump capacity. The graphs refer to the following conditions respectively:

Graph	Surface in sq. ft.	Average load per sq. ft. in lb. per hr.	Vacuums	Remarks
a	24,000	5.9	28.5—26	
b	18,000	7.5	28.5—25.6	
c	18,000	4.07	28.6—27.5	Water velocity low compared to *b*)
d	18,000	8.09	8.4—26.7	
e	21,000	5.95	27.6—26.7	

10 Fig. 4 refers to a large number of readings on a 25,000 sq. ft. condenser[1] under winter and summer conditions at different vacuums and loads.

Graph	Surface in sq. ft	Load per sq. ft. in lb. per hr.	Inlet Water Temperature	Vacuum	Remarks
f	25,000	6 to 9.6 lb.	32 to 79	29.4—27.5	Coefficient corrected to velocity of 4 and mean water temperature of 75 deg. in proportion to variation indicated by Fig. 22

11 Fig. 5 refers to a number of condensers of different sizes and makes, the first two with wet vacuum pumps, the next three with hydraulic air pumps. The last refers to a water works condenser, in which steam is condensed within the tubes.

[1]Tests on exhaust turbine base condenser by H. G. Stott and R. J. S. Pigott. See Trans. Am. Soc. M. E., vol. 32, p. 69.

Graph	Surface in sq. ft.	Load per sq. ft. lb. per hr.	Inlet Water Temperature	Vacuum	Remarks
g	1,235	3.38— 8.05	58 —60	24.8 —25.3	Wet vacuum pump
	2,700	6. — 7.04	75	27.9 —27.3	Leblanc
	25,000	6.80—11.8		28.25—29.33	Leblanc
	25,000	5.9 — 9.7		28.4 —27.8	Leblanc
	2,130	4.65— 5.5		29.24—29.33	Leblanc
	2,120	7.00— 7.75	32 5—35.3	28.25—28.47	Water works condenser with dry vacuum pump

12 Fig. 6 shows a similar relation between coefficient and air-steam mixture depression (in this case the actual air-steam mixture was observed, not the condensate) for an evaporator. The graph is based on tests on calandria A, paper by E. W. Kerr on Transmission of Heat in Vacuum Evaporators.[1] It will be noted that the graph passes through zero air depression, indicating the complete absence of air. Mr. Kerr explains that this part of the tests is slightly inaccurate as "it is reasonable to suppose that there was a small amount (of air) present." The conditions were:

Graph	Surface	Load per sq. ft. in lb. per hr.	Temperature of Vapor Space[1]	Vacuum	Remarks
h	56	6.4 to 1.54	152	12.4	Calandria A

[1]Corresponds to circulating water temperature.

13 Figs. 2 to 6 are plotted in one figure in Fig. 7. Graphs *a* and *d* coincide, although *a* refers to a 24,000 sq. ft. condenser loaded to 5.9 lb. per sq. ft. and *d* to an 18,000 sq. ft. condenser loaded to 8.09 lb. per sq. ft. Similarly, graphs *b* and *e* refer to condensers of different sizes and loads. Graph *c* is low, explained by the lower water velocity through the tubes, also perhaps by fouling of the surface, the very low load and poor steam distribution.

14 Graph *f*, for the 25,000 sq. ft. base condenser, matches the others quite closely. In this graph the coefficient rises to a value of 850 B.t.u. for a single set of observations, whereas a value of 700 for the same velocity and mean temperature are obtained under the ideal conditions of Orrok's experimental single-tube condenser. (See Figs. 21 and 22.) Any error in the vacuum reading would have a large influence on the calculated average coefficient with the small mean temperature difference here existing.

[1]Trans. Am. Soc. M. E., vol. 35, p. 731.

15 Graph *g* agrees with the other curves quite closely, considering that it refers to five condensers of different sizes, loads and vacuums. Graph *h* for the vacuum evaporator also follows the general trend of the others.

16 Fig. 8 shows similar graphs based on results obtained by Prof. J. A. Smith[1] with experimental laboratory apparatus. The four lower curves show the relations existing between the coefficient of heat transmission and the number of degrees which the temperature of the air-steam mixture is below the steam temperature corresponding to the total pressure. Calling the heat transfer at zero

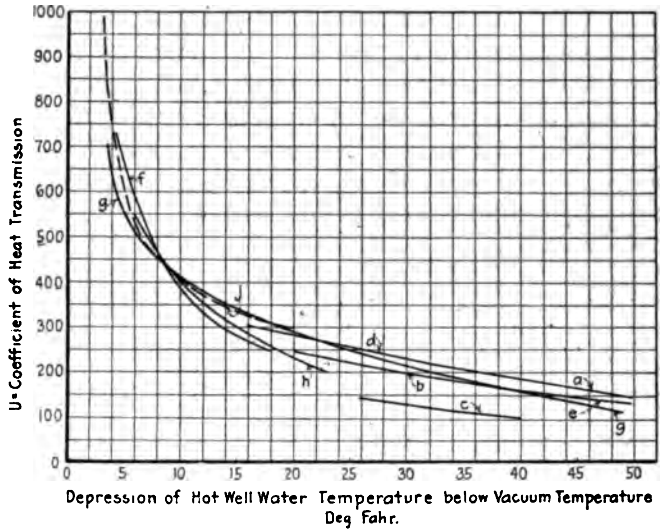


FIG. 7 RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION. (FIGS. 2 TO 6 PLOTTED TOGETHER)

air 100 per cent, all the results for different steam temperatures can be combined in a single percentage of efficiency curve, as shown by B, Fig. 8.

17 The maximum heat transmission obtained by Smith with pure steam is low as compared with that obtained by Orrok and others. This may have been due to low water velocities (2.2 ft. per sec.) and also to the fact that in Smith's single-tube "condenser boiler" there was no steam velocity, except that incidental to the movement of the steam towards the condensing surface. Under

[1]Victorian Institute of Engineers, Dec. 9, 1905.

these conditions even a minute quantity of air might seriously hinder heat transmission by forming a blanketing film around the tube. His curves also apparently show a marked effect due to steam density or temperature below 100 deg. tahr., although other experimenters have seemed to show that steam density has no effect on heat transfer.[1]

18 Graph B is replotted as the dotted graph j in Fig. 7 by assuming the maximum coefficient to be 1000 at a condensate temperature 3 deg. below steam temperature, in order to bring it into the region

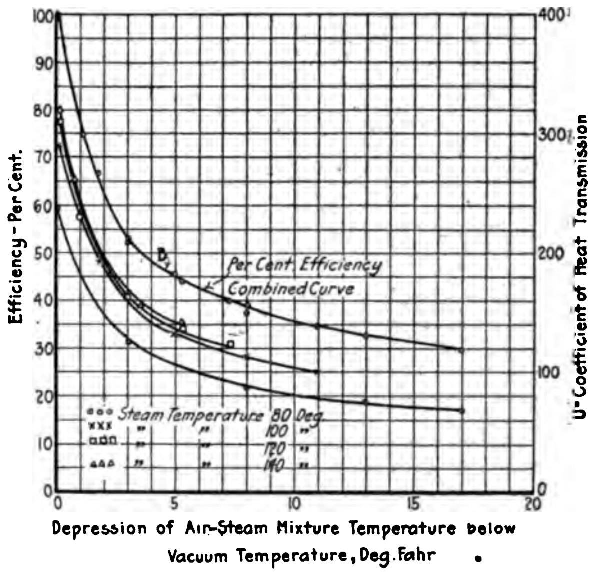


FIG. 8 RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION. ALSO EFFICIENCY VS. TEMPERATURE DEPRESSION

of the other graphs representing commercial condensers. Graph j is then seen to have the same trend as the others.

19 The graph of Fig. 9 was obtained by averaging the results represented in graphs a, b, c, d, e, f, g, and h, and expressing the heat transmission as a percentage of the maximum obtainable under the

[1]See Conclusion and Table 3, Tests at Vacuums from 27 to 7 in., Transmission of Heat in Surface Condensation, Geo. A. Orrok, Trans. Am. Soc. M. E., vol. 32, p. 1139. Also The Efficiency and Design of Surface Condensers, Stanton, Proc. Inst. C. E., vol. CXXXVI.

conditions of rate of heat transfer through the tubes and from tubes to water, prevailing in the condensers considered. It was assumed that when the coefficient rises to 1000 in Fig. 7, this maximum condition exists, all the surface being active, the condenser being filled with pure steam and the depression of temperature being zero. As it is impossible to obtain complete and equal activity of all the surface in the commercial condenser, wherein there is always some pressure loss and air in leakage, the 100 per cent condition can only be approximated. The graph of Fig. 9 is not offered as a final result and the phenomenon requires further study. For instance, no reason is at present known why a degree fall in temperature due to

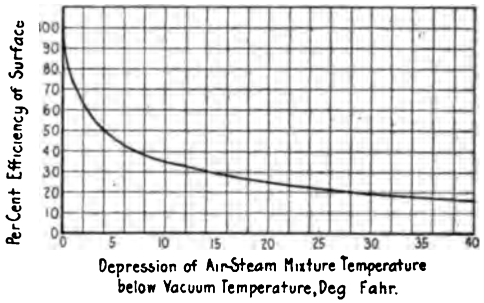


FIG. 9 EFFICIENCY OF SURFACE VS. TEMPERATURE DEPRESSION

resistance to flow should have the same effect in reducing the average coefficient of transmission as a degree fall due to air concentration, and possibly it does not.

20 The better the condenser the more the proportion of active surface and the nearer it approaches the 100 per cent mark. The performance of a condenser is usually dominated, to the exclusion of other factors, by the extent of the active and inactive zones of condensation. Thus tests show that the coefficient averaged for all the surface may *increase* as the water velocity *decreases*.[1] The explanation is to be found in the lower vacuum and smaller specific volume of steam corresponding to the lower velocity smaller quantity and higher final temperature of circulating water, under which conditions there is greater steam penetration and more surface is brought into action, as will be explained later.

21 Other explanations of the variations in heat transmission coefficient have been offered by an editorial writer in *Engineering*,[2]

[1]Tests by E. Josse, Zeitschrift des Vereins deutscher Ingenieure, Feb. 1909.
[2]Engineering (London), January 2, 1914, et seq.

Orrok,[1] Loeb,[2] and others, founded principally on the variables affecting the water side of the tube or on the theory that a constant coefficient would be obtained by assuming the total heat transfer to vary as some fractional power of the instantaneous temperature difference instead of the first power. These are examined in Appendix 1.

22 We may assume for the present, that the coefficient of heat transfer for a single tube immersed in steam, depends on the water

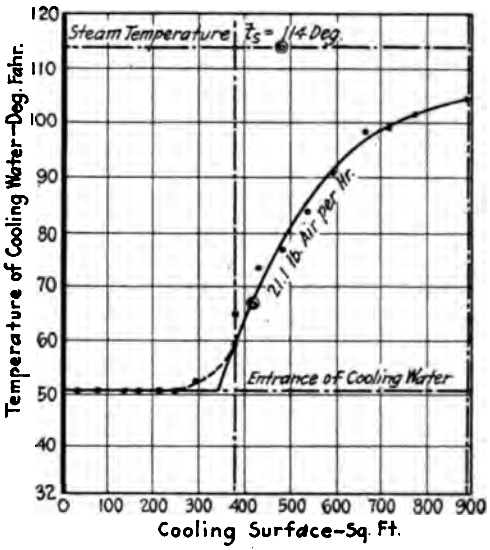


FIG. 10 RISE OF WATER TEMPERATURE IN PASSING THROUGH CONDENSER INDICATES ACTIVE AND INACTIVE ZONES

velocity (in an open tube, or on the manner of agitation in other tubes), size of tube, material and cleanliness of tube and mean water temperature, but that all of these factors taken together do not explain the wide disparity of results obtained with actual condensers. We may also assume in the light of evidence given in Appendix 2 that the total heat transfer is a function of the first power of the temperature difference, and not of a fractional power as suggested by Loeb and Orrok.

COMBINED EFFECTS OF PRESSURE DROP AND AIR

23 Measurements of rise in temperature of circulating water

[1]Trans. Am. Soc. M. E., vol. 32, p. 1139.

[2]Heat Transmission and Tube Length in Marine Feed-water Heaters, Jour. Am. Soc. Nav. Eng., May, 1915.

in multiple pass condensers have shown that there are two fairly well defined zones in a surface condenser: one in which condensation takes place actively, the other wherein very little condensation takes place. In other words, the bulk of the steam, in the condition in which it originally enters the condenser, does not penetrate entirely through the bank of tubes. Fig. 10 from a paper by Prof. Josse[1] shows readings of water temperature and of heat absorbed in an

TABLE 1 B.T U. AT VARIOUS PARTIAL PRESSURES

(1) Temperature in Deg. Fahr.	(2) Partial Pressure of Steam in in. of Hg.	(3) Partial Pressure of Air in in. of Hg.	(4) Volume of 0.0002 lb. air at Temp. (1) and Press. (3)	(5) Cu. ft. of Steam Condensed	(6) B.t u. Given up Latent Heat per lb. =1047
79	1.0	$\frac{1}{8000}$	657	0	0
79	1.0	$\frac{1}{4000}$	328	329	524
79	1.0	$\frac{1}{2000}$	164	493	784
9	1.0	$\frac{1}{1000}$	82	575	915
79	1.0	$\frac{1}{500}$	41	616	980
79	0.99	$\frac{1}{100}$	8.2	648.8	1030
78.5	0.98	$\frac{1}{50}$	4.1	652.9	1039
78.5	0.978	$\frac{1}{40}$	3.28	653.7	1041
78.2	0.967	$\frac{1}{30}$	2.46	654.5	1042
77.5	0.95	$\frac{1}{20}$	1.64	655.4	1043
76	0.9	$\frac{1}{10}$	0.823	656.2	1045
72.[illegible]	0.8	$\frac{1}{5}$	0.412	656.59	1045
59	0.5	$\frac{1}{2}$	0.164	656.84	1045

actual condenser plotted against surface traversed. About 40 p[illegible] cent of the surface is inactive.

24 To explain what goes on, let us assume a pound of steam-a[illegible] mixture containing 0.9998 lb. of steam and 0.0002 lb. of air, an ave[illegible] age entering mixture for a tight condenser installation. At 1 [illegible] of mercury absolute pressure and a steam temperature of 79 de[illegible] fahr., the volume of 0.9998 lb. steam is 657 cu. ft. and the volu[illegible] of the 0.0002 lb. of air is also 657 cu. ft., from which its part[illegible] pressure is calculated as 1/8000 in. of mercury. This pressure i[illegible]

[1]Schiffsbautechnische Gesellschaft, Berlin, June 16-18, 1908.

small as to appear almost negligible, and it would seem that such steam could be called air-free. In fact it is practically air-free steam and acts as such, giving a high heat transmission coefficient so long as the partial pressure of the air remains so small.

25 When half the steam has been condensed, the volume will be half and the partial air pressure 1/4000 in., causing a corresponding decrease in the partial steam pressure and temperature, as shown by Table 1. The steam can still be called pure steam. In fact, about 99 per cent of the steam must be condensed before there is any appreciable increase in partial pressure of the air, or appreciable

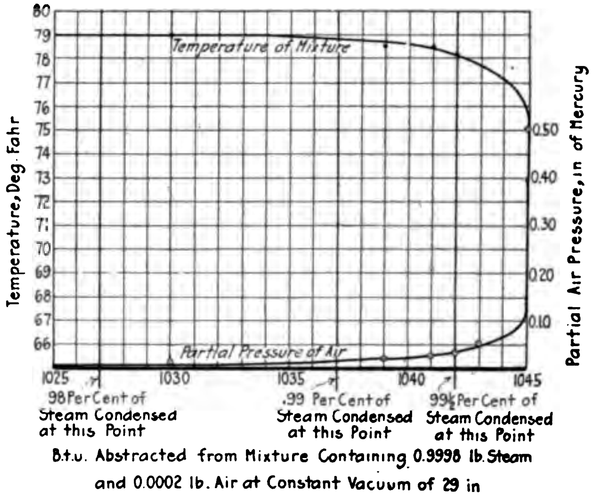


FIG. 11 RELATION OF HEAT ABSTRACTED FROM MIXTURE OF STEAM AND AIR, TO TEMPERATURE AND PARTIAL AIR PRESSURE

decrease in temperature of the air-steam mixture as shown by Fig. 11. When 99 per cent has been condensed, the partial air pressure is then 1/80 in., but the air is still very rarified and its volume is still too large for removal by a vacuum pump.

26 Thus a process of air compression must take place after nearly all the steam has been condensed, which explains the phenomenon shown by Fig. 10 and similar tests in which there is no appreciable rise in circulating water temperature in the bottom of the condenser where the air concentration and compression is taking place. No appreciable amount of steam remains to be condensed, but, still, surface is needed to condense the small amount present, the heat exchange being greatly impeded by the presence

of the air. The richer the mixture delivered to the air pump is in air, the greater the work to be performed by, and hence (other things being equal) the larger, this "inactive" zone of the condenser.

27 Let us consider a condenser, Fig. 12A, in which steam condensation occurs up to line $a\,b$ and air concentration during the flow over the remainder of the tube surface; i.e., nearly all the heat is transmitted to the water in that part of the surface preceding the line $a\,b$. The water temperature is assumed constant at T_w. In the first zone the temperature drop to T_1 is almost entirely due to the pressure drop required to overcome the pneumatic resistance offered by the condenser structure to the flow of a large volume of steam. From this region on, the temperature drop corresponds to the reduction in partial pressure of the water vapor, the drop in

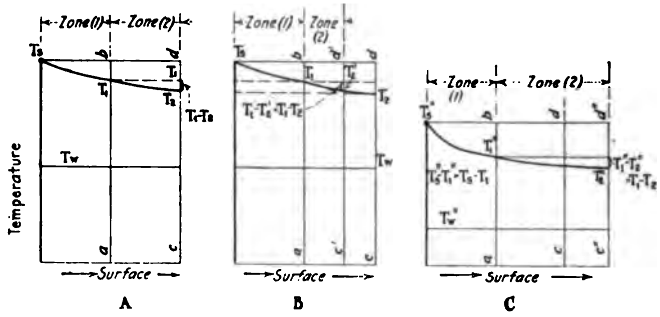


FIG. 12 VARIATION OF PROPORTION OF ACTIVE AND INACTIVE ZONES, UNDER DIFFERENT CONDITIONS OF AIR LEAKAGE AND VACUUM

total pressure being negligible, due to the fact that the volume flowing in this zone of the condenser, where only 1 or 2 per cent of the steam remains, is very small. The partial pressure of the air at exit is represented by the difference of the vapor pressures corresponding to the temperatures T_1 and T_2, T_2 being the final air pump suction temperature.

28 Now assume that the air leakage is reduced, or the air pump capacity increased, so that the air withdrawal occurs at some lower partial air pressure represented by the smaller difference T_1—T_2' (Fig. 12B). A part of the zone (2) may then be dispensed with and, as the same initial steam temperature T_s is maintained with less surface, the average coefficient of heat transfer will figure out higher.

29 Again, assume the case in Fig. 12C, wherein the water tem-

perature is reduced and a higher vacuum maintained. If all of zone (1) up to *a b* is to remain active, the pressure drop expressed in head of steam must increase approximately in proportion to the square of the steam velocity and specific volume. Temperature T_1'', is therefore closer to the water temperature T_w'' than before. Furthermore, the difference between this temperature T_1'' and the air pump suction temperature T_2'' must increase in order to give the same partial air pressure with a lower total pressure. This is explained by Fig. 13. For a constant partial air pressure, which means constant air leakage with constant pump displacement, the temperature difference and the percentage of air rich-

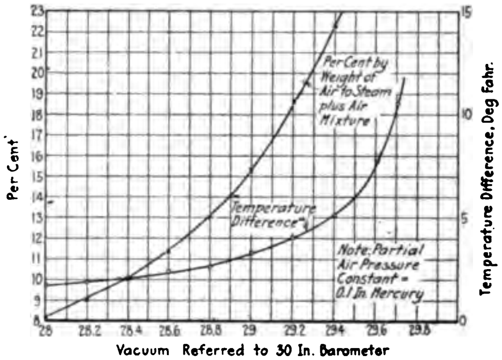


FIG. 13 PERCENTAGE AIR IN MIXTURE AND DIFFERENCE OF TEMPERATURES CORRESPONDING TO TOTAL PRESSURE AND THAT ACTUALLY EXISTING, FOR CONSTANT AIR PRESSURE OF 1/10 IN.

ness must both increase rapidly as the vacuum increases. For these two reasons more surface will be required in zone (2) of Fig. 12c extending it to c″—d″ and the average coefficient of heat transfer will accordingly figure out lower. [The coefficient will also be further reduced because the average temperature of the steam is now lower through zone (1) on account of the greater pressure drop.]

30 Many variables have to be taken into account to calculate the actual relative depths of these zones in a condenser. Among the independent variables which would need to be considered may be mentioned water velocity, initial water temperature, weight of water, tube diameter and cleanliness, arrangement of water passes, steam temperature, amount of air, air pump capacity, amount of surface and, perhaps most important of all, the arrangement or

geometrical configuration of the heat transmitting surface. However, we find that if the difference between vacuum temperature and air outlet temperature $T_s—T_2$ is increased by increasing pressure drop or by increasing the amount of air, there is a decrease in the average coefficient of transmission; and conversely if $T_s—T_2$ is decreased by decreasing pressure drop or decreasing the amount of air, there is an increase in the average coefficient. Empirical results showing the same relation have been given in Figs. 2 to 9.

PRESSURE DROP

31 If the proportion of steam condensed per tube, the depth of penetration and the steam density were all constant, the loss of head through the first zone of the condenser would vary as the square of the velocity, or according to the well known relation for inelastic fluids $h = v^2/2\,g$. If the amount of steam condensed per tube and the steam density remained constant, but the penetration increased in proportion to the velocity of entrance, the pressure drop would vary as the cube of the entrance velocity.

32 Pressure drops actually observed usually fall between these limits. By dividing both the steam weight flowing and the pressure drop resulting by the mean specific weight of the steam, test readings can be expressed in terms of steam velocity and "head" of steam.

33 In Fig. 14 are plotted on logarithmic paper the heads and proportional velocities obtained in experiments at Annapolis by Loeb, on a Bureau of Steam Engineering feed water heater.[1] The points agree very closely with a line of slope 2.5; that is, the head varies as $V^{2.5}$.

34 The exponent of 2.5 can be explained on the basis that at increased loads the steam penetrates farther into the heater; it is like pumping water into a leaky distribution system. This is borne out by the quantities plotted in Fig. 15 for three condensers of large size. In these tests the increase in steam velocity (weight per hour divided by specific weight) is due entirely to increase in vacuum and specific volume, the weight flowing actually decreasing at the higher velocities. These are the same tests from which graphs *a*, *b*, and *e* were plotted in Figs. 2 and 3. The turbine load was held substantially constant, while the vacuum was caused to decrease by admitting air to the condensers. The drop in vacuum increased the steam rate of the turbine and the total steam consumption, but the

[1]Jour. Am. Soc. Nav. Eng., May, 1915.

decrease in vacuum offset this increase sufficiently to cause a net decrease in steam velocity and in pressure drop. At the same time, the artificial admission of air to the condenser increased the extent of the inactive air-blanketed zone in the condenser, as is shown by the decrease in average heat transmission coefficient. Accordingly it is to be inferred that as the steam velocity decreased, the depth of steam penetration decreased; and as the steam velocity increased the depth of penetration increased. The slopes of the lines show exponents of 2.85 in the case ot the condensers corresponding to graphs *a* and *b* and of 3.00 in the case of the condenser corresponding to *e* of Figs. 2 and 3.

35 To show the probable steam penetration, we can compare the following sets ot figures tor condenser *a*, the last column being based upon the assumption that actual transmission coefficient for each element of surface is the same, so long as it is in contact with pure steam:

Load in Lb.	Average Steam Pressure in In. of Hg.	Entrance "Velocity" = W/Density $\times 1000$	Head = Pressure Drop ÷ Density	Coefficient of Heat Transmission	Approximate Per Cent Increase Steam Penetration[1]
149,950	3.74	27,700	9.3	144	——
124,760	1.62	53,000	85.0	272	50

36 The proportionate steam velocity in the second case is double that in the first, so that the head lost should be four times as great, or about 37 instead of 85. But the average transmission coefficient increases at the same time from 144 to 272, with an increase in steam penetration of about 50 per cent. The velocity at the exit of the active zone in each case may be considered zero. Assuming equal condensation on each row throughout the active zone, the mean velocity for *A* of Fig. 16 is proportional to $(27700+0) \div 2$ or 13,850. Now the active zone in *B* is increased 50 per cent and the steam is only 67 per cent condensed and 33 per cent remains uncondensed when it has traversed the tubes that were active in *A*. Hence the mean velocity in this zone is as $(53{,}000+33$ per cent of $53{,}000) \div 2 = 35{,}000$ which is considerably more than twice the mean velocity in *A*. Besides there still remains 33 per cent of the steam which

[1]The coefficient of heat transfer is based on water temperatures measured at the inlet and outlet. Actually, the entire quantity of water is not active, part of it flowing through tubes which are not reached by the steam, even in the upper pass. The increase in penetration is therefore somewhat less than the percentic increase in average coefficient.

penetrates further into the condenser, with further pressure loss. These taken together account for the fact that the loss of head increases faster than the second power of the steam velocity at the entrance.

37 The simultaneous conditions of increased velocity and increased depth of penetration were artificially produced in the case of the foregoing experiments. They are found to occur together, however, in ordinary practice when the load is increased. The increased weight of steam is accompanied by a fall of vacuum, but the decrease in specific volume is insufficient to offset the increased weight, and the velocity increases. At the same time the depth of penetration also increases, due to decrease in the inactive zone.

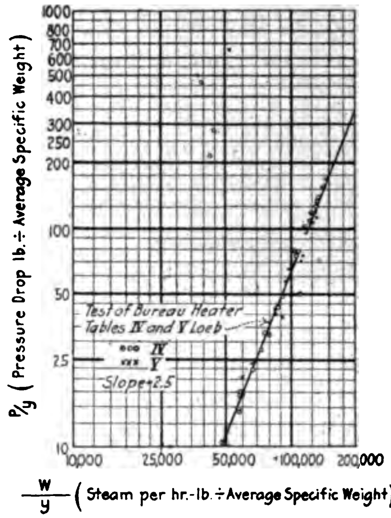


FIG. 14 RELATION BETWEEN PRESSURE DROP AND STEAM VELOCITY

in which the proportion of air is less at lower vacuum and also due to the decrease in total air leakage because of the higher pressures and hence less leakage in the turbine casing. Furthermore, at lower vacuum the air pump can remove the air more efficiently.

38 In some cases the increase in steam velocity is accompanied by decrease in active zone, and then it would be expected that the head of steam would vary as a function of the velocity less than the square. This occurs, for example, when the load is decreased but the vacuum greatly increased, as by the use of very cold water. The decrease in load is accompanied by greater air leakage and this

taken together with the fact that the air-steam mixture must be richer at high vacuum, as has been shown by Fig. 13, causes an increase in the inactive zone and shortening of the active zone. At the same time the velocity of the steam increases due to the higher vacuum.

39 If the depth of the active condensing zone is assumed to remain constant, however, we can assume that the head lost varies at least as the square of the velocity. In estimating performance, this is a provisional basis of calculation; for, as will now be evident, to calculate the pressure drop it is necessary to know the depth of penetration, which in turn depends on the average heat transfer which is the final result sought by calculation.

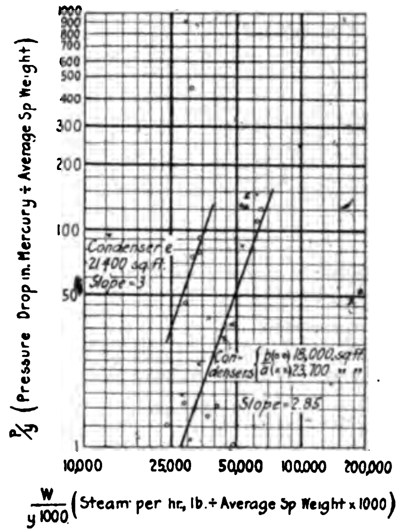


FIG. 15 RELATION BETWEEN PRESSURE DROP AND STEAM VELOCITY

CALCULATION OF SURFACE EFFICIENCY

40 Assume that the several characteristics of a surface condenser are known for 28.5 in. vacuum, under which conditions the pressure drop through the condenser is 1/10 in. of mercury column and the air pump capacity and air inleakage are such that the partial pressure of the air is 0.124 in. of mercury, so that the depression of the air pump suction temperature is 5 deg. below the steam temperature, corresponding to an efficiency, from Fig. 9, of 45 per cent.

41 We wish now to know the conditions at 29-in. vacuum. We have seen from Fig. 12 that additional cooling surface is required in zone (2) in order to reduce terminal pressure, so that the weight of steam at the greater volume corresponding to the higher vacuum can penetrate to the same depth in zone (1), wherein, in other words, greater pressure drop will occur. In the actual condenser, the surface remains constant, so that there must be a re-adjustment of working conditions, zone (1) becoming smaller because of resistance to the flow of the greater steam volume, also because more surface is required in zone (2) for air concentration.

42 [We must bear in mind that zone (1) is not only affected by pneumatic conditions but by the performance of zone (2) and of the

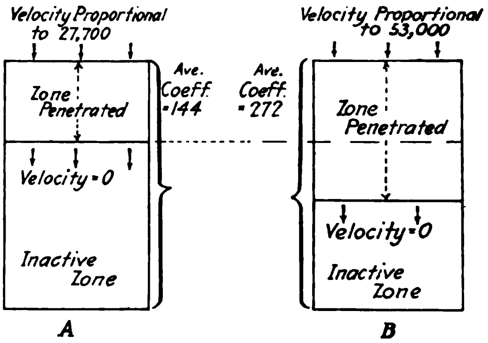


Fig. 16 Example of Greater Penetration Accompanied by High Steam Velocity

air pump. Thus if we slow down the air pump, zone (1) will be shortened with consequent decrease in vacuum.]

43 We can calculate the pressure drop for the assumed elastic condenser of Fig. 12, wherein zone (1) always remains the same. This, in connection with the depression due to air pressure, leads to a temperature difference T_s—T_2 and a corresponding surface efficiency, which we will apply to the actual condenser.

44 Thus the pressure drop at 28.5 in. of 0.1 in. becomes 0.148 in. for the larger volume at 29 in. vacuum and the total pressure at the air suction is $(1-0.148) = 0.852$.

45 Modern air pumps, particularly the hydraulic types, running at constant speed, remove practically a constant volume of air, that is, have constant displacement or equivalent displacement under wide conditions of operation. Therefore with constant air in-

leakage the partial pressure of the air in the air-steam mixture withdrawn from the condenser must also be a constant, neglecting the slight effect of temperature on air volume.

46 The partial air pressure thus remains equal to 0.124 and the partial vapor pressure is 0.852—0.124 = 0.728 in. This pressure corresponds to a steam temperature of 69.5 deg., which is a depression of 9.5 deg. below the vacuum temperature, so that the efficiency of the condensing surface is now 36½ per cent, according to Fig. 9. The results of using this percentage as the amount of surface in active condensation will be approximate only, because a change in the number of tubes penetrated also entails a change in the amount of water actually heated.

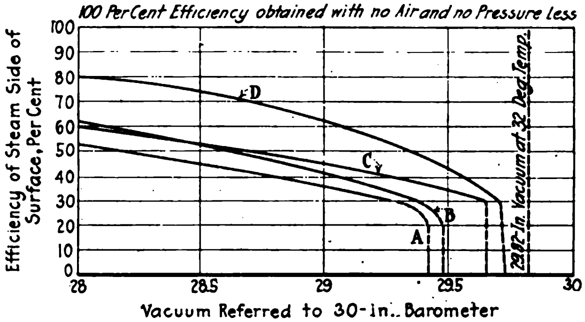


Fig. 17 Effect of Air and Pressure Loss on Efficiency

47 Similarly the depression for 29.2 in. vacuum is found to be 13.8 deg. and the efficiency 32 per cent and so on for other vacuums, and a graph can be plotted as shown in Fig. 17, graph A.

48 Graphs B, C and D were obtained similarly, B being for a lower partial air pressure, due to less air leakage or a larger capacity pump, C for the same air conditions as A, but less pressure drop (as with better spacing of tubes), and D for both reduced air and less pressure drop.

49 Graph B: *Efficiency When Air Leakage is Reduced or Larger Air Pump Used.* Reducing the air leakage (or increasing pump capacity) gives less temperature depression and hence raises the efficiency particularly at moderate vacuums.

50 Graph C: *Efficiency when Pneumatic Resistance is Reduced.* Reducing the pressure drop, raises the high vacuum end of the efficiency curve and also extends it to higher vacuums. The effi-

DISCUSSION

Leo Loeb. To one who has been keenly interested in the subject of heat transfer, this paper raises certain points which cannot be passed without question.

The authors state it may be assumed, in the light of evidence given in Appendix 2, that the total heat transfer is a function of the first power of the temperature difference, and not of a fractional power as suggested by Loeb and Orrok, but the appendix contains no evidence to support this claim. However, in discussing certain data, they state it was concluded from these tests "that the coefficient was constant if the total heat transfer was assumed to vary as some fractional power of the temperature difference rather than unity." This is a misstatement, as there was no assumption of total heat transfer as an exponent of the temperature difference. It remains a fact, determined from many experiments, that the total heat transfer is proportional to a power of the temperature difference. This power is not necessarily less than unity, although in most of the tests I have examined of the transfer of heat from steam to water under practically air-free conditions the coefficient was less than unity. However, in the case of heat transmission from condensing steam and air, the coefficient is usually greater than unity. The quotation should therefore be recast somewhat as follows, "and from these it was shown the total heat transfer was proportional to some power of the temperature difference, usually less than unity, hence the coefficient for one rate of flow would be constant."

The manner in which this law was investigated in the experiments referred to was to introduce water at a reasonably low temperature into a short-pass heater, keeping the steam temperature as nearly constant as possible. The outlet temperature was carefully noted after a test of reasonable duration, and then the water was re-introduced at the inlet of the heater at the temperature at which it appeared at the outlet during the previous test, and so on, carrying this up along a true temperature gradient. When the gradient was plotted it was in exponential form, and not in logarithmic form as would be required by the law of heat transfer proportional to the first power of the temperature difference.

The authors call attention to the fact that in the tests on the Bureau heater, the steam temperature varied only 2 deg. In the experiments on the apparatus, we tried to keep the temperature constant, but the construction of this particular heater was such that with high rates of water flow there was a small condenser action and a slight drop in pressure within the shell making it almost impossible for the temperature to be kept absolutely constant.

The authors further state that it is reasonable to expect that if the variation in temperature difference were obtained entirely by varying the steam temperature only, the exponent would be found to attain unity. Now in marine feedwater heaters, and in closed heaters generally, the object is to attain a standard of temperature; and while we did investigate the effect of varying pressure, and while the authors have taken three points from these tests and found them alike on a straight line with a slope of unity, that is simply a coincidence. The indisputable fact remains that a long series of experiments on single tube and commercial marine feedwater heaters shows the temperature gradient to be exponential, and no reasonable expectancy on the part of those who have not "lived" with the apparatus will change the fact.

The curves in the writer's paper to which the authors refer were plotted from heaters in which we were forced to use a different method of obtaining the temperature gradient, and while it does introduce a little more complexity, still these are the facts, and we must abide by them.

R. N. Ehrhart. The first part of the author's paper consists in the collection and tabulation of results obtained with various surface condensing plants. As such, it is interesting. However, we do not believe that curves derived from the performance of various sizes and types of condensers of different manufacturers can be consistently grouped in curves to show typical results. It is quite evident that one style or design of condenser might give certain characteristics which could be successfully shown graphically, but to take all other types and put them on the same curve sheet with the expectation of getting something that is really of great value is impossible.

The latter part of the paper is based on misconceptions of the performance of modern condensers. For example: The performance of the condenser as shown in Fig. 10 is not at all in line with that given by modern design. In days gone by, we used condensers in which there was little rise in temperature in the lower pass, but condensers of today, if properly designed, will have about 60 per cent of the work done in the lower pass. It is obvious that this should be so. If the condenser is perfectly scavenged of air, the greatest temperature head exists in the lower portion of the condenser, which, in the contra-flow type, is nearest the end of the condensation zone, and in the beginning of the water circulation zone. The use of a hydraulic air pump makes it easy to get such results without the use of external coolers.

The authors attempt to show that there is a compression of the air in the condenser, and that this necessitates a lowering of the temperature of the steam space. Such a phenomenon need not exist at all. Where a hydraulic air pump is employed, using water of the temperature of that delivered to the condenser, the compression can all take place within the pump itself; that is, the readjustment of vapor and air pressures does not need to take place in the condenser.

The authors dwell on a suggested form of zone condenser, in which the lower pass uses little water and consequently little power is there absorbed. As pointed out above, the modern condenser does most of the work in the bottom pass, so that the suggested form of zone condenser, instead of accomplishing something desirable, turns out to be about the most undesirable modification that we could make in a modern condenser. Therefore, it can have no place in modern condenser practice.

George H. Gibson (written). In none of Mr. Loeb's tests did he work with a constant fixed water temperature and different steam temperatures; and that would be necessary in order to eliminate the variations that might be due to the water temperature, per se, or else the influence of the water temperature must be allowed for. In the tests described by Loeb, the influence of water temperature was ignored. It is therefore improper, on the basis of his tests alone, in which the variations in temperature difference were obtained by varying the water temperature, to state just what precise effect temperature difference by itself had upon the rate of transmission.

In reply to Mr. Ehrhart, the condenser tests presented are those to which we had access in publication and in reports by manufacturers and users. If there are in existence tests showing no temperature drop caused by enrichment of the air mixture, we would like to examine them. We concede that by putting on a supplementary condenser the mixture can be drawn out of the first condenser before the effects of air are apparent, that is to say, before ubes of the condenser. Fig. 19 shows how the same conditions are obtained using two circulating pumps.

59 In Fig. 20 is shown a three-pass condenser in which the velocity in the lower zone is small, in the middle zone fairly high, and in the upper zone the maximum. All of the water goes through the upper zone, but only part of the total through the other zones. By manipulation of the valve, the quantity of water going through the zones is varied to take care of different conditions of operation in summer and winter, and of load, air leakage, air pump capacity and vacuum. A three-pass zone condenser may also be built with, say, half the tubes in the bottom or first pass and the other half of the tubes divided between the two upper passes. The objection to this design is that the inactive zone is too large (one-half the total) and the water velocity still too high.

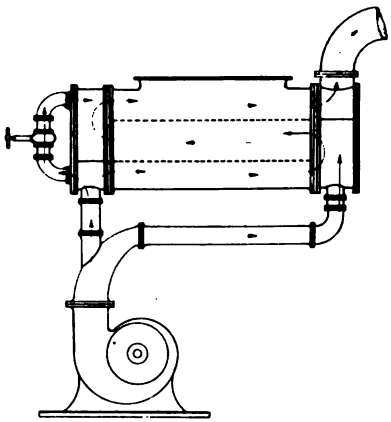


FIG. 20 THREE-PASS CONDENSER WITH THREE ZONES OF DIFFERENT ACTIVITY

CONCLUSIONS

60 The lower average heat transfer coefficient obtained with cold circulating water is due principally to a greater proportion of the surface being inactive, which in turn is due to greater pressure drop incidental to the flow of a larger volume of steam and to the greater proportion of air to vapor existing in the mixture stagnating in the outlet zone of the condenser.

61 Better efficiency of surface at high vacuum can be obtained by preventing air inleakage, increasing pump capacity and decreasing the pressure drop or pneumatic resistance of the condenser by

proper arrangement of steam path and spacing of tubes, these being of great importance at very high vacuums.

62 The pressure drop through a condenser, expressed in terms of head of steam, varies as a function of the velocity of the steam, greater than the square if the increase in velocity is accompanied by an increase in average heat transmission coefficient and in depth of penetration, or smaller than the square if the increase in velocity is accompanied by a decrease in average heat transmission coefficient and in depth of penetration.

63 In purchasing high vacuum condensers, comparison should be made of the pneumatic resistance of the structures on the basis of velocity of flow at each row of tubes and the number of rows in series through which steam must flow. Attention should also be given to possibilities for the formation of air pockets out of the line of flow, considering both transverse and longitudinal sections.

64 The highest coefficients of heat transmission are obtainable in condensers of moderate size in which the smaller depth of tube bank lessens the pneumatic resistance.

65 The depression of the air suction temperature below the inlet vacuum temperature is an index of the surface efficiency, on the steam side.

66 By analyzing the temperature depression in a given condenser into that due to pressure drop and that due to partial air pressure, it is possible to determine whether flow conditions or air conditions offer the greater possibility for improving efficiency and vacuum.

67 By means of accurate electrical resistance thermometers temperatures can be taken at a multitude of points on both steam and water sides of the surface of a condenser undergoing changes in operating conditions, which would enable one to isolate the factors influencing the extent of the active and inactive zones.

68 A high velocity of circulating water, or the equivalent increase in water film agitation by the use of cores or spirals, is desirable in the tubes of the active zone of a condenser; and this can be obtained, without additional power consumption in pumping the circulating water, by reducing the velocity of the water through the tubes of the inactive zone.

No. 1516

CIRCULATION IN HORIZONTAL WATER TUBE BOILERS

By Paul A. Bancel, New York, N. Y.
Junior Member of the Society

With high combustion rates and high furnace temperatures, the load on boiler surfaces exposed directly to the fire is very intense. For a furnace temperature of 2500 deg. fahr. and a boiler temperature of 400 deg. fahr., the rate of radiation per square foot of boiler surface is about 100,000 B.t.u. per hr.; for 2750 deg. fahr., 150,000 B.t.u., and for 3000 deg. fahr., 200,000 B.t.u., corresponding approximately to loads of 100, 150 and 200 lb. of steam per sq. ft. respectively, whereas the load averaged for all the surface in a boiler is usually only 3.5 to 10 lb. per sq. ft.[1]

2 It is important that the tubes subjected to these loads have ample circulation of water, and be kept clean of scale as well, otherwise the resistance to heat transfer on the water side of the tube will increase, causing an increase in the temperature of the metal, which if high enough, will result in rupture. An increase in tube temperature will also decrease the heat absorbed by radiation and reduce the efficiency. It is only in the region of these tubes that the velocity of circulation can have any appreciable effect on the efficiency of heat absorption.

3 In addition, it is generally believed that a high velocity of circulation will retard scale formation and thus indirectly reduce the resistance to heat transfer and prevent overheating of the tubes.

4 Before considering the complex circuit of the actual boiler, let us consider the simple boiler of Fig. 1, with one leg containing solid water and the other a rising column of steam bubbles and water. The conditions may be likened to those in an air lift wherein the aeration of a column of water forming one leg of a U creates the flow.

[1] F. Munzinger, Zeit. d. Ver. deut. Ing., Nov. 1, 1913.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

Jordan reported results of experiments in which the temperature of the metal wall was measured at four points in its length by accurately calibrated thermocouples, and in this way the heat transmission from metal into the water was studied separately.

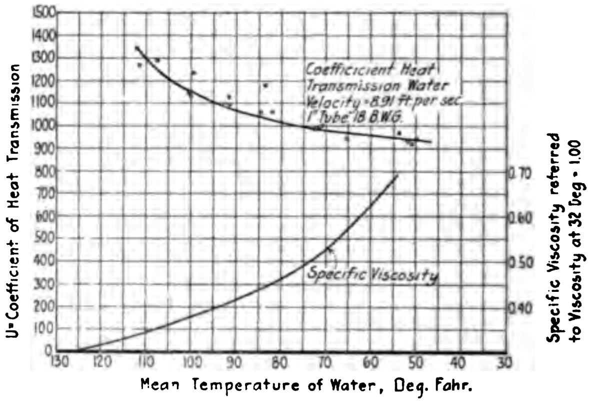


FIG. 21 HEAT TRANSMISSION VS. MEAN WATER TEMPERATURE

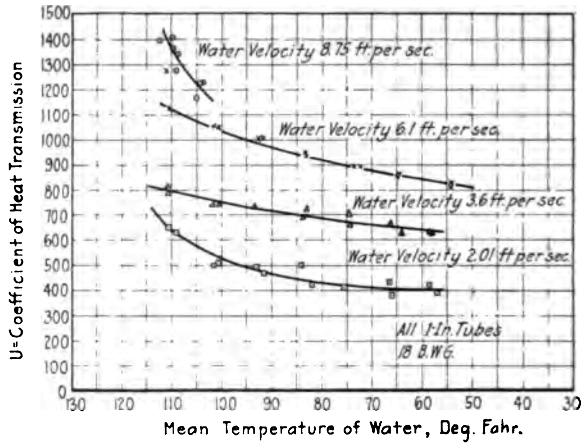


FIG. 22 HEAT TRANSMISSION VS. MEAN WATER TEMPERATURE

73 In Fig. 23, taken from Jordan's paper, are reproduced sults of experiments, each graph representing a constant m water film temperature. The arithmetical average of the m temperature and mean water temperature was taken to repre the mean water film temperature. As the metal wall was at

slightly higher temperature than the water, the figures given are very close to the mean water temperatures. It will be apparent that a considerable increase in water temperature results in only a small increase in coefficient of heat transmission. Thus for a flow of water 62.5 lb. of water per sq. ft. per sec., corresponding to one foot per second velocity, the mean water temperatures and corresponding coefficients of heat transmission can be set down as follows:

Mean Water Temperature	U = coefficient of heat transmission from metal to water in annular passage of mean hydraulic depth, m = 0.097; Jordan	U = coefficient of heat transmission from steam to water. See Fig. 22 for 3.6 ft. velocity in 1-in. tube of m = 0.23.
55	630	620
85	685	690
110	740	810

74 In the third column, values of U for 3.6 ft. velocity in Orrok's experiments from Fig. 22 are listed for comparison. The high coefficients obtained by Jordan at the comparatively low water velocity are explained by the facts (1) that the coefficient here considered is from the metal to the water, and the resistance due to the metal itself, as also that due to the fluid film on the steam side of the metal, are not included, and (2) that the water was flowing through an annular passage formed between a core and the tube, whose mean hydraulic depth was approximately 0.097, whereas for a condenser tube, the mean hydraulic depth is $d/4$, where d is the internal diameter (equals 0.902 for a one inch 18 B.W.G. tube, and m is equal to 0.23,.

75 Jordan's experiments also showed that the coefficient increased as a linear function of the water velocity, whereas the experiments by Orrok with heat transmitted from steam on one side to water on the other showed an increase about as the square root of the velocity. The explanation of this seeming disagreement can probably be found in the fact that at higher water flows when more heat is being transmitted per one degree difference between metal and mean water temperature, there is a correspondingly greater amount of heat being transmitted through the metal, which calls for a greater temperature drop therein, and also a greater amount of heat being transmitted through the fluid film on the steam side of the surface, where there is also a greater temperature drop. Neither the transmission through the metal nor that through the film on the steam side is directly affected by the water velocity, so that the

of An Investigation of the Air Lift Pump, by Davis and Weidner.[1] The quantity of water delivered rises to a maximum when the two fluids are equally mixed by volume, or the specific gravity is 0.5. The specific gravity falls off thereafter, due both to the increasing air volume and decreasing water volume. These experiments indicate that slip and friction change the relation of air volume and mixture velocity to nearer a square root law as compared to the theoretical cube root relation. The greater the proportion of air to water the less is the slip.

8 Slip is also influenced by the velocity, the ratio of air to water in the mixture remaining constant. Fig. 3, based on Fig. 20 of

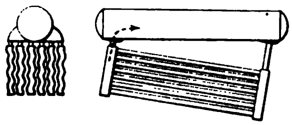


Fig. 4

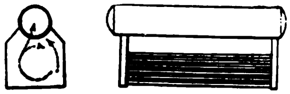


Fig. 5

Fig. 4 Babcock & Wilcox Type Boiler
Fig. 5 Heine Type Boiler

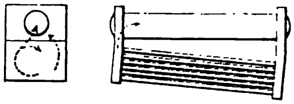


Fig. 6 Edge Moor Type Boiler

Three Types of Water Tube Boilers

Davis and Weidner's paper, shows the total head lost in both slip and friction for a constant ratio of air to water of 3, that is, for a mixture containing 25 per cent water and 75 per cent air, at all velocities. The head lost rises rapidly at low velocities but falls to a straight line at higher flows, whereas a friction head alone would rise approximately as the square of the flow. The loss in slip therefore decreases as velocity increases. The comparatively large loss at low velocities is particularly noticeable.

9 While the results of these tests are not reduced to general laws, they define the conditions essential to good circulation. These we can now apply to the circuit of the actual water tube boiler which contains a number of sharp turns and abrupt changes in flow area and is composed of a bundle of parallel tubes subjected to different loads, discharging into a common header. Three different

[1]Bul. of the Univ. of Wis., 1911, vol. 6, no. 7.

types, Figs. 4, 5 and 6, the Babcock and Wilcox, the Heine and the Edge Moor, respectively, are representative.

10 In these boilers the steam volume formed in each tube decreases as we pass upward from the furnace and the specific gravity of the mixture discharged increases. This results in an increasing heaviness of mixture as we progress from the bottom to the top of the front waterleg and tends to choke circulation in the bottom tubes.

11 In the second place, the cross sectional area of the front header being relatively large, an abrupt decrease in velocity occurs at the point of discharge of the bottom tubes, so that the steam

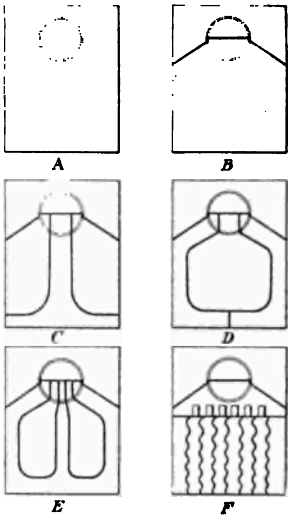


Fig. 7 Different Types of Headers Tested

tends to slip up without lifting the water. In the upper part of the header, the mixture comes from the tubes at relatively small velocity, and contains less steam, so that the conditions favorable to slip still exist. In fact, liberation of steam commences within the header, and the water tends to remain in place as in a large vessel—a tubular boiler, for example—instead of being pumped up into the drum. The head producing flow of water from the waterleg into the drum is created almost entirely by the tubes, and principally the bottom rows, whereas the rising steam bubbles in the waterleg should greatly assist the circulation in the tubes. Under these conditions, the higher the boiler the poorer may be the circulation; yet, in applying formulas of the kind given, it is assumed that the head producing

WATER TUBE BOILER CIRCULATION

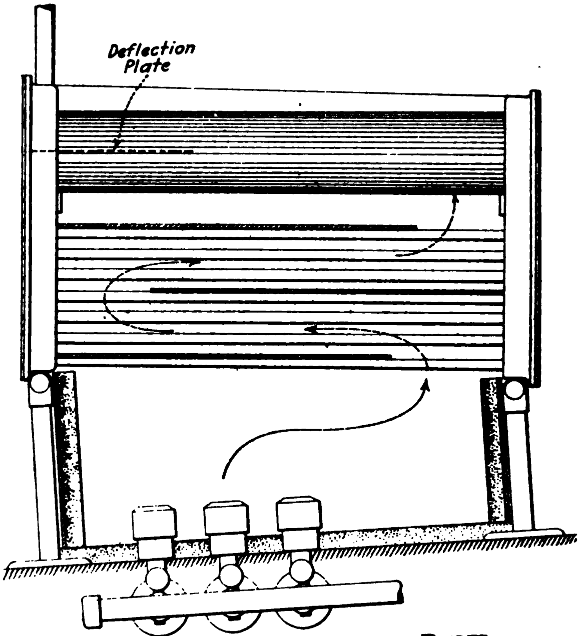


FIG. 8a MODEL WATER TUBE BOILER

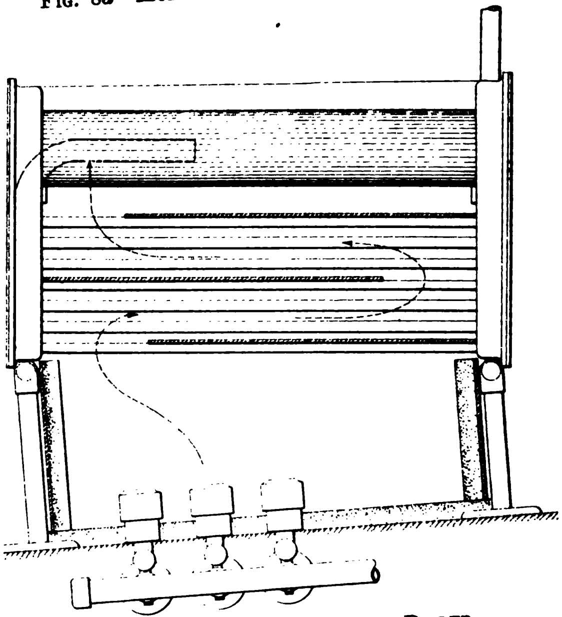


FIG. 8b MODEL WATER TUBE BOILER

velocity is measured by the height from centre of tube to water level, and increases with the height of the boiler.

12 In waterlegs of the types in Figs. 5 and 6, there is also a loss due to the internal currents, as will be shown. The sectional construction of Fig. 4 confines the flow to practically straight lines.

13 In all cases, there is a certain amount of resistance to circulation, principally that due to the sharp turn and constriction at the point of discharge of the front header into the drum, which is increased if a horizontal baffle or deflection plate, lying across the path of the steam and water, is used. A smooth large radius elbow is preferable. In the Babcock and Wilcox type, Fig. 4, the discharge occurs through nipples, and the reduction in area from the header

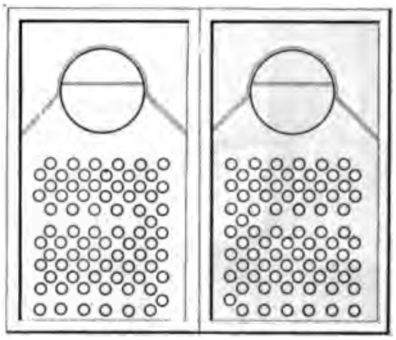


FIG. 8c MODEL WATER TUBE BOILER

to the nipple is considerable. In the Heine type, the discharge is through an opening in the drum of one half, one third, or less, of the flow area in the header. In the Edge Moor type, the header is carried up full width, but some head is lost since there must be sufficient hydraulic gradient to cause the flow from header into the drum which probably always acts as an open channel.

14 An expression for the velocity in any row of tubes, or in any other part of the circuit cannot be written since it would have to take all these indeterminate variables and coefficients into account, and therefore the effect of changes in design cannot be predicted with mathematical precision. It seems to follow, however, that the velocity in the bottom tubes can be increased if the front header is designed so that the discharge of steam and water from those rows is isolated and led up into the drum in its own section of the header, as shown in Fig. 7*C*, *D* and *E*. The light steam-water mixture discharged by the bottom tubes does not impinge upon a denser mass

of slow moving fluid in a large chamber but, instead, is segregated all the way up into the drum. The velocity of the mixture is maintained at its original figure so far as possible and abrupt changes in velocity are avoided, thus largely preventing the occurrence of slip. While only the same total inlet area to the drum is available, half or more is allowed for the discharge of the lower tubes and resistance due to constriction at this point avoided.

15 The circulation flues are carried up into the drum and

Fig. 9 Fig. 10

Fig. 9 Model Boiler Set Up for Test with Metal Cover Plates on Front and Glass Cover Plates on Rear, as shown in Fig. 10

Fig. 10 View of Model Boiler, showing Outlet Pipes with Separators on Each Half of the Boiler and Gages for Measuring Output

should make a smooth large radius bend, preserving constant cross-section. A horizontal extension of the flue in the drum, shown in dotted lines in Fig. 8, provides separate liberating space for the large volume of steam formed in the bottom tubes, which under those circumstances does not have to boil up through superimposed water.

MODEL BOILER FOR STUDYING CIRCULATION

16 In order to study circulation and the nature of the flow in different types of horizontal boilers, a model was constructed as shown in Figs. 8, 9 and 10.

steam condensation is completed, in which case only such drop of temperature as might be due to the steam flowing through the tube bank would exist. With the tubes properly arranged, this temperature drop would be comparatively small, at least at low vacuums and the attending low specific volumes.

However, if the mixture is to leave the tube bank at nearly its original temperature, a considerable percentage of the steam will remain to be compressed or condensed in the air pump. The objection to using the air pump for steam recompression is self-evident, while if a hurling water air pump is used in order to condense the steam, the temperature of the hurling water will be raised and the efficiency of the air pump impaired. If a large amount of water is used in order that the temperature rise in the pump may be small, a heavy expenditure of power to drive the pump will be encountered. Water handled by the circulating pump may have a temperature rise of 10 deg. fahr. and may be pumped against a head of 15 ft., whereas water handled by the air pump may have a rise of only a few degrees and the head pumped against will be 150 to 250 ft. To condense a pound of steam, 50 to 75 times as much power is taken by the air pump as compared with the circulating pump, for which reason it is not a good plan to allow much of the steam to pass to the air pump, which is what Mr. Ehrhart's claims really imply.

Paul A. Bancel (written). Replying to Mr. Ehrhart, the performance of the condenser from Josse's tests was cited to show conditions existing in a small condenser working at moderate vacuum with a considerable air leakage. The same conditions are noticeable, it is maintained, in large vacuum condensers even with small air leakage and large pump capacity (irrespective of the design of the pump), because due to its large specific volume, the steam cannot penetrate into the condenser and, due to the great rarification, the air cannot be removed except after some preliminary compression and concentration in the condenser.

The condition of temperature equivalent to pressure alleged by Mr. Ehrhart precludes any leakage whatsoever or else an air pump of such capacity that the air can be removed at immeasurably small partial pressure, together with an unavoidably large volume of steam.

With a *small* condenser heavily loaded under summer conditions

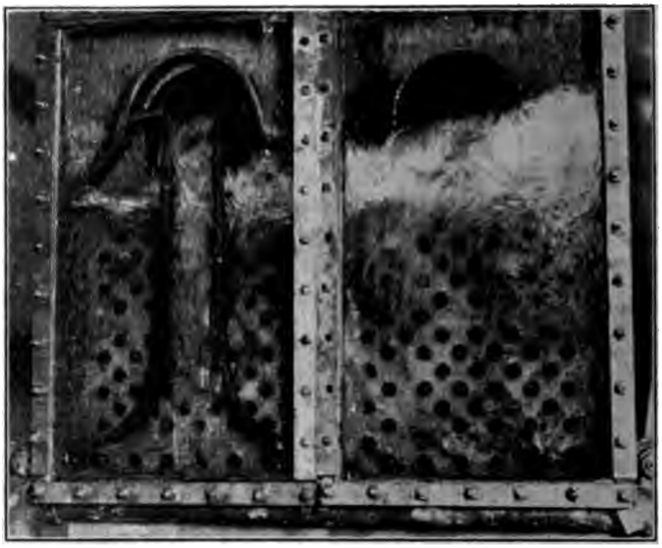


FIG. 11 SHOWING FRONT HEADERS; LOAD 85 PER CENT; SETTING AS IN FIG. 8b

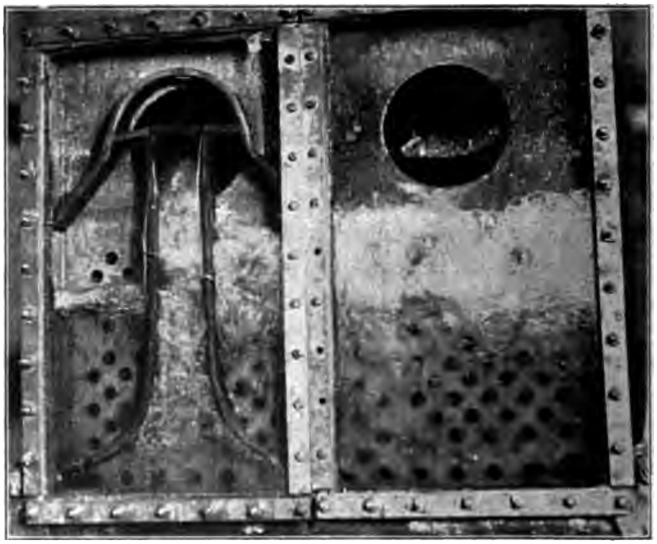
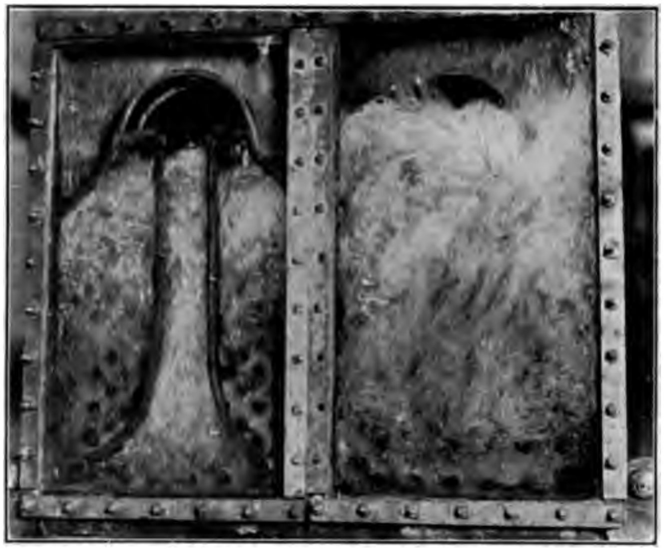
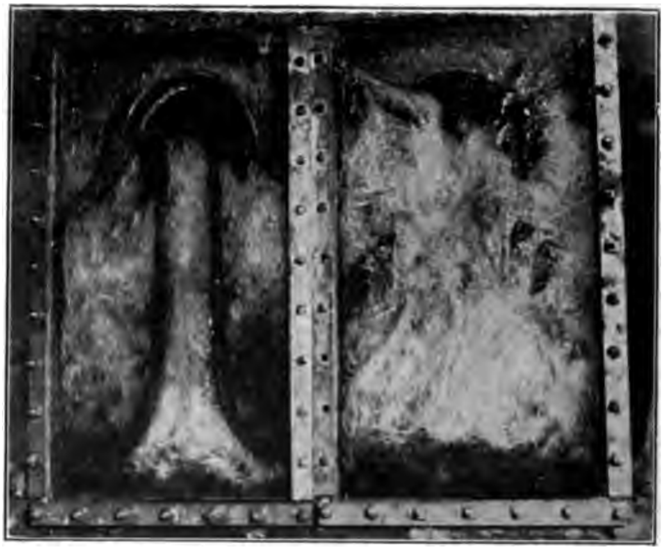


FIG. 12 SHOWING FRONT HEADERS; LOAD 85 PER CENT; SETTING AS IN FIG. 8b; WATER LEVEL BELOW NORMAL



...G. 13 SHOWING FRONT HEADERS; LOAD 275 PER CENT; SETTING AS IN FIG. 8b



...G. 14 SHOWING FRONT HEADERS; LOAD 450 PER CENT; SETTING AS IN FIG. 8b

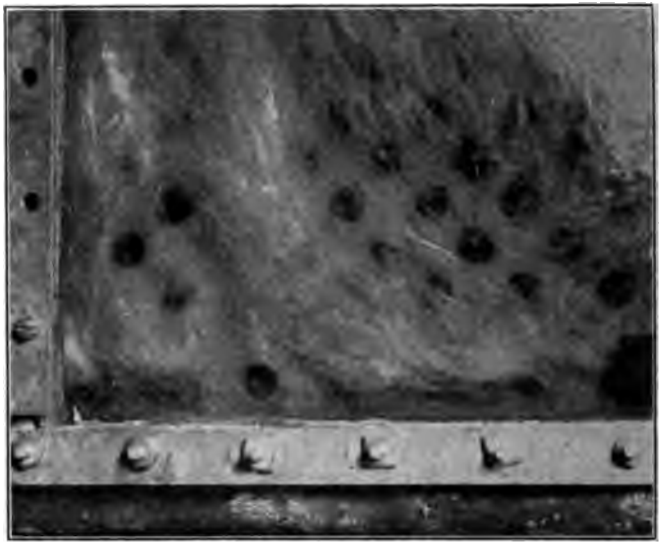
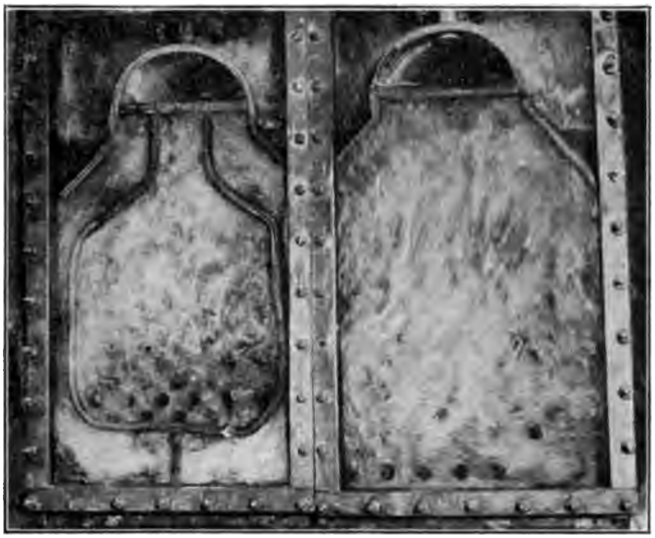


Fig. 15 Showing Lower Section of same Boiler as in Fig. 14; Loa 250 Per Cent; Setting as in Fig. 8b

Fig. 16 Showing Front Headers; Load 275 Per Cent; Setting as in Fig. 8a



IG. 17 SHOWING FRONT HEADERS; LOAD 500 PER CENT; SETTING AS IN FIG. 8a

IG. 18 SHOWING FRONT HEADERS; LOAD 450 PER CENT; SETTING AS IN FIG. 8a

of An Investigation of the Air Lift Pump, by Davis and Weidner.[1] The quantity of water delivered rises to a maximum when the two fluids are equally mixed by volume, or the specific gravity is 0.5. The specific gravity falls off thereafter, due both to the increasing air volume and decreasing water volume. These experiments indicate that slip and friction change the relation of air volume and mixture velocity to nearer a square root law as compared to the theoretical cube root relation. The greater the proportion of air to water the less is the slip.

8 Slip is also influenced by the velocity, the ratio of air to water in the mixture remaining constant. Fig. 3, based on Fig. 20 of

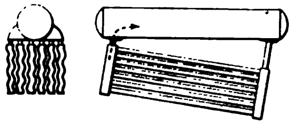


FIG. 4

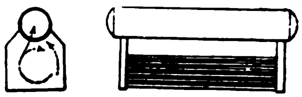


FIG. 5

FIG. 4 BABCOCK & WILCOX TYPE BOILER
FIG. 5 HEINE TYPE BOILER

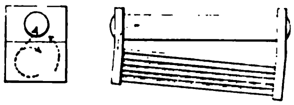


FIG. 6 EDGE MOOR TYPE BOILER

THREE TYPES OF WATER TUBE BOILERS

Davis and Weidner's paper, shows the total head lost in both slip and friction for a constant ratio of air to water of 3, that is, for a mixture containing 25 per cent water and 75 per cent air, at all velocities. The head lost rises rapidly at low velocities but falls to a straight line at higher flows, whereas a friction head alone would rise approximately as the square of the flow. The loss in slip therefore decreases as velocity increases. The comparatively large loss at low velocities is particularly noticeable.

9 While the results of these tests are not reduced to general laws, they define the conditions essential to good circulation. These we can now apply to the circuit of the actual water tube boiler which contains a number of sharp turns and abrupt changes in flow area and is composed of a bundle of parallel tubes subjected to different loads, discharging into a common header. Three different

[1]Bul. of the Univ. of Wis., 1911, vol. 6, no. 7.

Rear Headers Corresponding to Headers of Fig. 19 in Front; Load 450 Per Cent; Setting as in Fig. 8a

Rear Headers Corresponding to Designs of Fig. 19 in Front; Load 85 Per Cent; Setting as in Fig. 8a

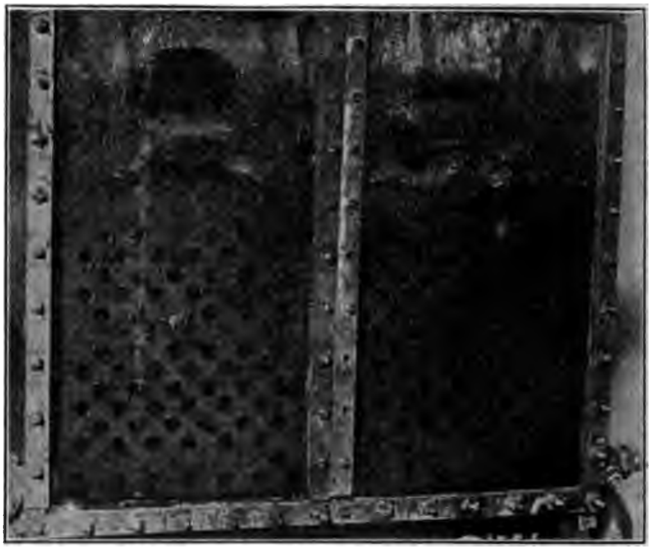


FIG. 23 REAR HEADERS CORRESPONDING TO DESIGN OF FIG. 16 IN FRON[illegible]
LOAD 85 PER CENT; SETTING AS IN FIG. 8a

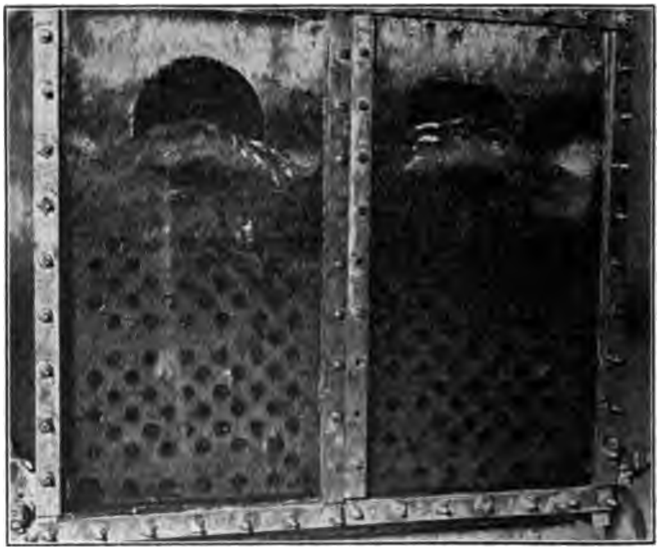


FIG. 24 REAR HEADERS CORRESPONDING TO DESIGN OF FIG. 18 IN FRONT;
LOAD 300 PER CENT; SETTING AS IN FIG. 8a

REAR HEADERS, CORRESPONDING TO DESIGN OF FIG. 11 IN FRONT; LOAD 450 PER CENT; SETTING AS IN FIG. 8b

REAR HEADERS CORRESPONDING TO DESIGN OF FIG. 7F AT THE FRONT HE LEFT HAND BOILER, AND THE DESIGN OF 7C AT THE FRONT OF E RIGHT HAND BOILER; LOAD 300 PER CENT; SETTING AS IN FIG. 8b of slow moving fluid in a large chamber but, instead, is segregated all the way up into the drum. The velocity of the mixture is maintained at its original figure so far as possible and abrupt changes in velocity are avoided, thus largely preventing the occurrence of slip. While only the same total inlet area to the drum is available, half or more is allowed for the discharge of the lower tubes and resistance due to constriction at this point avoided.

15 The circulation flues are carried up into the drum and

Fig. 9

Fig. 10

Fig. 9 Model Boiler Set Up for Test with Metal Cover Plates on Front and Glass Cover Plates on Rear, as shown in Fig. 10

Fig. 10 View of Model Boiler, showing Outlet Pipes with Separators on Each Half of the Boiler and Gages for Measuring Output

should make a smooth large radius bend, preserving constant cross section. A horizontal extension of the flue in the drum, shown in dotted lines in Fig. 8, provides separate liberating space for the large volume of steam formed in the bottom tubes, which under those circumstances does not have to boil up through superimposed water.

MODEL BOILER FOR STUDYING CIRCULATION

16 In order to study circulation and the nature of the flow in different types of horizontal boilers, a model was constructed as shown in Figs. 8, 9 and 10.

17 The boiler was built in two units, each complete with drum, headers and tubes. There were 75 half-inch outside diameter No. 18 B. & S. brass tubes about 2 ft. long, of a total surface of 17 sq. ft. in each unit. The headers were rectangular as in an Edge Moor boiler, and by inserting suitable strips of brass the design could be changed to represent the different constructions of Figs. *7A*-*7F*. Glass covers were placed on the front and back headers to permit observation of conditions and the taking of photographs. Rubber packing formed of split rubber tubing was placed on the edge of the strips under the glass, and this explains the seemingly large thickness of partitions.

18 The boiler was set with a slope of 1 in. to the foot over a gas furnace consisting of three large burners capable of burning 200 cu. ft. of gas per hr. One-inch asbestos lined the furnace and boiler casing. The baffling for the gases was of the parallel type in two passes. The boiler was set so that the gases entered at the rear of the bottom pass, Fig. 8*a*, and also with the relative position of furnace reversed, the gas entering the first pass directly in front, as shown in section of Fig. 8*b*. An induced draft fan drew the gases through the boiler.

PHOTOGRAPHS OF THE BOILER CIRCULATION

19 Photographs of the moving steam and water in the headers were taken at a speed of approximately 1/100 sec. with flashlight illumination. At first experiments were made with water tinted red and also with water containing powdered mica, but these proved of no value, and afterwards no coloring was used in the water.

20 When the circulation is poor and steam is slipping through the water the latter shows up transparent and the header appears almost empty, whereas where the steam and water are well mixed or emulsified, there is a distinct whitish color and marked contrast in the photographs.

21 Before a photograph was taken, the water was brought to the same level in each header without having the burners on. The level was slightly below the bottom of the drum in the front header as shown in Fig. 20. The camera was within a few feet of the boiler and the reduction was about half size, so that the movement of an inch on the original boiler equalled about half an inch in the photograph. A small group of bubbles moving at a velocity of 1 ft. per sec. would move 1/100 ft. during the exposure and make a streak 1/200 ft., or say 0.06 in., on the plate (8 by 10 in.). Wherever bub-

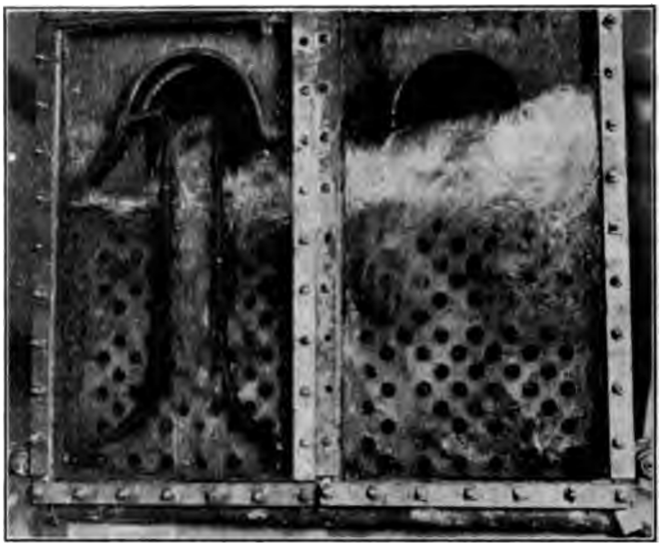


Fig. 11 Showing Front Headers; Load 85 Per Cent; Setting as in Fig. 8b

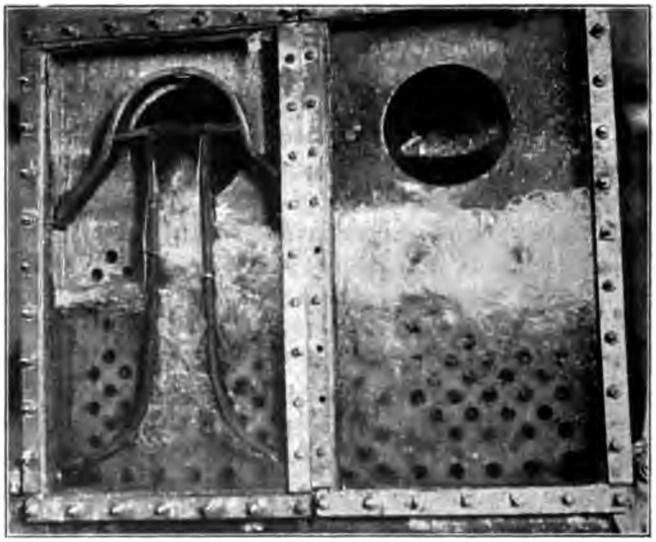


Fig. 12 Showing Front Headers; Load 85 Per Cent; Setting as in Fig. 8b; Water Level Below Normal

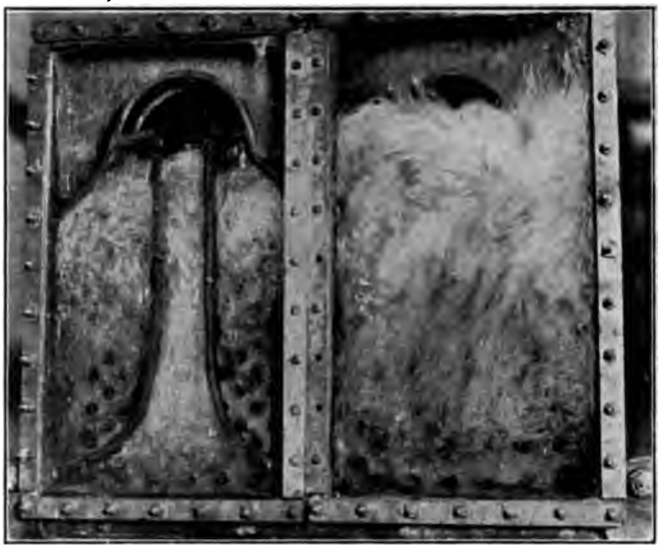


FIG. 13 SHOWING FRONT HEADERS; LOAD 275 PER CENT; SETTING AS IN FIG. 8b

FIG. 14 SHOWING FRONT HEADERS; LOAD 450 PER CENT; SETTING AS IN FIG. 8b

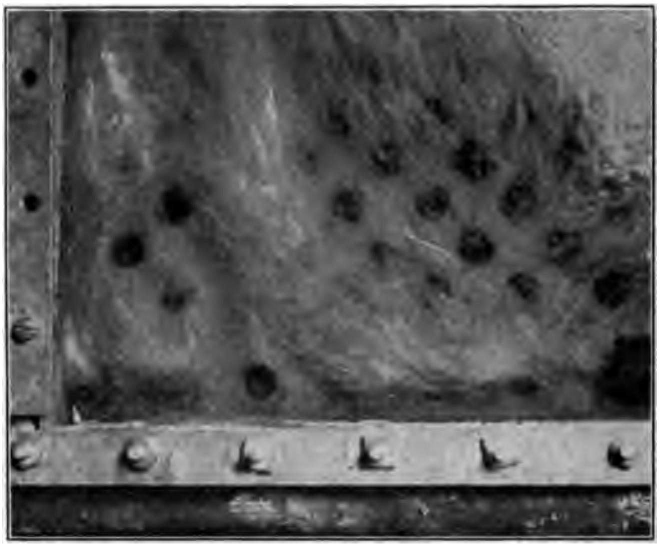


Fig. 15 Showing Lower Section of same Boiler as in Fig. 14; Load 250 Per Cent; Setting as in Fig. 8b

Fig. 16 Showing Front Headers; Load 275 Per Cent; Setting as in Fig. 8a

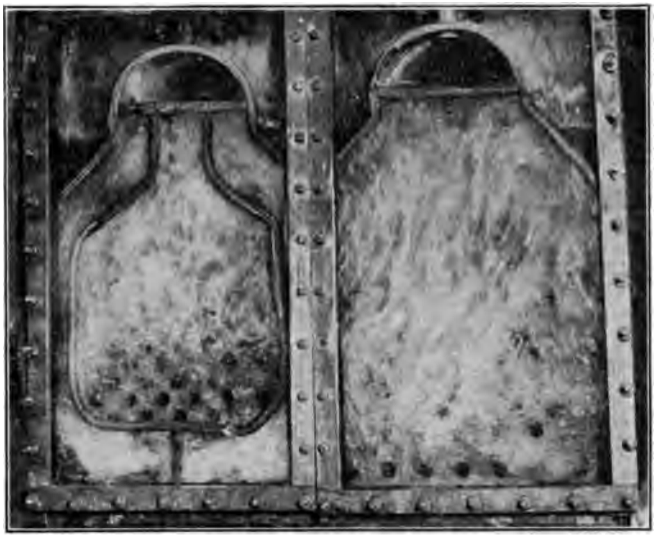


FIG. 17 SHOWING FRONT HEADERS; LOAD 500 PER CENT; SETTING AS IN FIG. 8a

FIG. 18 SHOWING FRONT HEADERS; LOAD 450 PER CENT; SETTING AS IN FIG. 8a

FIG. 19 SHOWING FRONT HEADERS; LOAD 450 PER CENT; SETTING AS IN FIG. 8a

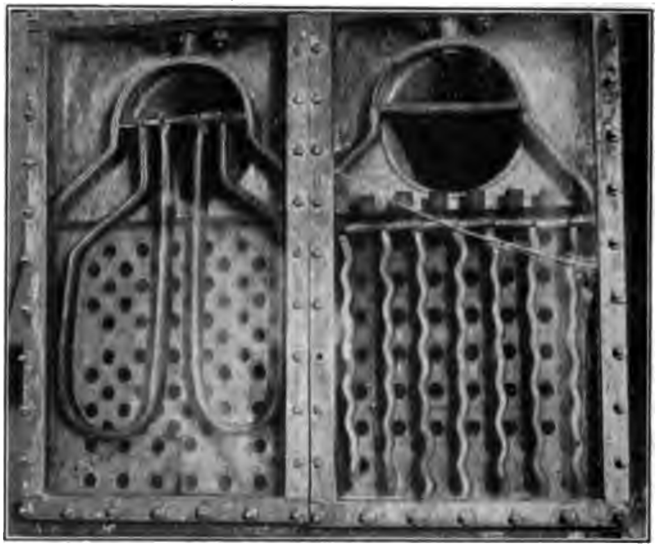


FIG. 20 BOILERS WITHOUT LOAD SHOWING TRUE WATER LEVELS, WHICH WERE ABOUT 1 IN. BELOW THE BOTTOM OF THE DRUM IN THE FRONT HEADERS, AND 1 IN. ABOVE THE BOTTOM OF THE DRUM IN THE REAR HEADERS

ing circulation in the bottom tubes, and on the right the square header.

25 In the left hand half of Fig. 11, the greater evaporation is obviously taking place in the bottom tubes connected into the circulation flue where there is a well directed upward flow. The upper tubes of the boiler discharge into their compartments of the header very quietly and the steam slips up through the water and is liberated in the header. The pumping action, as in an air lift, is shown clearly in the central flue. The discharge is vigorous enough to strike the horizontal deflection plate in the center of the drum.

26 The right hand half of Fig. 11 illustrates how slip occurs in a header of relatively large area; the steam slips up through the water and is liberated largely in the header as indicated by the wide belt of foam formed at the surface of the water, which is to be contrasted with the quiescent conditions on the left. The steam and water issued from the bottom tubes as a homogeneous mixture filling the bore of the tubes and preserved this state as it turned up, but after a short distance it seemed to be absorbed by the comparatively large mass of water in the header. The tubes carrying lighter loads discharged steam and water from the upper part of the tube only.

27 A rotary flow can be seen in the square header, Fig. 11. The current rises at the left and falls at the right, the velocity being low or next to nothing where the bubbles can be distinguished.

28 Fig. 12 was taken at the same load but with low water level to show the effect of the circulation flue in the left hand boiler. The bottom tubes of this boiler are discharging water and steam up the flue into the drum, whereas no water is delivered to the drum from other sections of the header or from the header on the right hand side. Return circulation occurs through the upper tubes.

29 Fig. 13 shows the boilers steaming at approximately 275 per cent load and Fig. 14 about 450 per cent load. The upper tubes of the left hand boiler are now working more actively, but the relatively small amount of steam generated is indicated by the fact that about half the rows of tubes can be seen through water, especially in Fig. 13, the steam slipping through without causing a flow of water. The flow in the circulation flue is smooth and full bore. On the right there is violent commotion with whirls and swirls in all directions. The steam and water rise nearly to the top of the drum which indicates the head needed to make the turn and discharge the mixture into the drum.

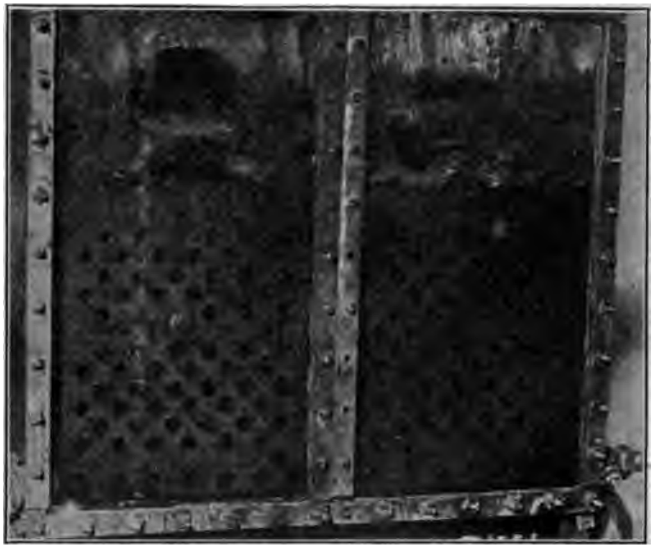


Fig. 23 Rear Headers Corresponding to Design of Fig. 16 in Front; Load 85 Per Cent; Setting as in Fig. 8a

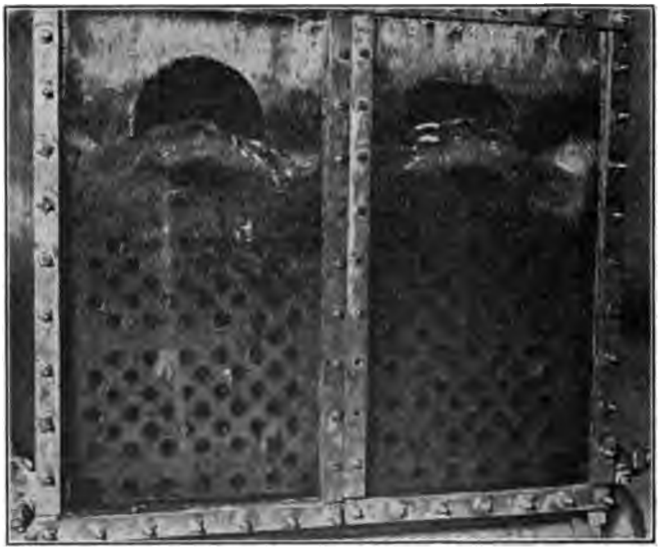


Fig. 24 Rear Headers Corresponding to Design of Fig. 18 in Front; Load 300 Per Cent; Setting as in Fig. 8a

FIG. 25 REAR HEADERS, CORRESPONDING TO DESIGN OF FIG. 11 IN FRONT; LOAD 450 PER CENT; SETTING AS IN FIG. 8b

FIG. 26 REAR HEADERS CORRESPONDING TO DESIGN OF FIG. 7F AT THE FRONT OF THE LEFT HAND BOILER, AND THE DESIGN OF 7C AT THE FRONT OF THE RIGHT HAND BOILER; LOAD 300 PER CENT; SETTING AS IN FIG. 8b

bles of steam appear to be at rest in the photographs, therefore, it may be assumed that their velocity was no more than 1 ft. per sec., while streaks indicate higher velocities.

22 Steam was generated at atmospheric pressure and the volume was measured by orifices in the outlet pipes on each half of the boiler. As the volume of steam at atmospheric pressure is very great compared to that at 200 lb. (for example) a low load in pounds per square foot per hour on the model corresponds to a high load in terms of steam volume. The equivalent loads given in connection

Fig. 27 Showing Rear Headers, same as in Fig. 26; Load 85 Per Cent; Setting as in Fig. 8b

with the pictures are based on a boiler pressure of 200 lb., at which the steam volume is about 0.08 that at atmospheric pressure.

23 A greater proportion of the steam was generated in the bottom tubes. The relative loads on different rows were probably about the same as the average in an actual boiler, wherein the distribution varies considerably depending on the furnace temperature.

24 Fig. 11 shows the circulation in the front headers at about 85 per cent of rating. On the left is a header with a flue for increas-

ing circulation in the bottom tubes, and on the right the square header.

25 In the left hand half of Fig. 11, the greater evaporation is obviously taking place in the bottom tubes connected into the circulation flue where there is a well directed upward flow. The upper tubes of the boiler discharge into their compartments of the header very quietly and the steam slips up through the water and is liberated in the header. The pumping action, as in an air lift, is shown clearly in the central flue. The discharge is vigorous enough to strike the horizontal deflection plate in the center of the drum.

26 The right hand half of Fig. 11 illustrates how slip occurs in a header of relatively large area; the steam slips up through the water and is liberated largely in the header as indicated by the wide belt of foam formed at the surface of the water, which is to be contrasted with the quiescent conditions on the left. The steam and water issued from the bottom tubes as a homogeneous mixture filling the bore of the tubes and preserved this state as it turned up, but after a short distance it seemed to be absorbed by the comparatively large mass of water in the header. The tubes carrying lighter loads discharged steam and water from the upper part of the tube only.

27 A rotary flow can be seen in the square header, Fig. 11. The current rises at the left and falls at the right, the velocity being low or next to nothing where the bubbles can be distinguished.

28 Fig. 12 was taken at the same load but with low water level to show the effect of the circulation flue in the left hand boiler. The bottom tubes of this boiler are discharging water and steam up the flue into the drum, whereas no water is delivered to the drum from other sections of the header or from the header on the right hand side. Return circulation occurs through the upper tubes.

29 Fig. 13 shows the boilers steaming at approximately 275 per cent load and Fig. 14 about 450 per cent load. The upper tubes of the left hand boiler are now working more actively, but the relatively small amount of steam generated is indicated by the fact that about half the rows of tubes can be seen through water, especially in Fig. 13, the steam slipping through without causing a flow of water. The flow in the circulation flue is smooth and full bore. On the right there is violent commotion with whirls and swirls in all directions. The steam and water rise nearly to the top of the drum which indicates the head needed to make the turn and discharge the mixture into the drum.

30 Fig. 15 shows the lower right hand corner of the same type header to a larger scale, with the boiler steaming at an equivalent load of 250 per cent. The circulation is upward on the left and downward on the right. The rush of circulation to the left of the header is presumably due to the fact that the heat is greatest in the center of the furnace, which corresponds to the left hand side of this header.

31 Fig. 16 shows Heine type headers in both parts of the boiler and double circulation flues on the left, these rising along the sides of the header in extensions which did not connect with any upper boiler tubes. No heavy mixtures were therefore discharged into the light steam water emulsion rising from the bottom tubes. The load was about 275 per cent. The discharge of the upper tubes of the left hand boiler slips through the water and the mixture in the center, near the top, is moving slowly, whirling in all directions. The right hand picture illustrates again the tendency for internal circulation in an open header; there is a marked rotary flow and downward current on the right. A closer study of the figure will show what seem to be light shadows to the left of the tube ends in the bottom row. This is the discharge of steam slipping through the water. Fig. 17 shows the same boiler steaming at 500 per cent load.

32 Fig. 18 shows the Heine type header on the right and the single circulation flue on the left, at about 450 per cent load. The white streak across the right hand header is a cemented crack in the glass. Numerous small eddy currents or whirls can be distinguished. Bubbles that are nearly stationary are distinguishable all along the lower section, directly in the center and in the upper right hand corner—these forming a vortex or whirl in a clockwise direction.

33 Fig. 19 shows a third arrangement of circulation baffle with three flues on the left, and a Babcock and Wilcox type header on the right; the load was about 450 per cent. Fig. 20, at no load, shows the construction of the header more clearly. The flow in the sections of the Babcock and Wilcox header was directly upwards, with no cross currents or whirls and only a slight sinuous movement, but was throttled by the nipples at the top of each section. Stationary bubbles can be distinguished in many of the headers. At low loads the slip through the headers was similar to that in the previous types, the bubbles rising through the water without causing discharge from the nipples.

34 Fig. 21 is a view of the rear headers when steaming at the

same rate. The left hand header (corresponding to the Babcock and Wilcox type on the right in front) is boiling backward from the upper tubes, causing a fluctuating and higher water level. The white mark across the bottom of the right hand drum is caused by the splashing of the water against the glass.

35 Fig. 22 shows the rear headers at about 85 per cent load, a slight boiling back being noticeable here also. This figure shows another feature, however—the left hand drum is dry due to the poor circulation of the front headers into the drum, whereas the right hand drum is receiving water which can be seen emptying into the rear header. The steam slips through the water in the front headers on one side, without pumping water into the drum. The true water levels are equal, but about 3 in. low.

36 Reversal of boiling also occurred with other types of headers. Fig. 23 shows the boilers with front header as in Fig. 16, steaming at 85 per cent. A number of tubes can be seen to be discharging steam. This occurred intermittently and would appear first at one tube and then another. Fig. 24 shows the rear of the boilers shown in front in Fig. 18, steaming at about 300 per cent load.

37 The back boiling was eliminated by reversing the boiler setting, that is, setting the boiler over the furnace so that the opening at the end of the lower horizontal baffling came at the front and higher end of the boiler, near the front header, as shown in Fig. 8*b*.

38 Fig. 25 shows the rear headers when the boiler was set in this manner, and the load was about 450 per cent. The rear headers were perfectly clear, and no discharge occurred from any of the tubes. The front headers were of the design shown in Figs. 11 to 15 which figures were also taken with the flow of hot gases entering the tube nest at the front.

39 Figs. 26 and 27 show the rear headers with the same setting, but with the Babcock & Wilcox type header at the front of the left hand boiler and the design of Fig. 7*C* at the front of the right hand boiler. The nipples of the sectional header were reduced one half in area by inserting sleeves. Fig. 26 shows a load of 300 per cent. While the true water levels were exactly the same, the left hand header shows a higher surface and also considerable commotion at the surface, due to reversal of circulation through the upper tubes. The water level fluctuated up and down with a swing of several inches. Fig. 27 shows the same boilers with 85 per cent load. The true water levels were equal but when steaming the

level in the left hand unit fluctuated, rising and backing up the water in the drum and then falling.

PRESSURE DIFFERENCE TESTS

40 Tests were made in which the difference in pressure between the bottoms of the front and rear headers was measured. The boilers were set with the hot gases entering at the front. A mercury U tube with legs at a small angle from the horizontal was used for these readings, which were then translated into equivalent inches of water. The gage was below the bottom of the boiler, as in Fig. 28, connected to F and B. The differential reading was therefore the pressure difference between F and B^1. The pressure head from B to B^1, less the weight of the mixture in the tube, was also causing circulation.

41 The pressure at F is equal to the sum of the heads due to the

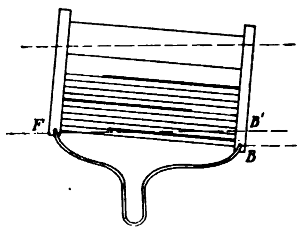


FIG. 28 POSITION OF GAGE FOR PRESSURE DIFFERENCE TESTS

weight of the fluid in the header and the head required to produce upward velocity and overcome friction. A decrease in this pressure, or increase in the differential pressure, indicates that the decrease in static head due to the mixture becoming lighter more than offsets the greater head required to produce velocity and overcome resistance.

42 Fig. 29 shows the pressure difference, at increasing loads, for the plain square header of the Edge Moor design, Fig. 7*A*. Fig. 30 shows the same relation for the Heine type, Fig. 7*B*.

43 Fig. 31 shows the pressure difference with the Babcock & Wilcox type header, Fig. 7*F*. Figs. 32, 33 and 34 show readings when circulation flues, as shown in Fig. 7*C*, *D* and *E*, were used in the front header. The curves for headers, Fig. 7*D* and *F*, practically coincide. Fig. 7*C* gives lower values at the higher loads and slightly higher at the low loads. This is probably due to the fact that with Fig. 7*C* the discharge from the bottom tubes is mixed with the dis-

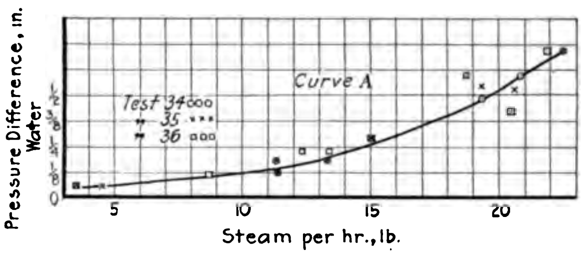


FIG. 29 TESTS WITH HEADER OF FIG. 7A

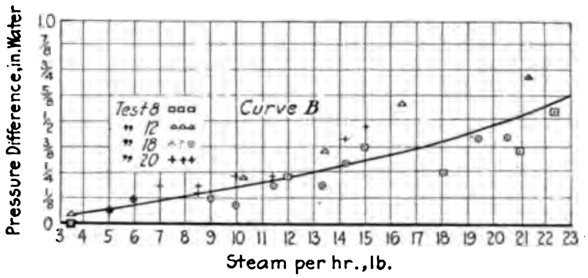


FIG. 30 TESTS WITH HEADER OF FIG. 7B

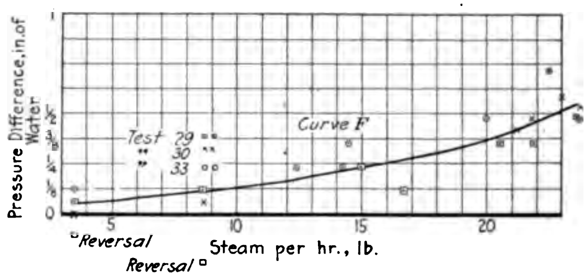


FIG. 31 TESTS WITH HEADER OF FIG. 7F

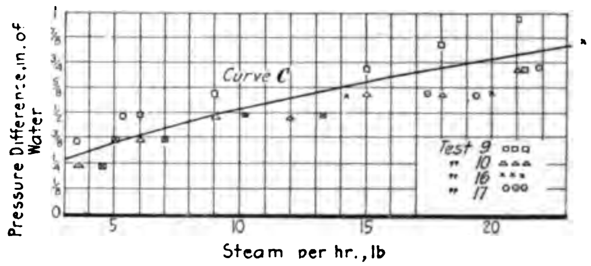


FIG. 32 TESTS WITH HEADER OF FIG. 7C

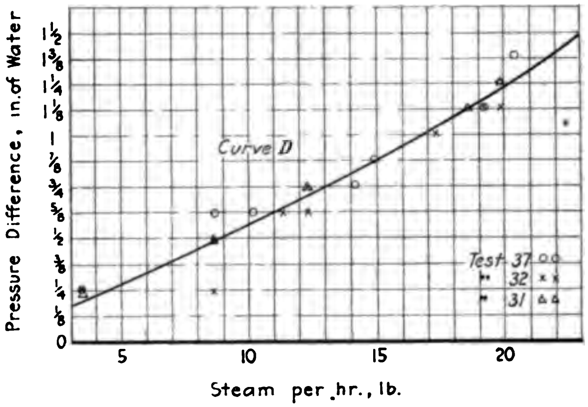


Fig. 33 Tests with Header of Fig. 7*D*

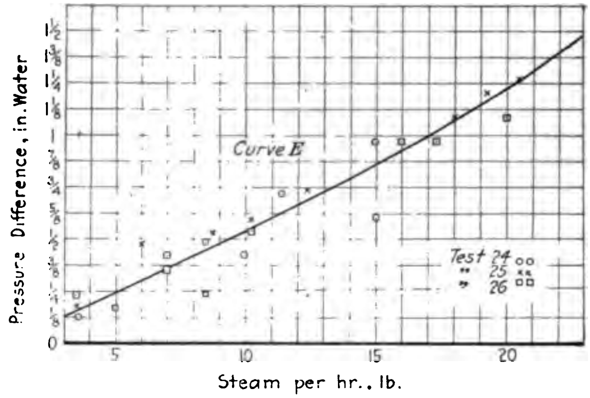


Fig. 34 Tests with Header of Fig. 7*E*

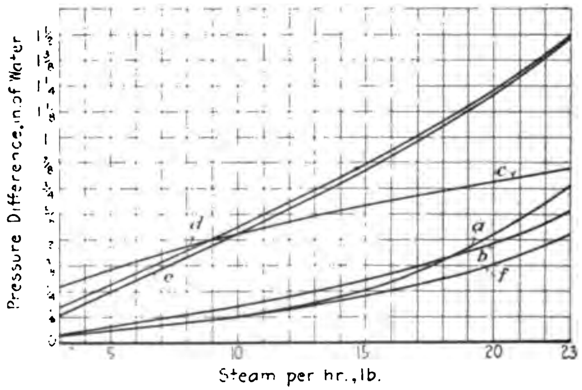


Fig. 35 Combined Chart of Figs. 29 to 34

charge from a considerable number of upper tubes in the boiler, which also discharge into the central part of the front header.

44 All the curves are grouped for comparison in Fig. 35. The circulation flues increase the pressure difference front and rear of the bottom tubes and add to the circulation created in the tube itself due to its slope. The effect of the nipples of the Babcock & Wilcox construction in throttling circulation as compared to the plain header construction is not noticeable until about 300 per cent load is reached.

CONCLUSIONS

45 Increased circulation in the lower rows of tubes of a horizontal boiler can be obtained by constructions which segregate the discharge from those tubes up through the front header into the drum, and by designing the circuit to prevent slip and to offer the least resistance, particularly at the entrance to the drum.

46 The velocity of circulation of a steam-water mixture increases about as the square root of the steam volume or load.

47 The volume of water circulated increases at first as the steam volume increases, reaches a maximum, and then decreases. The maximum load on a tube should not exceed that coincident with maximum water delivery.

48 The tendency for reversal of circulation is lessened by setting the boiler so that the gases strike the tube bank at its higher end.

49 Boiling back, particularly from the upper tubes, takes place if the discharge area at the junction of header and drum is constricted.

DISCUSSION

George L. Fowler said that a few years ago he had made some investigations of the circulation in locomotive boilers, and had also watched the apparent circulation in some very finely constructed models of locomotives; and his whole experience led him to be very cautious about accepting the indications of circulation in a model.

He understood that the experiments with the model under consideration were made at atmospheric temperature. If the evaporation per square foot of heating surface were to be the same as in a full-size boiler, naturally in a boiler working under 100 or 200 lb. per sq. in. the bubbles would be smaller than they would be in a model boiler, on a proportion of perhaps 1 to 4 in diameter.

Now, the water spaces in a model boiler are small also, and proportionately small in comparison with the full-size boiler of the same general dimensions and proportions, and he thought the whole gist of the argument simmered down into what the author stated, that the rate of circulation is dependent upon the size of the bubble of steam passing through the boiler. Therefore the circulation would be very much more rapid in the model than in the full-size boiler. He originally had the idea that the circulation in a locomotive boiler was torrential, especially as investigation had shown that the rate of evaporation in the firebox was very much higher than it was in the tubes.

In his own experiments he had started with the idea he would have to provide for the measurement of a circulation of 400 or 500 ft. per sec., but after thinking the matter over, mentally came down to providing apparatus for measuring 125 to 150 ft. per sec. At about this time he saw an advertisement, in the form of a large inverted globe partly filled with water with air bubbles coming up through it. He timed these bubbles and found that they rose at a rate of about 4 ft. per sec. He reasoned that steam bubbles should not come up through water any faster than these air bubbles did.

By the time he started his investigations he had arrived at the conclusion that if he provided for about 8 or 10 ft. per sec., he would be about right. As a matter of fact, the highest circulation he got in a locomotive firebox, when the evaporation was of the rate of 12 lb. of water per sq. ft. of heating surface per hour, was about 2 ft. per sec. The condition of a locomotive boiler seems to be that we have a slow, gradual movement through the shell backwards and then some agitation, and the flowing up, on the inside sheet, and then a downward movement on the outside sheet, but the general tendency is back, and just about enough movement to supply evaporation, apparently.

Some recent work on full-size boilers corroborated that statement, he said. The temperature in the lower corner of the water leg at the front was never within 75 deg. of the temperature of the steam when the engine was working full. Perhaps it came within 50 degrees, but it was markedly below, whereas with that slow movement of water back through the firebox, in the back corner of the waterleg down close to the mud ring, there was a temperature up to that of the steam. These experiments led him to be rather skeptical in regard to accepting any demonstrations from a model, where necessarily the water spaces are quite limited and the steam bubbles correspondingly large. That there is a general correspondence between a model demonstration and a full-size boiler demonstration he did not doubt, but from a quantitative standpoint, he questioned the value of the model investigation described.

William Kent. The tests of the model boilers described by Mr. Bancel are interesting, but it is difficult to draw any conclusions from them that will be of any service in the design of full-size boilers. Some years ago the speaker had occasion to assist in the tests of a quarter-size model of a Rust vertical water tube boiler. The tubes were about 1 in. diameter and 5 ft. long. The ends of both upper and lower drums were fitted with discs of plate glass, and to the bottom end of each tube a slender white thread was attached by means of a fine wire so that the threads would show the direction of the circulation. Some surprising results were obtained. When the boiler was steaming at a moderate rate some of the tubes of the front bank would be carrying water upward while others were carrying it downward, but the upward currents would often change from one tube to another without any apparent cause.

Experiments on model boilers of the inclined water tube type cannot reproduce the conditions that exist in a full-size boiler, for in the latter there is at least 5 ft. distance between the water level in the drum and the highest point of the bottom row of tubes. This would make a difference of pressure of over 2 lb. per sq. in. if the water was at rest, and a greater difference if the water was in motion, on account of the frictional resistance of the upward passage. This extra pressure in the lower tubes would tend to prevent the generation of steam in them, and it is probable that they carry only water which is at a higher temperature and pressure than that in the overhead drum, and which begins to generate steam only when it passes into the region of lower pressure in the upper part of the headers and in the drum.

John C. Parker. The author says: "In addition, it is generally believed that a high velocity of circulation will retard scale formation and thus indirectly reduce the resistance to heat transfer and prevent overheating of the tubes." Some seventeen years ago I visited our late President, Prof. R. H. Thurston, at Cornell, and made the statement to him that the circulation was so rapid in our boiler that it would carry the solid matter along like dust in a strong wind and prevent scale formation. He said it would carry the insolubles, but not the solubles, and it turned out that way. Our experience has been that scale forms wherever the steam is made, and that if there is any carrying along of the scale it is after it is formed and then cracked loose.

It is quite desirable in designing a boiler to get the most rapid circulation possible. The strongest flow would be secured if we could get nothing but steam in the upcast and nothing but water in the downflow tubes. Complete evaporation would then be secured in the lowest tubes providing there were sufficient heating surface.

ARTHUR M. GREENE, JR. wrote that the experimental part of Mr. Bancel's paper was interesting in its qualitative nature but he could not see where quantitative results could be obtained from it.

In the rates of radiation given in the first paragraph, the author does not state the assumption on which the results are obtained. If the Steffan-Boltzmann law has been used with no corrections it should be so stated.

The formula $v = \sqrt{2\,g\,H}$ is only true for a free fall. The true formula is

$$v = \sqrt{\frac{2\,g\,H}{1 + z}}$$

in which z is equal to the sum of loss coefficients and is quite large in the case under discussion. The formula

$$v = \phi \sqrt{\frac{v\,h}{A}}$$

is not a balanced equation of time and distance unless ϕ contains these elements; ϕ must be variable and the variations must be great.

A. A. CARY agreed with Professor Kent as regards caution needed in drawing conclusions from the action of steam and water occurring in small glass models, although he had very frequently found that the action shown gave a good indication of what actually occurred in the full-size boiler. He said that he had had considerable experience in testing with glass models and in verifying the action they indicated by making tests in the full-size boiler. This experience taught him that the glass model, if properly constructed, would generally show the underlying principles of action occurring in the full-size boiler, but care should be exercised in drawing conclusion from the action in the model until verifying tests were made in the boiler itself.

Hosea Webster (written). The photographs reproduced in Mr. Bancel's paper are interesting, and the aim in his experiments is commendable. Unfortunately, however, he has based on the results of his experiments conclusions which are not warranted.

Small models may be used to show certain general features of circulation; for example, the first of the classical experiments made by Yarrow were with small models, but Yarrow was careful to make experiments finally on full-size boiler sections, to determine what actually occurred in his boiler. Our experience has shown conclusively that the results secured with small models of boilers may not hold for larger boilers built on the same plan. The difference between the boiling point of the water at the bottom and that at its highest level in the boiler is greatly different in a small model boiler from that which exists in a commercial boiler, and has an important effect on the circulation.

A correction is made to determine the circulation characteristics of a commercial boiler running at 200 lb. pressure by multiplying the actual percentage of rating developed by the model boiler by a factor of about 12, with the idea that the bubbles of steam in the water in the boiler at 200 lb. pressure will have about $\frac{1}{12}$ the volume of the bubbles of steam at atmospheric pressure which existed in the model, running at 40 to 50 per cent of its rating. This would be a dangerous conclusion, as, following this line of reasoning, a boiler run at 600 lb. working pressure would have the same characteristics respecting circulation when run at 1500 per cent of its rating as the model boiler when run at 40 per cent to 50 per cent.

The curves showing the pressure difference at the bottom of the front and rear headers, given in Figs. 29 to 35, are made the basis of certain conclusions respecting the circulation in the lower rows of tubes. This pressure difference is not a measure of the circulation in the lower rows of tubes, as the density of the column of steam and water in the tubes is one of the governing factors.

Apparently, the pressures given in these curves were taken near the outer walls of the models, or, in the case of the right hand boiler in Fig. 19, for the narrow lane on the right hand side of the boiler. It is evident, from the appearance of the photographs, that the amount of steam produced in this lane was considerably less than the average amount for all of the lanes. In Fig. 31, therefore, the ordinates representing the various observations should be moved to the left, and the curve would be given a greater inclination thereby.

Again, the method of measuring the pressure itself was not as exact as it should have been. As the pressures were measured with a mercury column, it follows that the actual difference in levels in the mercury column corresponding to ½-in. water column would be about 0.036 in., and an error of 0.01 in. head in measuring the difference in pressure by the mercury column would have amounted to about 30 per cent of error.

These experiments were made on a model boiler, the drums and tubes of which were inclined at an angle of about 4½ deg. with the horizontal. The water level was carried below the bottom of the drums. The flow of gases in all experiments was parallel with the tubes.

It is unfortunate that adjustments for comparisons of different types of headers were made at the front end of the boilers only.

While the experiments are interesting, they certainly are hardly conclusive with regard to types of boilers in which the water level is carried normally at the center line of horizontal steam drums connected at each end through similar headers to tubes which are inclined more than 4½ deg., and across which, rather than parallel to which, the furnace gases travel.

The lack of tube troubles in many boilers with the latest types of furnaces, daily operated at from three to often over four times their normal rating, indicates that in present day water tube boiler designs, ample provision is made for efficient circulation.

The Author. In reply to the members who have questioned the value of experiments on models, the author wishes to point out that in the solution of problems involving the flow of fluids, that is in hydrodynamics and aerodynamics — the use of models is invaluable.

The fundamental data of the subject of aerodynamics have been derived very largely from experiments on models. Langley's figures for model planes, modified by Lilienthal's experiments, provided the information from which the earliest flying machines were constructed. The work of Lancaster, Bryan and Eiffel was based largely on experiments with models.[1] As to ship propulsion, the testing of models of ships has been carried on for years.

The circulation in a water tube boiler is a highly complex problem, involving not only a complicated circuit with tubes working at different loads, discharging into a common header, but also the movement of two fluids, steam and water, at dissimilar and varying velocities. In quoting results of researches with a model, the writer made no attempt to predict or state the circulation or the pressures influencing circulation in the various types of actual boilers, confining his remarks to the model. His knowledge of the laws of similitude connecting a model water tube boiler and its prototype when circulating non-homogeneous mixtures of steam bubbles and water, is not sufficient to permit of such predictions, but he sees no reason why the results obtained should not be similar in kind, at least, to those to be obtained in an actual boiler. He does not believe that "the circulation of all boilers is good enough," as stated by Professor Kent, first, because cases of tube failures, some of which caused loss of life, have occurred where the tubes were new and it was known that they were free from scale; second, because boiler loads and furnace temperatures are constantly increasing, and third, because the investigations undertaken indicate that present designs do not conform in several respects to the necessary requirements for the maximum delivery of water through the tubes nearest the fire.

Replying to the point raised by Mr. Fowler regarding the size of the bubbles in an actual boiler as compared to those in the model, experiments have shown that the size of the bubbles seems to be independent of the pressure, depending on the conditions of ebullition and the character of heating surface. See experiments on a small boiler in which pressures up to 12 kg/cm were carried, by M. Emanaud, Le Génie Civil, January 6, 1911.

Professor Kent states that "it is doubtful whether there is any steam at all in the lower tubes." The 2-lb. pressure calculated by Professor Kent would amount at 200 lb. boiler pressure to about 0.8 deg., and for every pound of steam formed by the "water bursting into steam" in the drum, there would have to be about 1000 lb. of water passing through the tube. For a tube working at 200 lb. of steam per sq. ft. per hr. evaporation, this would call for approximately 10 cu. ft. flow per second or a velocity through the tube of

[1]Bairstow, The Laws of Similitude, Aeronautical Society of Great Britain, Feb. 2, 1913. See Engineering, London, Feb. 14, 1913.

about 150 ft. per sec. Experiments show that the velocity of water entering a tube is on the order of 5 ft. per sec.

The statement made by the author regarding scale formation and referred to by Mr. Parker was not advanced except with reservation. Experimental evidence on the effect of water velocity on scale formation not only in boilers but feed heaters, evaporators, condensers, etc., would add to the value of the present discussion.

Professor Greene has pointed out that ϕ in the equation in Par. 6 "must be variable and the variation must be great." In fact, the variation is so great as to make the expression practically useless, and it is for that reason in part, that the model experiments were undertaken. The conditions as to the existence and effect of slip have not heretofore, to the author's knowledge, been investigated. In the classic lecture on circulation by Mr. Babcock at Cornell University, the speed of circulation was calculated neglecting resistance, by the use of the equation $v^2 = 2\,g\,H$; H being determined by assuming an homogeneous mixture of steam bubbles and water and freedom from slip throughout the circuit.

Mr. Webster seems to fear that the results observed in the model experiments would not hold for large boilers because of the greater difference in pressure at the bottom and top of an actual boiler, but gives no explanation.

As to the figuring of the loads, it was believed that the equivalent loads given, for example, 450 per cent in connection with Fig. 14, were nearer correct than the actual heat load, but in any case, this is a matter which does not affect the main arguments of the paper regarding the factors conducive to good circulation.

As pointed out by Mr. Webster, the pressure due to the slope of the tube itself has to be considered. This is mentioned in Par. 44.

Regarding the pressure existing across the bottom of the header at the same elevation, experiments were made in which a probe was shifted back and forth and no variation in pressure was found.

As stated in Par. 40, the mercury U tube had legs at a small angle from the horizontal, and this made the readings considerably more accurate than pointed out by Mr. Webster.

In his last paragraph Mr. Webster intimates that attempts to improve upon the circulation in water tube boilers are unnecessary because of the perfection of present day design. A recent article in *The Locomotive,* published by the Hartford Steam Boiler Inspection and Insurance Co. lists 20 cases of tube ruptures in water tube boilers.

No. 1517

UNIQUE HYDRAULIC POWER PLANT AT THE HENRY FORD FARMS

By Mark A. Replogle, Akron, Ohio
Member of the Society

The hydraulic power plant at the Henry Ford Farms, Dearborn, Mich., contains two turbines designed to develop 85 h.p. each at 110 r.p.m. under 8-ft. head, together with electric generators. The plant supplies current for light, heat and power for Mr. Ford's residence, for the village pumping station, and for the miscellaneous requirements of the farms. It was built, to operate under somewhat unusual conditions of head and contains features which the author believes to be novel and unique.

2 The plant is located on the river Rouge, near its junction with the Detroit River, which latter varies considerably with the levels of the Great Lakes. The Rouge River is subject to the widest extremes in flow. At the point of location of the plant is a dam over which the normal run-off is 100 sec-ft. There are short intervals, however, when the run-off is so great that it fills the valley with water so that no sign of the dam is apparent; at these times the head over the dam is completely destroyed, but the condition lasts only a few hours until the high water wave has passed. There are longer periods when the level is affected by the Great Lake conditions causing back water in the tail race and consequent lowering of the head for days at a time. Also, there may be weeks of surplus flow at semi-high head. In fact, the conditions may be summarized as

a Low water with normal head

b Low water with head lowered by Great Lake conditions

c Normal flow under variable heads

d High water with approximately half normal head, which is also variable and dependent upon Great Lake conditions

e Very high water, but with head almost destroyed

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

3 High water, with head varying from 2 to 5 ft., may persist for weeks at a time, so that provision to meet this condition was necessary. The condition of very high water but practically no head only obtains for a few hours at a time, and no adequate provision can be made for it, except the steam reserve.

4 To secure the maximum power effects and constant generator speeds through such a wide range of heads, and without employing a multiplicity of units with variable speeds, was quite a problem, and a description of its manner of solution should be of interest to engineers concerned with hydraulic power plant design.

THE POWER HOUSE

5 The power house, an adjunct to Mr. Ford's private laboratory, is a concrete structure. The foundation consists of a monolith of reinforced concrete. Conditions made it necessary to floor the head race, power house site and tail race with concrete for several feet in thickness. The head race, 31 ft. wide and 10 ft. deep, was completely arched over for approximately 150 ft., and the tail race was similarly arched, both for architectural reasons.

6 The power house is shown in cross section in Fig. 1. It is a 3-story building, the lower floor containing the two turbines, the middle containing the auxiliary apparatus for controlling the turbines, and the upper the two dynamos, the governors, the switchboard and one steam reserve unit. The turbine floor is divided into three separate penstocks, connected by an overhead gallery.

THE POWER PLANT

7 Ordinary precaution led to the provision of two units, each having the capacity of the normal flow of the stream under 8-ft. head. The speed of 110 r.p.m. was established for full head conditions and this speed was maintained at all heads, varying from 1 ft. up to the full head of 8 ft. Herein lies the unique feature of this plant—the maintenance of full speeds and good power through a wide range of low heads and in the face of the condition that, under ordinary settings, the turbines employed require slightly over 4 ft. head to bring them to maximum speed at full gate and no load conditions.

8 The turbines are of the vertical type, designed, as stated, to develop 85 h.p. each on the turbine shaft when operating under 8 ft. head at 110 r.p.m. They have wicket gates, operated by draw rods from a gate shaft reaching directly to the governor arm. The

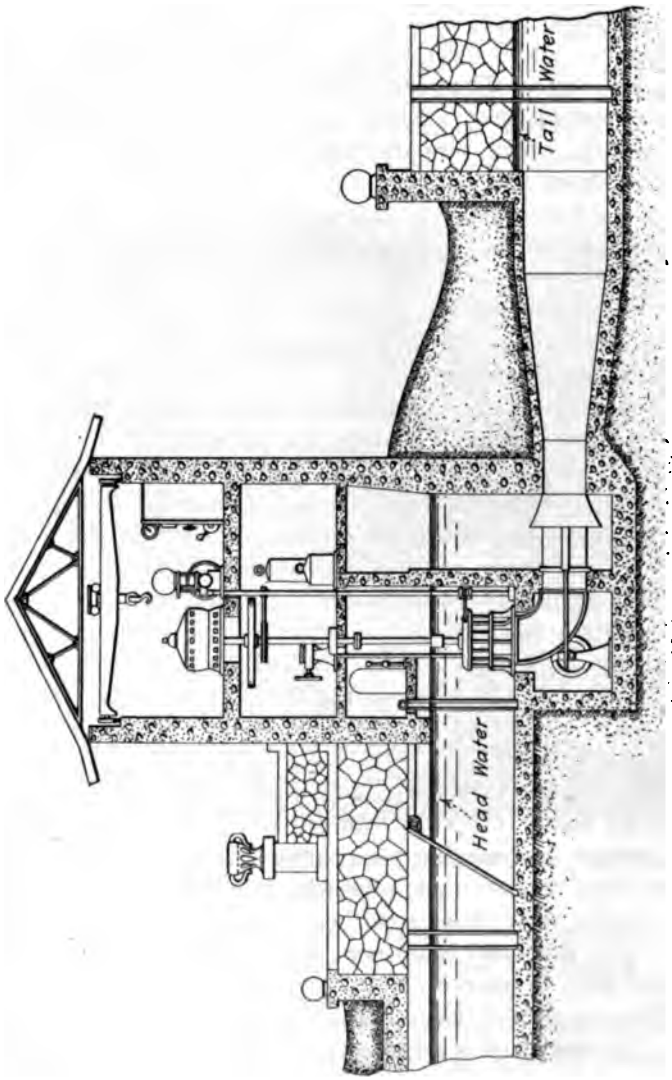


Fig. 1 Cross Section of Power House at the Henry Ford Farms

turbine chambers are scroll-shaped, arranged for their respective right and left hand turbines. The weight ot the runners is sustained by roller bearings on top of the generators. All under water bearings are of bronze.

9 The two turbo-generators are ot the vertical type, 55 kw. each, at 250 volts. They are direct current machines and have ample overload capacity. Each is self-contained, having a roller bearing on top which supports all the rotating parts, including the balance wheels and turbines.

10 A third steam engine generator is provided. This is horizontal, and is of 35-kw. capacity at 250 volts.

11 The governors are vertically driven, and direct connected. They are of the open system, oil-pressure type.

12 Full apparatus is provided in the top floor of the power house for recording test conditions.

13 The three penstocks on the lowest floor can be closed by folding head gates, manipulated by the overhead crane. Of these penstocks, the middle one is a common feeder for the others, each of which contains what has been termed a "turbine discharge accelerator." These accelerators are a form of draft tube, into which each turbine discharges and into which, also, water from the head race is discharged through a feeder. The head race water accentuates the turbine flow through the draft tube, having the same effect as an added head of head water. An illustration of one of the turbines with accelerator is shown in Fig. 2.

DISCHARGE ACCELERATORS

14 The purpose of the discharge accelerators is to transfer energy from surplus water (otherwise running over the dam or wasting) to the turbines, augmenting their power under subnormal heads after they have reached their otherwise full power at those heads. In other words, their effect is to boost the power of the turbines, working under low head conditions, beyond that which the conditions would seem to warrant.

15 The principle of the accelerator may be followed by reference to Fig. 3. In this figure is also shown a common siphon for purposes of comparison.

16 In the case of the siphon, the induced or suction head cannot be greater than the inducer, or pressure head. With the accelerator, the induced head may be greater than the inducer head. The accelerator is not an ejector.

17 The effect of the accelerator on the turbine is to partially nove the pressure at the discharge end without changing the essure at the intake end. Under ordinary conditions, the pressure both the head and tail water is 14.7 lb. per sq. in. alike. If the rbine were to discharge into tail water the pressure on which was, y, only 10 lb. per sq. in., the result would be the equivalent of ding slightly over 10 ft. head to the turbine, and as far as practical ults were concerned, the turbine would now develop energy cor-

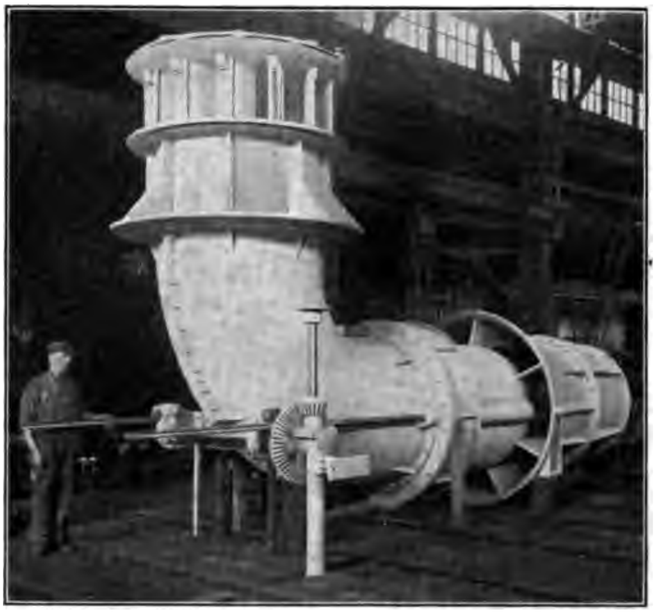


FIG. 2 85-H.P. TURBINE WITH DISCHARGE ACCELERATOR

ponding to the increased head. This is the effect of the discharge celerator, which may therefore be likened in principle to a connser of a steam engine.

18 Each accelerator consists of a super draft tube, a conntrator, and an infuser as shown in Fig. 4. The infuser is proled with a gate, by means of which the amount of accelerator water ay be varied at will, or shut off entirely, in the latter case leaving ly the turbine discharging into the super draft tube.

19 The area of cross section of the concentrator is less than at of the turbine draft tube. The effect of this narrowed section

on the turbine discharge during its ordinary operation was in doubt during the construction period, but when the installation was completed tests showed that the effect was negligible.

ACCELERATOR TESTS

20 A series of tests was made on the accelerator of one unit on March 18, 1915, and the results are plotted in Fig. 5. In this figure, the scale of turbine horsepower is given on the left, kilowatts on the right and turbine gate opening in tenths below. A normal speed of

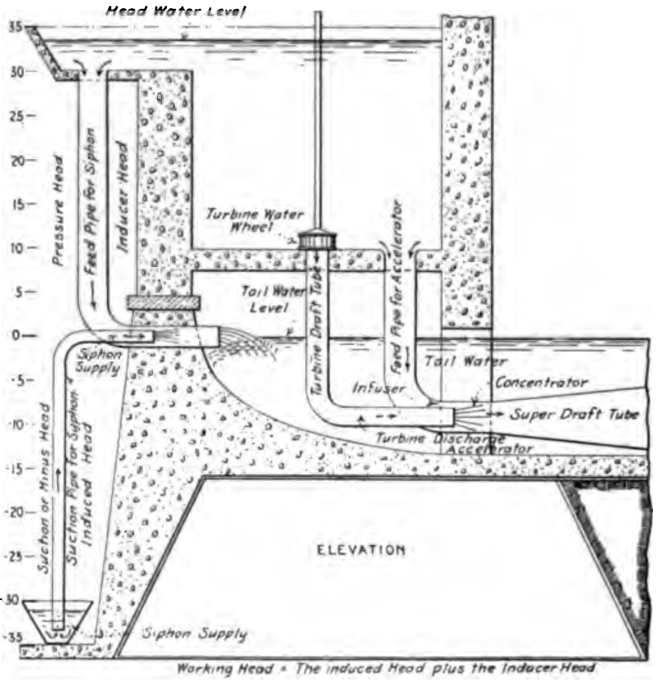


FIG. 3 COMPARISON OF SIPHON AND ACCELERATOR

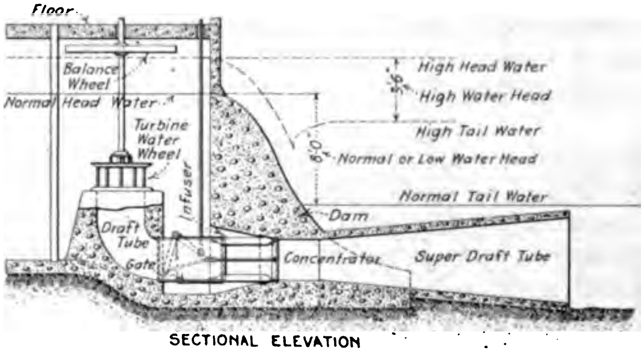
110 r.p.m. was maintained throughout the tests. The energy was absorbed by a water rheostat, and the power was measured by the switchboard instruments, which are new, therefore substantially correct.

21 The purpose of Series 1 was to show the maximum power of the turbine when the accelerator gates were closed. The curve shows approximately 90 turbine h.p. at between 9/10 and full gate, but the head was slightly over 8 ft.

22 Series II shows readings made at the same turbine gate openings, but the accelerator gate was open 20 per cent of its range throughout the series. Although the actual head is lowered, there is an actual increase in power.

23 Series III was made with the accelerator gate open about 40 per cent of its travel. This resulted in a still further increase of power.

24 Series IV was made with the accelerator gate open about 60 per cent of its travel. Throughout this series, the accelerator was sucking some air, but the power increased.



SECTIONAL ELEVATION

FIG. 4 85-H.P. TURBINE AND DISCHARGE ACCELERATOR

25 Series V was made with the accelerator gate open about 80 per cent of its travel. The accelerator sucked air violently during these last readings, but the power increased even though the head was less than 7 ft.

26 The excessive taking in of air prevented the further opening of the accelerator. Therefore its limit of action is unknown.

27 The curve in Series V shows 116 turbine h.p. at 6.94 ft. head. This curve, reduced by the well known formula, to an 8-ft. head shows that 143.5 h.p. can be obtained by the use of the accelerator from an 8-ft. head in connection with the identical turbine that had been guaranteed to develop 85 h.p. under 8-ft. head.

28 No attempt was made to measure the accelerator water used, as this test was made solely to demonstrate that a turbine can use waste water that would be detrimental to the power effects of the turbine if the accelerator were not used. The matter of measuring the water is a part of proposed further development, which will be continued until all the limitations are discovered.

29 The turbine discharge accelerator is the outcome of carefu observation of draft tube phenomena for a term of years. During this observation, some of the apparently unexplainable charac teristics of draft tubes have been fathomed.

30 Although other engineers have been working along the lines of this development, it is believed that the present instance is th

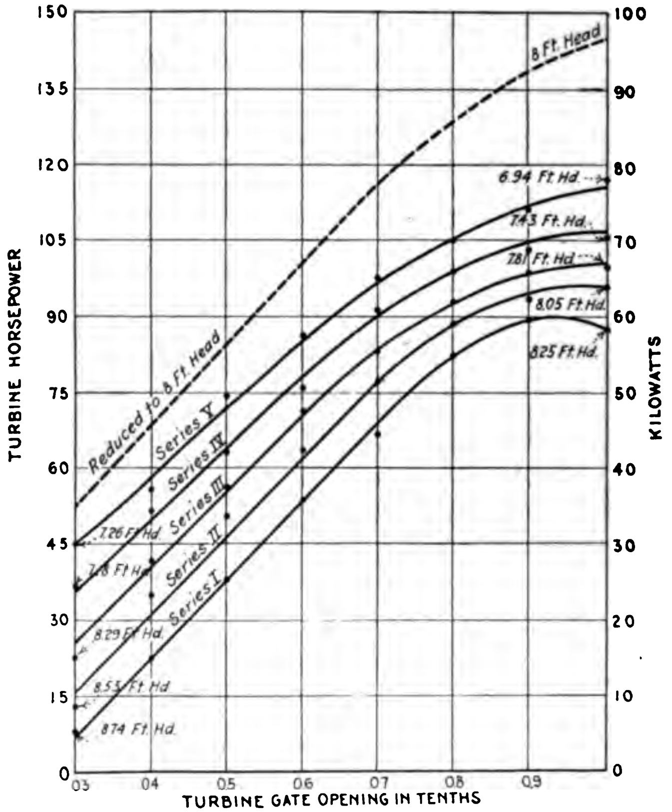


FIG. 5 RESULTS OF TESTS ON ACCELERATOR

first practical demonstration of the decided advantage to be sec by the application of the turbine discharge accelerator, and th sign of the device is believed to be original.

31 The possibilities and limitations of the accelerator hav yet been fully determined, although several years of experimen have been spent in its development. In its operation, there to be some new and definite law, which has not yet been form

32 So far, indications are that low heads may be boosted as much as from 100 to 500 per cent, depending on the conditions. The extreme limit appears to be the atmospheric head.

33 There seems to be every reason to believe that full turbine power from a water turbine can be obtained from all the water used (when based on the actual or working head) even though only a portion of the water passes through the turbine proper. A water power equipped with an accelerator can be speeded for full head at low water condition, and the same turbine can have its capacity practically doubled under the same head, if sufficient water is available. The power unit can develop its normal rated power at one-half head when sufficient water is available, and the turbine can furnish considerable power when the working head is less than 25 per cent of the normal head, and all this may be done at good efficiency.

34 A like reflection will show that with this system many water powers may become independent of steam or gas reserves, with the saving of the large expense these auxiliary power units now entail. With the accelerator, too, it is not necessary to provide so many power units in a given development, as the elasticity permits of good efficiency with fewer turbines.

35 There are very many cases in which low heads only are available and the water cannot be utilized practically and profitably. The accelerator can be used to boost the head in such cases to an amount at which development of the water power is possible. The accelerator can convert an actual head of 3 to 10 ft. into a working or effective head of 8 to 20 ft.

DISCUSSION

Clemens Herschel[1] (written). The idea of utilizing that portion of the freshet river flow that wastes over the dam, for the purpose of increasing the head acting on the turbines, at those times suffering from a diminution of the normal fall (or from "back-water"), is not new. Experiments on an apparatus of this kind were made by M. Saugey, superintendent of the Chevres power plant, owned by the city of Geneva, Switzerland, as early as June, 1905, and perhaps earlier; and Saugey's system was described in the Zeitschrift d. V. D. Ingenieure, about 1907, and in other

[1] 2 Wall Street, New York, N. Y.

journals. A pamphlet, without date, issued by the Société Hydro-Motrice, of Geneva, describes these and later experiments.

The efficiency of the Saugey apparatus, in terms of water lifted a certain height by means of other water falling a certain height, was about 3 per cent. This is pretty poor, even for an apparatus that operates by induced currents of fluids, but it gave the writer the impetus to accomplish something better, and led to his development of a hydraulic apparatus[1] for the purpose named above, which was called the "fall increaser," with a maximum efficiency found, as described above, of 30.4 per cent. To turbine builders and others, this does not sound like a very high efficiency, but it is believed to be very good for an induction current hydraulic apparatus.

It is quite fitting that the results of the experiments made with the fall increaser at the Holyoke Public Testing Flume in 1907 and 1908 should be recorded at this time. A brief mention of the fall increaser, with an up-to-date design of a power house fitted with fall increasers, may be found in Trans. A. S. C. E., November, 1915, in a discussion of a paper on Induced Currents of Fluids, by F. zur Nedden, Mem. Am. Soc. M. E.

In the field of the apparatus the paper describes, other things, principally cost, being equal, the most mechanically efficient apparatus alone will survive. From this viewpoint, it is to be regretted that there is no word or figure in the paper to permit a guess at the mechanical efficiency of the "discharge accelerator." Having conducted experiments with both the fall increaser, and also, to some extent at least, with a "discharge accelerator," the writer has not much faith in the efficiency of the latter. In the article mentioned above, he wrote: "The fall increaser is not an ejector, and experiments made with an ejector form of throat piece, and a 5½-in. nozzle endeavoring to operate it, gave so poor results (efficiency) that there was no encouragement to continue along those lines." As the author states in Par. 16 that "the accelerator is not an ejector," it will be necessary to add that what has been called above "an ejector form of throat piece," was similar to the discharge accelerator now shown. The difference consists in this: The turbine discharge entered the throat piece or mixing chamber through the annular area around the nozzle, instead of the operating water entering through the annular area, as in the discharge accelerator, with the turbine discharge blowing in through the nozzle. There cannot,

[1]See Harvard Engineering Journal, June, 1908, for description.

in the opinion of the writer, be any material difference in efficiency between these two arrangements. The discharge accelerator arrangement is, however, the better one for regulating or varying the discharge of the operating water.

Another quotation from the 1908 article is: "Nor does it seem to me that the forms of formulae found in the books and learned transactions for computing the work of ejectors, based on the assumption of an impact of the nozzle stream upon the water within the throat piece, are based upon a proper assumption to produce a correct formula for representing ejector action. To my mind an ejector is only another form of negative pressure apparatus, in which suction causes the water to enter the throat piece through a ring-shaped orifice situated all around the nozzle (in the accelerator through the nozzle), rather than through holes fashioned in the throat piece itself, and distributed over its whole outside surface, as in the fall increaser."

It might be thought that inasmuch as all these low fall turbine aids use freshet runs of the river to furnish the operating water, their efficiency is of no consequence, but as will be shown in detail, this is not so, except at exceptional times (at the Henry Ford Farms the times called e, which according to Par. 3, last only, as one may judge, certain set periods of hours during the year). From an experience of seven years in designing these plants for river situations of all kinds, the writer can state positively that the mechanical efficiency of the apparatus is of great importance, and so is the regime or character of the river on which the fall increaser is to be used.

It is all a question of kw-hr. produced by the fall increaser in the course of an average year's run of the river, set off against the construction cost; and fall increasers may or may not be of economic value, according to their efficiency. They have this in their favor, that their product is of annual recurrence, forever, while their construction cost (operation and maintenance are negligible quantities) is incurred but once. As a numerical example and to fix ideas, in a case examined some years ago, when the fall increaser was as yet new, the annual product of the fall increasers would have been 158 million kw. in an average year, delivered at times of high water and low fall, lasting in all some 180 days of the year: and the writer's estimated construction cost was one million dollars. From this case—a very favorable one—the net advantages range towards nothing, until in other cases, with rivers of a different regime, those advantages wholly disappear.

Fig. 6 shows the test apparatus of the fall increaser, as used in 1907 and 1908. *C* is the cast iron throat piece, the "soul" of the whole apparatus; situated as it is, suction, or negative pressure, is produced within it when water from the "operating water" penstock *B* flows through it. In the experiments, *B* was a 16-in. pipe, whose discharge was continuously measured by a venturi meter. The throat piece *C*, serving also as a mixing chamber for the operating water with the water that represented the turbine discharge, of "water lifted," (which was discharged into the "vacuum box"

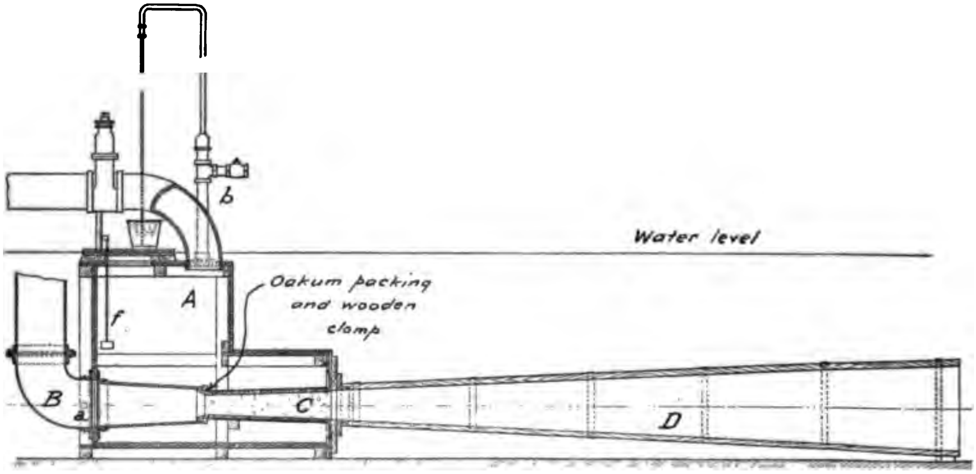


Fig. 6 Test Apparatus of Fall Increaser

A, through a 12-in. pipe), had for the best summit cone angle tested, 3½ degrees. "Water lifted" was continuously metered by a venturi meter. In the experiments, *C* was of various exact diameters at the upstream and downstream ends, approximating to 12 in. at the downstream end in all of them, and the two diameters varying as the throat piece was bored out to increase the diameter at one end or the other, so as to change the summit cone angle.

A was made of wood, was well braced inside, and of the peculiar L-shape shown, to fit into the space available in the testing flume. In practice, the vacuum chamber would be a concrete vault, with manhole entrance, and open for inspection when the turbine is shut off and the operating water is on, The pipe b, tapped into the vacuum box, and sucking colored water out of the pail shown, served to measure the vacuum produced. Working with the turbine penstock shut off, the vacuum produced would easily hold suspended a water column 26 ft. high.

f is a float that indicated the water level within the vacuum box. To see b indicate a vacuum, while f showed the box *full* of water, was startling at first, but one soon learned to consider a body of contained water as readily subject to negative, as to positive pressure.

D is the draft tube proper, whose summit cone angle was 5 degrees. The working of the whole apparatus is not very sensitive as regards the length of this draft tube. A practical rule would be to make the discharge velocity not much over 8 ft. per sec.

The holes in the throat piece in actual use are recommended to be made either bevelling so as to facilitate flow downstream, or with rounded edges, of a uniform diameter, say 6 in., and sufficient in number to cause the velocity through them to be less than 4 ft. per sec. This obviates the need of any special outlet construction for the turbine discharge at times when the operating water is shut off, that is, on days ranging from normal to extreme low water in the river, which usually comprise about 180 days in the year.

Although a little turbine was mounted in the line of the 12-in. cast iron pipe, it was not used during the experiments, being clamped "still," and was only used as a supply orifice. We all know the effect of draft-tube suction on turbines, and this suction having been constantly measured, we know the effect it would have had on the turbine and its power had the turbine been allowed to revolve and generate power.

Dr. Ernst Duebi, at one time of Zürich, Switzerland, repeated the writer's tests in 1911, at Zürich, and added much information to that previously known concerning the fall increaser. His results were published in book form by Rascher & Cie., of Zürich and Leipzig, 1912.

Dr. Duebi saw fit in his experiments designedly to let the turbine revolve during the tests, and thus proved once again that added suction within the draft tube of a turbine, adds fall to the fall otherwise "acting on the wheel." The experiments were conducted with great care; they were made wholly independently of those at Holyoke, and it has been very gratifying to have their added testimony to the efficiency of the fall increaser.

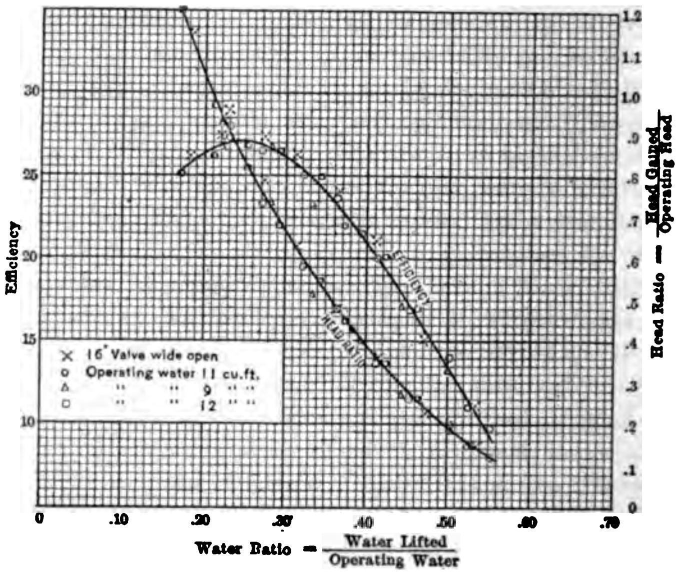


FIG. 7 EXPERIMENTS ON "FALL INCREASER." THROAT PIECE "F," ALL HOLES OPEN

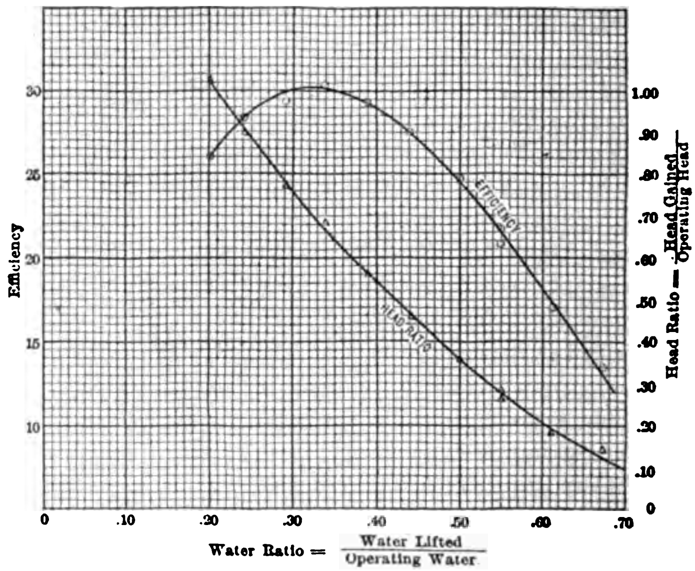


FIG. 8 EXPERIMENTS ON "FALL INCREASER" THROAT PIECE "B," ALL HOLES OPEN

ꞁBLE 1 EXPERIMENTS ON "FALL-INCREASER" THROAT PIECE "F," ALL HOLES OPEN.

February 24, 1908

Number of experiment	Time	Height of water in Forebay	Height of water in 16 in. pipe next the vacuum-box	Height of water in Tail-race	Operating Head Col. 4 — Col. 5	Operating Water.	Water Lifted	Head gained by use of Fall-Increaser	Ht. of Water in vacuum-box by float	Air-valve	Gauge on 12 in. valve	Water-ratio = Col. 8 / Col. 7	Head-ratio = Col. 9 / Col. 6	Quantity (a)	Efficiency = Col. 15 / Col. 6 x Col. 7
		Feet				Cubic Feet per second		Feet							%
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	9.05	99.23	94.89	81.96	12.93	11.39	6.00	1.93	Full	Shut	56.1	0.53	0.15	16.62	11.3
2	9.10	99.25	94.70	81.95	12 75	11.63	5.5)	2.88	—	"	52.8	.47	.23	22.58	15.2
3	9.13	99.29	94.37	81.95	12.42	12.06	5.00	4.52	—	"	47.2	.41	.36	31.64	21.1
4	9.16	99.35	94.11	81.95	12.16	12.33	4.50	5.87	—	"	43.0	.36	.48	36.20	24.0
5	9.20	99.41	93.82	81.95	11.87	12.84	4.00	7.50	—	"	39.0	.31	.63	40.25	26.4
6	9.23	99.55	93.53	81.94	11.59	13.20	3.50	9.10	—	"	35.0	.27	.79	42.10	27.5
7	9.27	99.67	93.26	81.94	11.32	13.60	3.00	10.83	—	"	31.5	.22	.96	42.22	27.4
8	9.30	99.73	92.97	81.93	11.04	13.97	2.50	12.58	—	"	28.2	.18	1.14	40.20	26.1
9	9.34	99.72	88.96	81.81	7.15	11.01	2.50	6.58	—	"	29.1	.23	.92	21.42	27.2
10	9.37	99.64	89.73	81.81	7.92	11.01	3.00	5.80	—	"	33.1	.27	.73	23.11	26.5
11	9.41	99.62	90.43	81.85	8.58	11.00	3.50	5.00	—	"	36.7	.32	.58	23.62	25 0
12	9.44	99.60	91.36	81.86	9.50	11.05	4.00	4.53	—	"	40.4	.36	.48	24.91	23.7
13	9.48	99.59	91.74	81.88	9.86	10.95	4.50	3.45	—	"	44.3	.41	.36	21.72	20.1
14	9.53	99.57	92.81	81.91	10.91	11.00	5.00	2.86	—	"	48 9	.45	.26	20.33	16.9
15	9.56	99.52	93.55	81.93	11.62	11.03	5.50	2.30	—	"	52.6	.50	.20	18.13	14.1
16	9.59	99 47	94.19	81.94	12.25	10.98	6.00	1.50	—	"	57.1	.55	.12	13.16	9.8
17	10.05	99.50	89.31	81.80	7 51	9 01	4.50	1.40	—	"	46.0	.50	.19	9.02	13.3
18	10.08	99.49	88.68	81.79	6 89	8.99	4.00	1.85	—	"	42.3	.44	.27	10.57	17.1
19	10.11	99 47	88.08	81 77	6.31	8.97	3.50	2.52	—	"	38.2	.39	.40	12.24	21.6
20	10.14	99.48	87.65	81.75	5.90	9.00	3.00	3.02	—	"	35.0	.33	.51	12.38	23.3
21	10.18	99.50	86.85	81.72	5.13	8.99	2.50	3.72	—	"	30.3	.28	.73	12.36	26.8
22	10.22	99.52	86.26	81.71	4.55	8.98	2.00	4.22	—	"	26.6	.22	.93	10.99	26.8
23	10.27	99.52	89.33	81.83	7.50	12.05	2.00	8.97	—	"	25.5	.17	1.20	22.82	25.2
24	10.30	99.54	90.25	81.85	8 40	12.00	2.50	8.17	—	"	29.3	.21	.97	26.46	26.3
25	10.33	99.56	90 94	81.86	9.08	11.99	3.00	7.43	—	"	32 4	.25	.82	29.34	26.9
26	10.36	99.57	91.80	81.88	9.92	11.99	3.50	6.75	—	"	36.0	.29	.68	31.53	26.5
27	10.40	99.60	92.55	81.90	10.65	11.91	4.00	5.83	—	"	39.6	.34	.55	31 71	25.0
28	10.44	99.59	93.85	81.94	11.91	12.11	4.50	5.30	—	"	43.8	.37	.45	32.89	22.1
29	10.48	99.58	94.32	81.95	12.37	11.98	5.00	4.28	—	"	47.5	.42	.35	29 89	20.2
30	10.51	99.57	94.83	81.96	12.87	11.92	5.50	3.37	—	"	51.4	.46	.26	26.26	17.1
31	10.55	99.57	95.22	81.96	13.26	11 48	6.02	1.93	—	"	56.8	.52	.15	16.94	11.1
32	10.59	99.56	95.27	81.95	13.32	11.32	6.00	2.30	79.65	open	59.4				
33	11.03	99.57	95.08	81.94	13.14	11.61	5.50	3.36	78.58	"	55.6				
34	11.06	99.58	94.92	81.93	12.99	11.81	5.00	4.13	77.80	"	52.0				
35	11.09	99.60	94.82	81.91	12.91	11.89	4.50	4.33	77.58	"	48.1				
36	11.13	99.62	94.86	81.89	12.98	11.92	4.00	4.38	77.50	"	43.9				

TABLE 2 THROAT PIECE "B," ALL HOLES OPEN.

February 18, 1908

1	9.07	99.66	95.88	81.92	13.96	10.83	6.00	3.70	Full	Shut	Not Taken	.55	.27	31.57	20.9
2	9.10	99.65	96.10	81.93	14.17	10.58	6.50	2.73	"	"		.61	.19	25.59	17.1
3	9.14	99.67	96.29	81.94	14.35	10.40	6.99	2.00	"	"		.67	.14	20.20	13.5
4	9.18	99.68	95.88	81.93	13.95	10.84	6.00	3.88	"	"		.55	.28	33.00	21.8
5	9.22	99.68	95.67	81.92	13.75	11.08	5.49	4.95	"	"		.50	.36	37.84	24.8
6	9.26	99 67	95 37	81.92	13.45	11.38	5.00	6.13	"	"		.44	.46	42 06	27.5
7	9.30	99.68	95.13	81.91	13.22	11.64	4.50	7.37	"	"		.39	.56	45.08	29.3
8	9.33	99.67	94.86	81.89	12.97	11.91	4.00	8.78	"	"		.34	.68	46.92	30 4
9	9.36	99.66	94.66	81.89	12.77	12.12	3.50	9.85	"	"		.29	.77	45.58	29.4
10	9.39	99.64	94.39	81.88	12.51	12.38	3.00	11.25	"	"		.24	.90	44.00	28.4
11	9.42	99.61	94.16	81 87	12.29	12.60	2.50	12.52	"	"		.20	1.02	40.37	26.1

The essential results found in the experiments at Holyoke are shown in Figs. 7 and 8, and are tabulated in Tables 1 and 2.

Table 1 gives the results of experiments on throat piece F, while Table 2 gives the full-gate operating-water experiments on throat piece B. As will be seen from Table 1, and from its representation in Fig. 7, the efficiency of the fall increaser is the same for equal ratios of (water lifted) ÷ (operating water) and of (head gained) ÷ (operating head), called respectively, the "water ratio" and the "head ratio." This having proved true in all the complete series of experiments, it authorizes the plotting of the diagram for throat piece B, from the results alone of the full-gate operating-water experiments made with that throat piece.

In an apparatus of this sort it does not palpably appear what is the true efficiency of the apparatus.

The fall increaser comprises the vacuum box, no less than the throat piece and its feed, and exhaust; and as such, causes an increased discharge of the turbine to which it is applied, as well as an increase of fall acting on the turbine, as has already been noted.

Let h = operating head, on 16-in. pipe

h' = head gained; or vacuum

Q' = water lifted = discharge of turbine under head $(h+h')$

Q = discharge of turbine under head h.

The work done by the turbine without the increaser is equal to Qh; and the work done by the turbine with the increaser $= Q'(h + h')$.

Also $Q : Q' = \sqrt{h} : \sqrt{h + h'}$, or $Q = \dfrac{Q'\sqrt{h}}{\sqrt{h + h'}}$

Work gained by the use of the increaser

$$= Q'(h + h') - Q\,h.$$

$$= Q'(h + h') - \frac{Q'\,h\,\sqrt{h}}{\sqrt{h + h'}} = (a)$$

And the efficiency of the increaser is this quantity, divided by the work employed to gain it; or

$$\text{Efficiency} = \frac{(a)}{\text{operating water} \times h.}$$

The fact that the efficiency of the fall increaser is the same for equal water-ratios and head-ratios is a valuable one, enabling the normal efficiency to be maintained for all the varying discharges of the turbine.

Fig. 9 shows a design of power house, with fall increaser, made for the City of Geneva, Switzerland, site not yet built upon. That the design had novelty in 1907 may be gathered from the fact that patents were issued for it by the United States, Canada, France, Switzerland, Italy, Sweden and Germany.

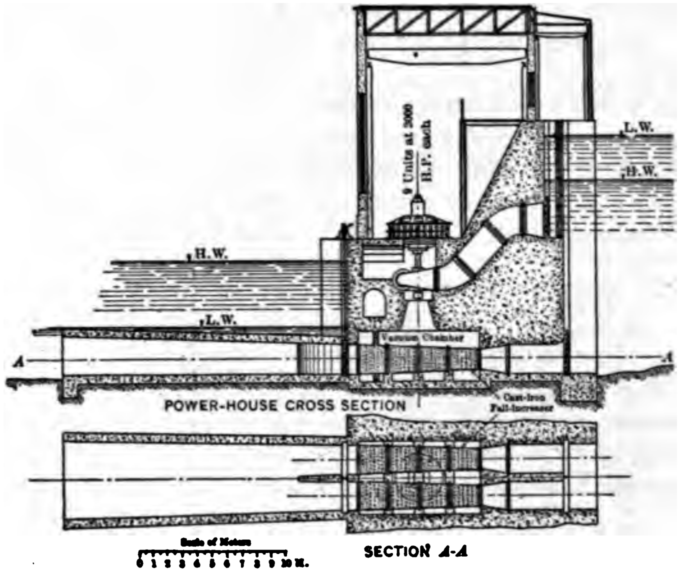


Fig. 9 Power House with Fall Increasers

At the site named, the water upstream from the power house is lowest (must be *held* lowest) in times of freshet in the river, there being lift gates designed to create the mill pond, instead of a dam. At the same time there is "back-water;" or in other words, the fall is reduced doubly (shutting up like an accordion), which accounts for the elevation of H.W., Fig. 9, being at a *lower* elevation than that of L.W.

Note the ready means supplied by fall increasers for getting rid of rack-trash. It needs but to be pushed *down*, into the suction area of the fall increasers, to pass through them and out through the tail-race, instead of being laboriously raked *up* and carried ashore and away.

It is evident that to enable the use of freshet water to operate the fall increasers, a head race of any material length is inadmissible. The power house must be at or very near the dam. This requirement eliminates the majority of hydroelectric plants from a consideration of fitting them with fall increasers.

The next elimination takes place when the character or regime of the river is taken into consideration. Rivers differ in this respect far more than one would suspect; and a careful analysis of their modes of flow during all the days of several years,—the more the better—are required, before their regimes may be adequately portrayed.

This is done by setting up for them what have been called "duration curves." Every engineer knows what hopeless looking messes a plotted series of daily discharge curves (or pictured saw teeth) make, the ordinates representing *consecutive* days of the year, and the abscissae gage heights, or sec-ft. of river flow. But let each year

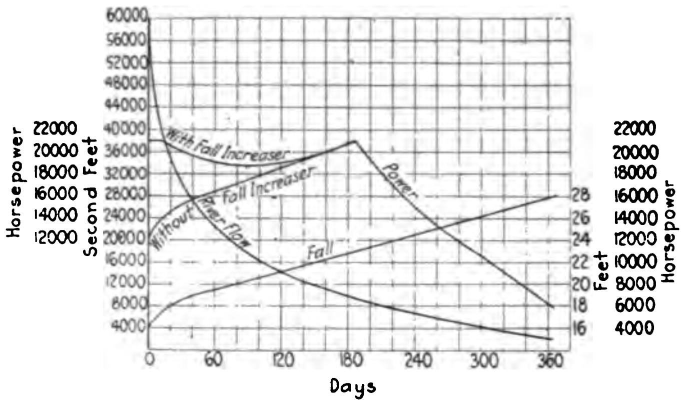


FIG. 10 EFFECT OF FALL INCREASERS

be represented by these same daily quantities, plotted in the order of their size, and we get at once a smooth curve[1] of quantities flowing in the river for the 365 days of the year, useful in more ways than one. Fig. 10 shows such a curve of river flow.

Besides showing a curve of river flow, Fig. 10 indicates the corresponding falls that obtain synchronously at the mill site under consideration: and from these two the power curve at this site, *without* the use of fall increasers, may readily be computed and plotted. From the power curve we may, with the aid of Fig. 8,

[1]See also Trans. A. S. C. E., 1907, A tabular analysis of 20 years' flow of the Connecticut River; and Doc. 1400, H.R., 62nd Congress, 3d Session, 15 years' flow of the Potomac River at Great Falls near Washington.

compute and plot the power curve for the back-water days of the year *with* the use of the fall increaser. The area included between the last two curves named gives the total horsepower-days or kw-hr. which fall increasers would produce annually forever on that mill site. Note that at first there is not water enough in the river to operate *all* the fall increasers, causing a sagging down of the curve and only a gradual bringing of the curve up and back to normal full power. Here is where the efficiency of fall increasers becomes of great moment.

There only remains to compare the productive value of fall increasers with the interest on the construction cost. The result depends mainly on the regime of the river. Fall increasers are useless where the river flow is effectively regulated by great lakes or reservoirs causing a uniform discharge of the river. They are uneconomical on rivers that have only a few days of the year of back-water. On the other hand at times a single year's output of kw-hr. will nearly pay for their construction cost; and heat engines put in to supplement the low water run of the river cannot afford to burn fuel in competition with the cost of a kilowatt-hour when produced by the fall increasers in the days of freshet water during the year.

It would also seem that, tide mills, which may have an inexhaustible water supply, but which all have a greatly varying fall during the 24 hours, could materially benefit by the use of fall increasers; and if the "discharge accelerator" will show the proper cost and efficiencies, it can presumably compete in this and other cases.

A prime mover, or apparatus to increase the power of a prime mover, is palpably without index by which to judge of its value, so long as its *mechanical efficiency* has not been determined. In the present case this would call for a statement of the amount of water the discharge accelerator consumer per horsepower developed.

A series of constructions, all aiming to utilize waste water, and known by such names as "compensators," "ejector flow increasers," etc., have been built and may be found at Warren, Ohio; at Eldora, Iowa: at the U. S. dam between St. Paul and Minneapolis, and on the Huron River near Ann Arbor, Michigan. None of these have been tested for efficiency.

R. L. DAUGHERTY.[1] The writer has been much interested in this paper by Mr. Replogle. It is hard for some people to realize that a water power plant may have to shut down because of a superabundance of water as well as because of lack of water. Such a situation is only met with in the case of a low head plant where the fall available may be almost destroyed in time of flood. The writer has found it necessary to explain a number of times why it is that such a fall decreases in time of high water. This point is illustrated in Fig. 11. This photograph, while of a relatively small stream, shows the effect just as well as one of a much larger stream and fall. In this particular case the depth of water flowing over

FIG. 11 LOW HEAD IN TIME OF HIGH WATER

the crest of the submerged dam was practically equal to the height of the dam above the bed of the canal. It may be seen that the dam does little more than create a disturbance in the flow of the water, and the fall is very slight. In fact a portion of this very small fall is due to the fact that some of the water is diverted at this point.

Under such circumstances the amount of water consumed by a turbine will be much less than the normal amount, and the head being less also the power will be seriously reduced. The device which Mr. Replogle has employed makes it possible to consume more water and thus to compensate for the reduction of head.

It would be of considerable interest if we knew the amount of water actually discharged by the turbines during the tests made by

[1]Asst. Prof. Hydraulics, Sibley College, Cornell University, Ithaca, N. Y.

the author, and also the additional water used to produce this effect. It is to be hoped that the author will be able to secure these data at some future time.

It is well known that it costs more per horsepower to develop a given amount of power under a low head than under a high head. As the author states, many low head powers are not utilized, though there is an abundance of water, due to the high cost. Such a device as this, used constantly, converts a low head plant into one of somewhat higher head. The actual amount of water consumed by the plant including that wasted through the accelerators may not be much more than the amount required to develop the same power under the lower head without the accelerators. It would be interesting if the author could give comparative efficiencies in such a case. Of course for the discharge of flood waters only, the efficiency is of no consequence, but for constant use it would be.

While this accelerator is different in detail, it seems to be similar in principle to the fall increaser described by Clemens Herschel in *Engineering News,* Vol. 73, p. 84.

The Author. In reply to Mr. Daugherty, data regarding the amount of water actually discharged during the tests could not be procured at the time the tests were made. Some data have been secured in preliminary tests of a very small turbine, and these compare favorably with the results secured by Mr. Herschel.

It was preferred to make no reference to efficiency tests until such tests could be made in a logical and comprehensive manner. These will be made in the course of further development.

Of constructions suggested by other engineers the author obtained his first knowledge through the U. S. Patent Office. He believes the construction described is original.

Any construction designed for the purpose of mechanically mixing the two streams of water is erroneous from an efficiency point of view. Mixing implies eddies, and eddy currents transform the kinetic energy in the inducer water into heat. In the preliminary tests the very poorest results out of several hundreds were from a carefully designed mixer.

In reply to Mr. Herschel, there can be no doubt of the real values to be obtained from the use of the atmospheric head with surplus water. The means applied are of no special importance. The doubt in Mr. Herschel's mind is in regard to efficiency, but as he says best overall efficiency is in returns from investment.

The efficiencies quoted by Mr. Herschel seem to be low. From the author's point of view the apparatus he shows has been provided with the best possible means to produce eddy currents and friction. The grating or perforated throat certainly impedes the inducer stream. The turbine water entering at right angles to the inducer stream causes endless eddy currents. The abrupt orifices are causes of much friction. It is possible that if the whole throat section were removed the efficiency would be as high as that stated.

In conclusion, it was thought that the facts gathered to date regarding the accelerator described in the paper might be of interest, but the author has substantial reasons to believe that much higher efficiencies than those given can be obtained from this class of apparatus.

No. 1518

PROPORTIONING CHIMNEYS ON A GAS BASIS

By A. L. Menzin, Philadelphia, Pa.
Associate-Member of the Society

The increasing tendency to operate boilers at higher overloads, the attention being given to baffling as a factor in improving boiler performance, and the efforts to improve the efficiency of combustion, resulting in a reduction of the volume of gases to be removed, seem to require a method for calculating the proportions of a chimney that will take into consideration all the factors involved. The author has reviewed the subject and here attempts to arrange the data in convenient form for practical applications.

I EFFECTIVE DRAFT AND HEIGHT OF CHIMNEY

2 The general equation for drafts in a boiler plant may be stated

$$P_g+P_b+P_v=[(P_s-P_c)-P_r]-P_d$$

where P_g is the draft required to overcome friction of grate and fuel bed, P_b is the draft loss through the boiler, P_v is the draft required to produce velocity up to the boiler damper, P_s is the maximum draft produced by the chimney, P_c is the draft loss in the chimney, P_r is the draft loss in the breeching and P_d is the damper friction. $P_g+P_b+P_v$ is "the draft required at the boiler damper"; P_s-P_c is "the effective draft of the chimney"; and $(P_s-P_c)-P_r$ is "the available draft at the boiler damper." In practice, P_d is varied arbitrarily to reduce the available draft to that required.

3 In any case, in a typical arrangement of a boiler plant, Fig. 1, the draft indicated at the gage d is the draft required at the damper for the conditions then existing. This, of course, must include the draft required to produce velocity at the damper. The gage f will indicate the draft required in the furnace and the difference between

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers.

the readings of the gages d and f will be the draft loss through the boiler. To indicate this in the general equation, $P_g+P_b+P_v$ may be replaced by P_f+P_b, where P_f is the draft in the furnace under any conditions and P_b is, as before, the draft loss through the boiler. Then, in general,

$$P_f+P_b=(P_z-P_c)-P_r-P_d \quad \ldots\ldots\ldots\ldots\ldots [1]$$

In applications to design of chimneys, $P_d=0$, since the damper will be assumed to be wide open. Also, since the maximum draft and the draft loss in the chimney are both proportional to its effective height, H, then P_z-P_c may be replaced by

$$\frac{H}{100}(p_z-p_c)$$

where p_z and p_c represent the maximum draft and draft loss, respectively, per 100 ft. of height. Making the above substitutions in [1]

$$P_f+P_b=\frac{H}{100}(p_z-p_c)-P_r$$

or

$$P_f+P_b+P_r=\frac{H}{100}(p_z-p_c)$$

The left hand side of this equation represents the draft required at the entrance to the chimney and may be replaced by the total effective draft, P_e. Then

$$P_e=\frac{H}{100}(p_z-p_c)$$

or, in final form for chimney calculations

$$H=\frac{100\,P_e}{(p_z-p_c)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where, to summarize, H is the effective height of the chimney in feet, P_e is the total effective draft required at the entrance to the chimney, p_z is the maximum draft produced by the chimney per 100 ft. of height, and p_c is the draft loss in the chimney per 100 ft.

4 It is evident that the effective height of the chimney is not its actual height but the height above the boiler damper. Vertical components of the breeching may be considered a part of the chimney.

II DETERMINING THE MAXIMUM DRAFT

5 If the boiler damper is suddenly closed, a gage as at z, Fig. 1, will indicate the maximum draft produced by a chimney under temperature conditions then existing, provided the effects of wind and leakage are negligible. The formula for maximum draft, as given in Kent's handbook in the section devoted to chimneys, but with different symbols, is

$$P_z = H\left(\frac{7.64}{T_a} - \frac{7.95}{T_c}\right)$$

where P_z is the maximum draft in in. of water, H is the height of the chimney in ft., T_a is the temperature of the external air in deg. fahr. absolute and T_c is the mean temperature of the gases in the chimney on the same temperature scale.

6 Since this formula is for the special case where the atmospheric pressure is about 30 in. of mercury, we may make it general by introducing the factor $P/30$, where P is any pressure in in. of mercury as read on the barometer. The formula then becomes

$$P_z = \frac{PH}{30}\left(\frac{7.64}{T_a} - \frac{7.95}{T_c}\right)$$

or, in final form,

$$P_z = 0.255\,PH\left(\frac{1}{t_a + 459.6} - \frac{1.04}{t_c + 459.6}\right) \quad \ldots\ldots\ldots\ldots [3]$$

where the respective temperatures of air and gases, t_a and t_c, are in deg. fahr.

7 Fig. 3 indicates graphically values of P_z per 100 ft. of height and for a pressure of 30 in. of mercury. It is evident that for any other pressure, P, the factor of correction is $P/30$. Fig. 4 gives values of this factor for the normal barometric pressures corresponding to different altitudes.

III DRAFT REQUIRED TO PRODUCE CHANGE OF VELOCITY

8 The general formula for the draft required to produce change of velocity at the same gravitational level is

$$P_v = \gamma \frac{v_2^2 - v_1^2}{2g}$$

where P_v is the net pressure or draft required, γ is the density of the fluid, v_1 is the initial velocity and v_2 is the final velocity.

9 Rankine gave a value for the density of flue gases at sea level in lb. per cu. ft. as equal to 0.084 $(T_o \div T_c)$, where T_o is the absolute temperature corresponding to 32 deg. fahr. and T_c is the absolute temperature of the gases. Hence, introducing the barometric pressure, P, in in. of mercury

$$\gamma = \frac{P\,(32+459.6)}{30\,(t_c+459.6)}0.084$$

or

$$\gamma = \frac{41.3\,P}{30\,(t_c+459.6)} \quad \ldots\ldots\ldots\ldots [4]$$

where γ is the weight in lb., of 1 cu. ft. of gases and t_c is the temperature of the gases in deg. fahr.

10 The general formula for the draft required to produce change of velocity may then be written, introducing the factor 0.192 for converting lb. per sq. ft. to in. of water,

$$P_v = 0.192\,\frac{P}{30}\left(\frac{41.3}{t_c+459.6}\right)\left(\frac{v_2^2-v_1^2}{2\,g}\right)$$

or finally

$$P_v = 0.123\,\frac{P\,(v_2^2-v_1^2)}{30\,(t_c+459.6)} \quad \ldots\ldots\ldots\ldots [5]$$

where P_v is the draft in in. of water required to produce the change in velocity, P is the barometric pressure in in. of mercury, v_1 is the initial and v_2 the final velocity of the gases in ft. per sec. and t_c is, as before, the temperature of the gases in deg. fahr.

IV LOSS OF DRAFT DUE TO SUDDEN ENLARGEMENT OF GAS PASSAGE

11 As stated in textbooks on hydrodynamics, eddies are set up when the velocity is decreased as a result of a sudden enlargement of the area. This is accompanied by a loss of pressure. The formula for the loss of head is

$$h = \frac{(v_1-v_2)^2}{2\,g}$$

To change this to loss of draft it must be multiplied by the factor for converting pounds per sq. ft. into inches of water and by the density of the fluid. Using the values of these given in the preceding section

$$P_v' = 0.192\frac{P}{30}\left(\frac{41.3}{t_c+459.6}\right)\frac{(v_1-v_2)^2}{2\,g}$$

$$=0.123\,\frac{P\,(v_1-v_2)^2}{30\,(t_c+459.6)}\,\ldots\ldots\ldots\ldots\ldots\ldots\ldots\,[6]$$

The symbols have the same significance as in the preceding section.

V FRICTION LOSS IN BREECHING AND CHIMNEY

12 The general formula used to express the resistance to the flow of fluids in pipes is of the form

$$P_r' = \frac{f\,\gamma\,l\,v_2}{m}\,\ldots\ldots\ldots\ldots\ldots\ldots\ldots\,[7]$$

where P_r' is the resistance in units of pressure, f is the coefficient of friction, γ is the density of the fluid, l is the length of the conduit, v is the velocity of flow and m is the quotient of the area of the conduit divided by its perimeter. For gases at a constant pressure, the density varies inversely as the absolute temperature; hence γ may be replaced by $k \div (t_c+459.6)$, where k is a constant. For circular and square conduits the quotient of the area divided by the perimeter is $d/4$, where d is the diameter of a circular conduit or the length of one side of a square conduit. Substituting the above equivalents and introducing the factor 0.192 for obtaining the friction in inches of water,

$$P_r = 0.192 \times 4\,k\,f\,\frac{l}{d}\,v^2\left(\frac{1}{t_c+459.6}\right)$$

$0.192 \times 4\,k\,f$ may be replaced by another coefficient of friction F. Then, in final form for calculating friction,

$$P_r = F\,\frac{l}{d}\left(\frac{v^2}{t_c+459.6}\right)\,\ldots\ldots\ldots\ldots\ldots\ldots\,[8]$$

where P_r is the total friction in in. of water, F is the coefficient of friction, l is the length of the gas passage in ft, v is the velocity in ft. per sec., d is the diameter of a circular conduit or the length of one side of a square conduit in ft., and t_c is the temperature of the gases in deg. fahr.

13 No careful experiments seem to have been made to determine the magnitude or variation of the coefficient of friction for chimney gases. Peclet[1] gave an estimated value for "sooted surface."

[1]Trans. Am. Soc. M. E., vol. 12, p. 97.

Professor Gale[1] determined some values roughly by a few experiments. The values for the friction losses of air in mêtal, concrete and brick conduits used by designers of ventilating apparatus should not be far off.

14 Reduced to the basis of formula [8] these are:

$F = 0.006$ by Peclet for "sooted surface"

$= 0.012$ by Professor Gale for brick conduits

$= 0.006$ from a coefficient for air in brick or concrete conduits.

Professor Gale's value seems too large. Peclet's value is presumably for sheet metal. As a safe value, which seems to give losses consistent with those observed in practice, the author has assumed $F = 0.008$ for the purpose of constructing the curves of Fig. 5. This value is one-third greater than that according to Peclet.

15 The factors of correction for the graphical values in Fig. 5 were determined as follows: Let f, l and m in formula [7] remain the same, then

$$P_r \propto \gamma v^2$$

16 The factor of correction for square conduits of side equal to diameter of circular conduits when γ is constant is v_s^2/v_o^2 where v_s is the velocity in the square conduit and v_o is the velocity in the circular one. But for the same volume of gas per second

$$\frac{v_s^2}{v_o^2} = \frac{A_o^2}{A_s^2} = \left(\frac{0.7854\,d^2}{d^2}\right)^2 = 0.62$$

Hence the friction in a conduit of square section is 62/100 of the loss in an "inscribed" circular section.

17 Since for the same weight of gas per second the velocity varies inversely as the density, and since $P_r \propto \gamma v^2$, then $P_r \propto 1/\gamma$. The density varies inversely as the absolute temperature and directly as the pressure, hence

$$P_r \propto \frac{T}{P}$$

18 The factor of correction for any absolute temperature T, the weight of gas per second remaining the same, is then T/T_o, where T_o is the assumed standard temperature; also, the factor of correction for any absolute pressure P is P_o/P, where P_o is the assumed standard pressure. The factor of correction applicable to Fig. 5 for any temperature, t_c in deg. fahr., is therefore $(t_c + 459.6) \div 1000$; and the factor for any pressure, P, in in. of mercury is $30/P$.

[1]Trans. Am. Soc. M. E., vol. 11, p. 456.

19 In the absence of evidence on the comparative cooling of gases in brick, or brick-lined, conduits and sheet metal conduits, it would seem inadvisable to make any allowance for the decreased friction in the latter, because the lowering of temperature due to cooling reduces the maximum draft and thus compensates to some extent the decreased friction loss. It is probable that the cooling in well constructed brick chimneys is very slight provided the air leakage is small. By calculation, the cooling in unlined metal chimneys may be, roughly, in the neighborhood of 80 deg. per 100 ft. of height.

VI DRAFT REQUIRED AT THE BOILER DAMPER

20 This involves so many arbitrary factors that it cannot be calculated by a general formula and must be assumed from a knowledge of the equipment and conditions. The draft required at the boiler damper is the sum of the draft assumed to be required in the furnace plus the assumed draft loss through the boiler. It should be possible to construct curves for each of these requirements that would be satisfactory for chimney calculations. Fig. 2 is typical for boilers.

21 The lower curve is based on observed draft losses during tests with coal fuel when the CO_2 in the gases was relatively high, on correction of these losses to a standard of 13 per cent CO_2 and on the assumption that the draft loss varies as the square of the percentage of rating developed. This curve may be considered as applicable to boilers 15 tubes high without superheaters. The points circumscribed by circles are for a boiler 14 tubes high, but with a superheater over the first and second passes; the circumscribed crosses are for a boiler 13 tubes high also with superheater.

22 C. R. Weymouth[1], gives draft losses for three-pass Babcock and Wilcox boilers fired with oil fuel under test conditions which are not very different from the above mentioned curve for Edge Moor boilers fired with coal.

23 The upper curve in Fig. 2 was constructed from the lower by allowing for increased air, but at a decreasing rate for increasing overload. While it is not unusual for a boiler to develop its rated capacity with 8 per cent CO_2 in the escaping gases, it is hardly to be expected that a boiler should develop 175 per cent of rating with less than 11 per cent CO_2. Hence the variation in the percentages of CO_2 assumed.

[1]Trans. Am. Soc. M. E., vol. 34, p. 645.

24 The upper curve may be considered as representative of ordinary firing under unfavorable conditions, while the lower curve is for good firing under favorable conditions. Since the chimney is one of those factors which should be designed large enough "to keep the plant running" under conditions that are likely to arise at any time, the upper curve should ordinarily be used for chimney calculations. It is apparent, from the difference between the two curves, that one way to increase the capacity of a chimney is to improve the efficiency of firing.

25 Different types of boilers probably have different losses for the same overload, and the same type of boiler but differently baffled also has different losses. Hence curves such as those in Fig. 2 should be used only for the equipment for which they were constructed, or for very similar equipment. This is as true for grates and stokers as it is for boilers.

VII CONVERTING BOILER HORSEPOWER INTO GAS VOLUME

26 Before the principles discussed can be used for proportioning chimneys, it is necessary to have at hand means for quickly reducing boiler horsepower to a gas basis.

27 Since the proportions of oxidizable matter in a pound of fuel vary between very wide limits, the minimum amount of air required for complete combustion varies accordingly. The air per pound of "combustible" is more uniform but still very variable, but the air per B.t.u. is almost the same if calculations for a limited number of samples of different kinds of fuels are representative of the general case. For several samples of lignite, bituminous, semi-bituminous and anthracite coal, and California crude oil, the calculated minimum moisture-free air per 10,000 B.t.u. by the bomb calorimeter did not vary more than 1½ per cent from 7.50 lb. This suggests using the B.t.u. basis for standard purposes.

28 With the accuracy of Dulong's formula in mind, when applied to bituminous, semi-bituminous and anthracite coal, the true combustibles in these fuels are practically total carbon, available hydrogen (H—O/8) and sulphur. Sulphur occurs in small quantities, requires little air for combustion and has a low heat value; it may therefore be neglected in approximate calculations. Hence coal may be considered as composed of carbon and available hydrogen.

29 Bul. 29, 1911, Bureau of Mines, on The Effect of Oxygen in Coal, gives the carbon and available hydrogen in over 300 samples of American coal. From a study of these it seems that an average

ratio may be taken as 5 parts available hydrogen to 100 parts carbon. Using this ratio, Dulong's formula for calorific value and a specific heat of gas equal to 0.237, the data given in Table 1 follow by direct calculation. The chimney losses given are not involved in calculations for chimney capacity, but were included because they are easily computed from the weights of gases and may be of interest as showing the effect of excess air on boiler efficiency.

30 It would seem that the weight of steam in gases may be neglected in approximate calculations, since the error in weight of gases and chimney losses will be well within 5 per cent, except in

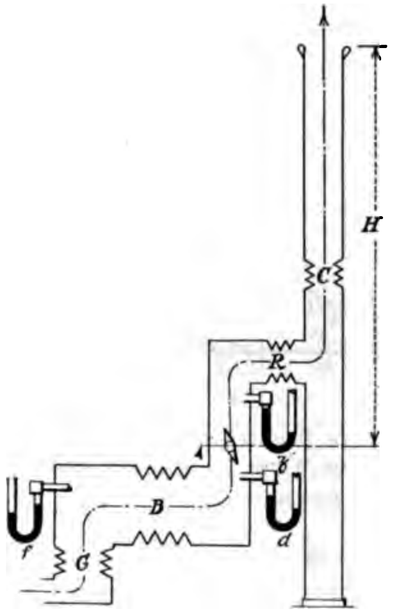


Fig. 1 Typical Arrangement of a Boiler Plant Showing Draft Losses

G Loss Through Grate and Fuel Bed, *B* Through the Boiler, *R* Through the Breeching and *C* Through the Chimney

unusual cases. It is probable that extreme variations from the assumed ratio of available hydrogen to carbon will affect the results by an error not more than 10 per cent, which is allowable where other factors entering into the problem may be in error as much as this.

VIII PROCEDURE FOR CALCULATING CHIMNEYS

31 The formulae and data required for calculating chimneys on a gas basis now having been reviewed or developed, it remains to

apply them to practical problems. Certain curves and tables which logically belong in the sections devoted to the discussion of the principles have been included instead in the following sections to facilitate calculations, and for the same reason final formulae have been repeated. The following symbols are used in this application:

A = area of a circular conduit, sq. ft.
d = diameter in in. corresponding to A
H = effective height of chimney, ft.
l = equivalent length of breeching, ft.

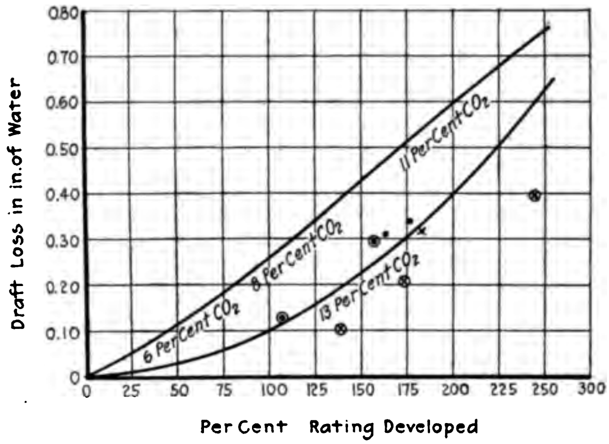


FIG. 2 DRAFT LOSS FROM FURNACE TO DAMPER OF A CROSS-BAFFLED THREE-PASS EDGE MOOR WATER TUBE BOILER WITH 18-FT. TUBES

For Efficient and Inefficient Firing

P = barometric pressure, in. of mercury
P_b = draft loss through the boiler, in. of water
P_e = total effective draft required at the entrance to the chimney, in. of water
P_f = draft required in the furnace, in. of water.
P_r = total draft loss in the breeching due to friction, in. of water
P_v = draft required to increase velocity, in. of water
P_v' = draft loss caused by sudden enlargement of the gas passage, in. of water
p_c = draft loss in the chimney per 100 ft., in. of water
p_z = maximum draft produced by the chimney per 100 ft., in. of water

q = volume of gases, cu. ft. per sec. per boiler h.p.
Q = total volume of gases, cu. ft. per sec.
t_a = temperature of the air, deg. fahr.
t_c = temperature of the flue gases, deg. fahr.
v = velocity of gases, ft. per sec.
γ = density of gases, lb. per cu. ft.

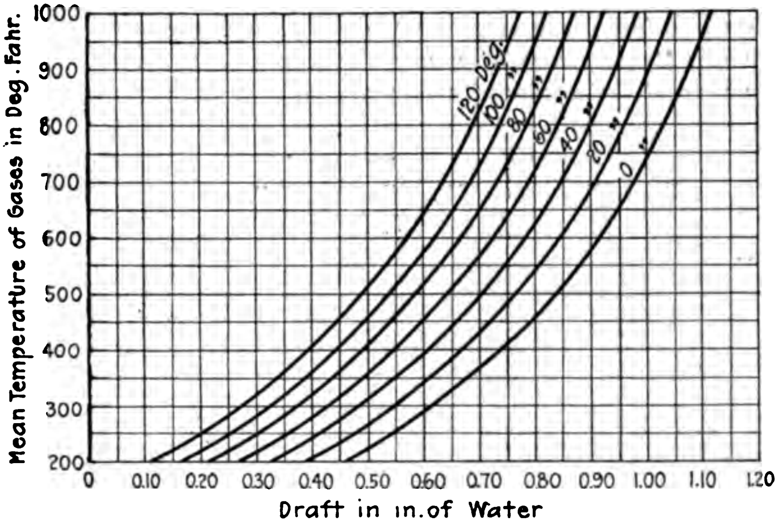


FIG. 3 MAXIMUM DRAFT AT SEA LEVEL PER 100 FT. OF CHIMNEY HEIGHT CORRESPONDING TO THE AIR TEMPERATURES NOTED ON THE CURVES

For any other height H in ft. multiply by 0.01 H
For any other altitude, multiply by the corresponding factor of correction from Fig. 4

32 To illustrate the applications of the formulae and data, the following problem will be considered: What sizes of chimneys will be suitable for two 500 h.p. boilers for continuous operation at not over 150 per cent of rating? The temperature of the air will not exceed 80 deg. fahr., and the breeching will be about 50 ft. long and have two right angle turns.

IX DETERMINING THE DRAFT REQUIRED AT THE BOILER DAMPER

33 As stated in Sec. VI, this must be assumed from a knowledge of the equipment and conditions. Ordinarily, the draft required

may be taken from characteristic curves, of which Fig. 2 is typical for boilers. An explanation of these curves is given in Sec. VI.

34 In connection with the draft required in the furnace for natural draft and coal fuel, the following formula will be useful.

$$C = \frac{(\text{h.p.}) \times 33480}{U\,E\,G}$$

where C is the maximum weight of coal to be burned in lb. per sq. ft. of grate surface per hr., (h.p.) is the maximum horsepower to be developed, U is the calorific value of the coal in B.t.u. per lb., E is the efficiency of the combined generator and G is the grate surface in sq. ft.

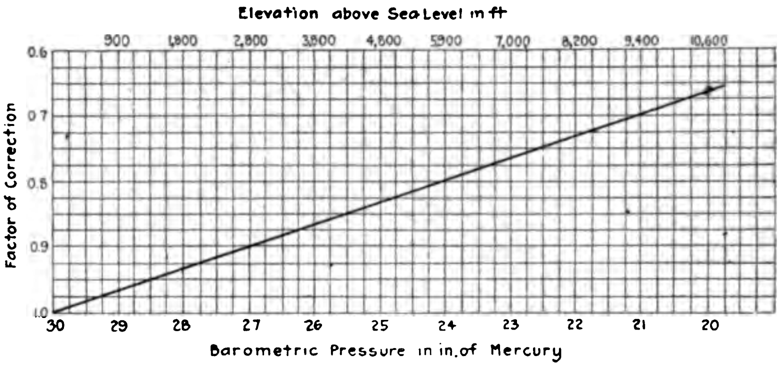


FIG. 4 FACTORS OF CORRECTION APPLICABLE TO FIG. 3

35 For the present problem, P_f will be taken as 0.25 in. and $P_b = 0.45$ in. The draft required at the boiler damper is then $0.25 + 0.45 = 0.70$ in.

X DETERMINING THE QUANTITY OF GAS TO BE TRANSMITTED

36 Table 1, which is based on the assumptions stated in Sec. VII, is for the general case.

37 The weight of gas per horsepower is approximately

$$W = \frac{33480\,w}{10{,}000\,(E + 0.02)}$$

where W is the weight of gas in lb. per hr. per boiler h.p., w is the weight of gas per 10,000 B.t.u., and E is the combined efficiency of the steam generator. w may be taken from Table 1. The quantity 0.02 added to E is an average allowance for carbon in the ash.

Example: At 9 per cent CO_2, $w=15.7$. Let $E=0.66$

$$W=\frac{33480\times 15.7}{10{,}000\ (0.66+0.02)}=77.3 \text{ lb.}$$

The weight per sec. is $W/3600$, and γ by formula [4] is $(41.3\,P) \div \{30\ (t_c+459.6)\}$. Hence the formula for the volume per sec. is

$$q=\frac{30\,W(t_c+459.6)}{41.3\,P\times 3600}=\frac{6.73\,W\ (t_c+459.6)}{1{,}000{,}000}\times\frac{30}{P}$$

where q is the volume of gas in cu. ft. per sec. per boiler h.p., W is the weight of gas in lb. per hr. per boiler h.p., P is the barometric pressure in in. of mercury and t_c is the temperature of the gases in deg. fahr.

TABLE 1 WEIGHT OF GASES AND PERCENTAGE OF HEAT REJECTED TO THE CHIMNEY FOR DIFFERENT PERCENTAGES OF CO_2 WHEN CO = 0. FOR COAL

Per Cent CO_2 in the Dry Gases by Volume	18.7	18.0	17.0	1 .0	15.0	14.0	13.0	12.0
Excess air in per cent of the theoretical minimum........	0	4	10	17	24	33	43	54
Weight of gases per 10,000 B.t.u. in the coal, lb.........	7.8	8.1	8.6	9.1	9.6	10.3	11 0	11.9
Chimney loss per 100 deg. fahr. in per cent of the calorific value of the coal.	1.85	1.92	2.04	2.16	2.28	2.44	2.61	2.82
Chimney loss per 500 deg. fahr..	9.25	9.60	10.20	10.80	11.40	12.20	13.05	1 .10

Per cent CO_2 in the Dry Gases by Volume	11.0	10.0	9.0	8.0	7.0	6.0	5.0
Excess air in per cent of the theoretical minimum	68	85	105	130	162	206	267
Weight of gases per 10,000 B.t.u. in the coal, lb	12.9	14.2	15.7	17.6	20.0	23.3	27.8
Chimney loss per 100 deg. fahr. in per cent of the calorific value of the coal	3.06	3.37	3.72	4.17	4.74	5.52	6 59
Chimney loss per 500 deg. fahr........	15.30	16.85	18.60	20.85	23.70	27.60	32.95

38 In ordinary problems, Table 2 may be used instead of calculating q by the formulae above. A variation of 100 deg. either side of the standard of 540 deg. fahr. alters the volume by 10 per cent.

39 Returning to the problem, the volume of gases to be removed by the chimney may now be calculated. The maximum horsepower to be developed is $2\times 500\times 1.5=1500$ h.p.

$q = 0.52$ from Table 2 for 9 per cent CO_2 and 66 per cent efficiency. Then the total volume of gases per sec. is

$$Q = 1500 \times 0.52 = 780 \text{ cu. ft.}$$

XI DETERMINING THE MAXIMUM DRAFT PRODUCED BY THE CHIMNEY

40 The general formula from Sec. II is

$$P_z = 0.255\, P\, H\left(\frac{1}{t_a + 459.6} - \frac{1.04}{t_c + 459.6}\right)$$

Values of P_z per 100 ft. of height and the factor of correction for different altitudes and pressures are given graphically in Figs. 3 and 4.

TABLE 2 WEIGHT AND VOLUME OF GAS PER BOILER HORSEPOWER AT SEA LEVEL

Assumed CO_2 per cent of dry gases by volume	8	8	9	10	12	12	14	14
Assumed combined efficiency of boiler, furnace and grate	63	66	66	68	70	75	68	78
Weight of gases in lb. per hr. per boiler h.p.	91	87	77	68	55	52	49	43
Volume of gases in cu. ft. per sec. per boiler h.p. for a temperature of 540 deg. fahr.	.61	.58	.52	.46	.37	.35	.33	.29
Suggested corresponding percentage of rated capacity of boiler to be used for proportioning chimneys	100	125	150	..	200	..	250	..

TABLE 3 ASSUMED ECONOMICAL VELOCITIES FOR A FIRST APPROXIMATION

Volume of gas to be removed in cu. ft. per sec.	10	50	150	500	1200	2500	5000	8000
Economical velocity in ft. per sec.	10	15	20	25	30	35	40	45

41 In the problem $t_a = 80$ deg. fahr. Let $t_c = 500$ deg. fahr. Then from Fig. 3, $p_z = 0.59$ in. If the altitude is 5900 ft. above sea level, then the factor of correction from Fig. 4 is 0.8. Hence $p_z = 0.8 \times 0.59 = 0.47$ in.

XII DETERMINING THE DRAFT LOSS IN THE CHIMNEY

42 The formula for this, as developed in Sec. V, and by means of which the values shown graphically in Fig. 5 were obtained, is

$$P_r = 0.008\frac{l}{d}\left(\frac{v^2}{t_c + 459.6}\right)$$

43 A variation of 100 deg. either side of the assumed standard of 540 deg. fahr. and a variation of 3 in. in barometric pressure, if uncorrected for, would each introduce an error of 10 per cent in the draft loss. Since the draft loss is usually small, it would seem that ordinarily only variations beyond these limits need be considered.

44 Before the curves in Fig. 5 may be used, it is necessary to assume a velocity. If v is assumed then A follows from the formula $A = Q/v$. For a first approximation, the velocity may be selected from Table 3.

45 In the present problem, the volume of gas to be removed is 780 cu. ft. per sec. Assuming a velocity of 25 ft. per sec., $A = 780/25 = 31.2$. The friction loss per 100 ft., by visual interpolation on Fig. 5, is $p_c = 0.08$ in. This is for a circular chimney at sea level.

46 For a square chimney of side equal to diameter of the circular chimney $p_c = 0.62 \times 0.08 = 0.05$ in. For a circular chimney and barometric pressure 24 in. of mercury, $p_c = (30/24)\ 0.08 = 0.10$ in. For a circular chimney at sea level but gas temperature equal to 640 deg., $p_c = \{(640+460) \div 1000\} 0.08 = 0.09$ in. For a square chimney, barometer at 24 in. and gases at 640 deg. fahr.,

$$p_c = 0.62 \frac{30}{24} \frac{640+460}{1000} 0.08 = 0.07 \text{ in.}$$

XIII DETERMINING THE FRICTION LOSS IN THE BREECHING

47 The customary assumption is that one sharp right angle turn has the same friction as 50 ft. of length. In the problem the breeching is to be 50 ft. long and to have two turns. Hence $l = (2 \times 50) + 50 = 150$ ft. Assuming the breeching will be square, with side equal to the diameter of the circular chimney, then $P_r = 1.5 \times 0.05 = 0.08$ in. The quantity 0.05 is the loss per 100 ft. as computed in Sec. XII. For a circular breeching of the same size as the chimney $P_r = 1.5 \times 0.08 = 0.12$ in. This does not indicate that the total loss in the square breeching will be less than that in the circular one because of the draft losses due to changes of velocity, as discussed in Sec. XIV.

XIV DRAFT REQUIRED TO INCREASE VELOCITY

48 The formula as developed in Sec. III is

$$P_v = 0.123 \frac{P\,(v_2^2 - v_1^2)}{30\,(t_c + 459.6)}$$

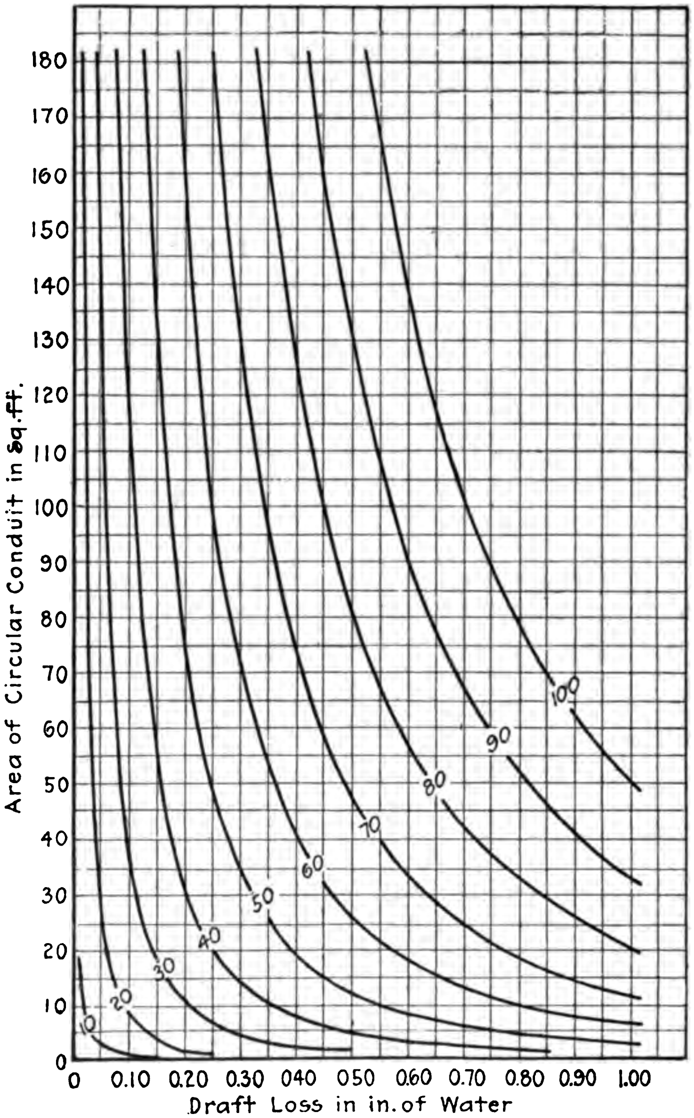


FIG. 5 DRAFT LOSS PER 100 FT. OF A CIRCULAR BRICK-LINED CONDUIT CORRESPONDING TO THE VELOCITIES IN FT. PER SEC. NOTED ON THE CURVES. FOR BAROMETER AT 30-IN. MERCURY AND GASES AT 540 DEG. FAHR.

For any other height or length H in ft., multiply by 0.01 H

For a square conduit of side equal to the diameter of a circular conduit, multiply by 0.62 when the volume of gas remains the same

For any other barometric pressure P in in. of mercury, multiply by $30/P$ when the weight of gases remains the same

For any other gas temperature t in deg. fahr. multiply by 0.001 $(t+460)$ when the weight of gases remains the same

The draft required to accelerate gases at sea level from zero velocity to velocities of 10, 20, 30 and 40 ft. per sec. for a temperature of 540 deg. fahr. is 0.012 in., 0.049 in., 0.111 in. and 0.197 in. respectively. The magnitude of this loss at the higher velocities explains the regulating effect produced by manipulating the damper.

49 The ratio of the area of a square to that of an inscribed circle is 1/0.7854. The velocity in the former will therefore be 0.7854 times the velocity in the latter. Table 4 shows the draft loss due to a velocity change based on this ratio for the case when gases pass from a square breeching into a circular chimney.

TABLE 4 DRAFT REQUIRED TO ACCELERATE GASES AT 540 DEG. FAHR., BAROMETER AT 30 IN.

Change of velocity in ft. per sec........	11.8 to 15	15.7 to 20	19.6 to 25	23.6 to 30	27.5 to 35	31.4 to 40	35.3 to 45	39.3 to 50
Draft required in in. of water..........	0.011	0.019	0.030	0.042	0.058	0.075	0.096	0.118

50 Except at the highest velocities, this draft is small enough to be neglected in the problem of calculating sizes of chimney; but as regards the design of the breeching this additional loss, for short breechings, may make the smaller circular breeching more desirable than the larger square breeching from an operating as well as a cost standpoint provided the gases leave the boilers at about the ultimate velocity.

51 Since the loss due to the velocity change will almost compensate for the decreased friction when the square breeching is used, if it is not long, then the friction loss per 100 ft. of breeching may be taken the same as in the circular chimney, whether the breeching will be square or circular.

XV LOSS OF DRAFT DUE TO SUDDEN ENLARGEMENT OF THE GAS PASSAGE

52 The formula from Sec. IV is

$$P_v' = 0.123 \frac{P\,(v_1 - v_2)^2}{30\,(t_0 + 459.6)}$$

For a change of velocity from 30 to 10 ft. per sec., at sea level and temperature equal to 540 deg. fahr. this loss is 0.049 in. Where there are several sudden enlargements in a long flue, the aggregate loss may be considerable. Hence gradual changes of cross section are preferable where there is something to be gained by enlarging the breeching.

XVI DETERMINING THE VALUE OF d IN INCHES

53 As stated in Sec. V, d is the diameter of a circular chimney or breeching, or the length of one side of a square chimney or breeching. The curves in Fig. 5 are based on circular conduits, hence d is always the diameter corresponding to the area A. Obviously, a quick way to determine d is to divide the area A by 0.7854 on the upper scale of a slide rule and read d in in. opposite 12 on the lower scale.

XVII PRACTICAL METHOD FOR CALCULATING CHIMNEYS

54 Solutions of chimney problems are very simple if carried out in the following form:

Equipment and Location. Two 500 h.p. boilers to be operated up to 150 per cent of rating. Breeching to have two right angle turns and to be about 50 ft. long. Chimney to be circular. Plant to be at about sea level.

Assumptions. $t_a = 80$ deg. fahr., $t_c = 500$ deg. fahr., $q = 0.52$ cu. ft. (from Table 2) $P_f = 0.25$ in. $P_b = 0.45$ in. $P_f + P_b = 0.70$ in.

Calculations. Maximum h.p. $= 2 \times 500 \times 1.5 = 1500$.

$Q = 1500 \times 0.52 = 780$ cu. ft. Assume $v = 25$ ft. (from Table 3).

$A = Q/v = 780 \div 25 = 31.2$ sq. ft.

p_c from A and v on Fig. $5 = 0.08$ in.

$l = (2 \times 50) + 50 = 150$ ft. $P_r = 1.5\ p_c = 1.5 \times 0.08 = 0.12$ in.

$P_e = (P_f + P_b) + P_r + P_v + P_v' = 0.70 + 0.12 + 0 = 0.82$ in.

$p_z = 0.59$ in. from Fig. 3. $p_z - p_c = 0.59 - 0.08 = 0.51$ in.

$$H = \frac{100\ P_e}{p_z - p_c} = \frac{100 \times 0.82}{0.51} = 161 \text{ ft.}$$

55 By writing the above calculations in the following tabular form (Case 1a) and assuming other velocities then other sizes may be obtained quickly.

Case	v	A	p_c	P_r	P_v	P_e	$p_z - p_c$	H	d
1*a*	25	31.2	0.08	0.12	0.0	0.82	0.51	161 ft.	76 in.
1*b*	20	39.0	0.04	0.06	0.0	0.76	0.55	138 ft	85 in.
1*c*	30	26.0	0.12	0.18	0.0	0.88	0.47	187 ft.	69 in
1*d*	40	19.5	0.26	0.39	0.15	1.24	0.33	376 ft.	6[illegible] in.

56 The value of P_v in Case 1*d* is based on the assumption that the gases leave the boiler at a velocity of 20 ft. per sec. In the other cases P_v was assumed to be negligible.

57 The best size of chimney will depend, of course, on relative cost, local and other conditions.

58 To illustrate further, assume that the plant will be at an elevation of 5900 ft. above sea level and that the chimney will be square. Then $P = 24$ in. The maximum draft, p_z', will be 0.8 p_z (Fig. 4) and the friction loss in the breeching and chimney per 100 ft. p_c' will be $0.62 \times 30/24$ $p_c = 0.78$ p_c, where p_c is the loss at sea level. Hence

$$p_z' = 0.8 \times 0.59 = 0.47 \text{ in. } P_r = 1.5\, p_c'$$

Case	v	A	p_c	p_c'	P_r	P_v	P_e	$p_z' - p_c'$	H	d
2*a*	25	31.2	0.08	0.06	0.09	0	0.79	0.41	193 ft.	7[illegible] in.
2*b*	20	3[illegible].0	0.04	0.03	0.04	0	0.74	0.44	1[illegible]8 ft.	85 in.
2*c*	15	52.0	0.02	0.02	0.03	0	0.73	0.45	162 [illegible].	97 in

59 Table 5 illustrates how the capacity of a chimney is affected by a variation of one or more of the governing conditions. The altered conditions are in italics. Since the volume of gas per sec. per h.p. is dependent on both temperature of gases and combined efficiency, its value has been altered accordingly when these are involved.

TABLE 5 VARIABLE CAPACITY OF A CIRCULAR CHIMNEY 76 IN. IN DIAMETER AND 160 FT. HIGH

Case	Temperature of Air, Deg. Fahr.	Temperature of Gases, Deg. Fahr.	Barometer, In.	Draft in Furnace, In.	Draft Loss in Boiler, In.	Draft Loss in Breeching, In.	Gas per Sec. per h.p. Cu. ft	Capacity of the Chimney B.h.p.
1*a*	80	500	30	0.25	0.45	0.12	0.52	1500
3	*100*	500	30	0.25	0.45	0.12	0.52	900
4	*20*	500	30	0.25	0.45	0.12	0.52	2700
5	80	*900*	30	0.25	0.45	0.12	*0.90*	2000
6	80	500	30	0.25	0.45	*0*	0.52	2100
7	80	500	*27*	0.25	0.45	0.12	0.52	750
8	80	500	30	*0.[illegible]0*	*0.22*	0.12	*0.[illegible]5*	3900
9	80	500	30	*0.10*	*0.22*	0.12	*0.35*	4400
10	80	*600*	30	*0.10*	*0.60*	*0*	*0.33*	4200

and temperature equal to 540 deg. fahr. this loss is 0.049 in. Where there are several sudden enlargements in a long flue, the aggregate loss may be considerable. Hence gradual changes of cross section are preferable where there is something to be gained by enlarging the breeching.

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4	*20*	500	30	0.25	0.45	0.12	0.52	2700
5	80	*900*	30	0.25	0.45	0.12	*0.90*	2000
6	80	500	30	0.25	0.45	*0*	0.52	2100
7	80	500	*27*	0.25	0.45	0.12	0.52	750
8	80	500	30	*0.[illegible]0*	*0.22*	0.12	*0.[illegible]6*	3900
9	80	500	30	*0.10*	*0.22*	0.12	*0.36*	4400
10	80	*600*	30	*0.10*	*0.60*	*0*	*0.33*	4200

60 Cases 1*a*, 3, 4, 6 and 7 are for a boiler in good condition but inefficiently fired. Cases 1*a*, 3, 4 and 7 are for different atmospheric conditions; Case 5 is representative of a boiler with baffles and heating surface in need of attention; Case 6 is for a chimney set directly over the boiler; Case 8 would apply to a natural draft installation efficiently operated; Case 9 is for an oil burning or forced draft installation efficiently operated; and Case 10 is for efficient operation at about 250 per cent of rating with forced draft and chimney set directly over the boiler.

DISCUSSION

A. G. Christie. The author has rendered a distinct service in presenting the derivation and application of the various formulae relating to chimney problems when considered on the basis of the gases handled. His methods are quite rational and could be applied readily if definite information were available on all the factors.

Engineering literature is distinctly lacking in available data on the performance of chimneys, particularly as regards what takes place in the chimney itself. The variations of temperature, density, and draft from top to bottom of the various types of chimneys are practically unknown. Hence the velocities and volumes in the chimney are also unknown. The effect of wind has not been worked out, particularly in regard to the air leakage into the chimney under heavy winds, to the cooling of the gases, or to the suction, if any, over the top of the stack. Some recent experimental work has shown that humidity has a very appreciable effect on the flow of air in pipes. It is, therefore, reasonable to expect that humidity in chimney gases, due to the combustion of the volatile matter of bituminous coal and oils, and particularly in the case of lignite, must also have an effect on the operation of the chimney.

The chimney of the power plant of the new Johns Hopkins University has been specially built with the particular object in view of studying certain of these problems, and some work along this line will be undertaken in the near future.

Mr. Menzin submits tables based on assumed gas volumes and velocities, but does not supply any experimental data to indicate which, if any, of these velocities gives the best performance, nor are his friction factors derived from actual stack performance. The value of the paper would have been greatly enhanced by the addition of these experimental data.

No. 1519

THE CONNORS CREEK PLANT OF THE DETROIT EDISON COMPANY

By C. F. Hirshfeld, Detroit, Mich.
Member of the Society

The phenomenal growth of Detroit's population and industries has been widely heralded, but it is probable that the extent of this growth and its significance to the central station industry is not appreciated by those not closely in touch therewith. For this reason the curves in Fig. 1 are shown. The upper curve, showing the variation of population during the past decade, is probably approximately correct, because it fits smoothly into the curve obtained by plotting United States Census figures and because it checks very closely with the more accurate of the estimates which have been made from time to time. This curve indicates that the population of Detroit in 1914 was about 1.6 times as great as it was in 1904, just ten years before.

2 The annual output of the central stations, as given by the next lower full-line curve, has increased much more rapidly than the population. In the year 1914 it was about 21.4 times as great as in 1904. The fact that the maximum annual peak was only ten times as great in 1914 as in 1904 indicates that a very large part of the increased annual output was due to increased industrial application. This is also partly indicated by the curve showing variation of load factor.

3 A map of the City of Detroit is given in Fig. 2. The heavy full lines radiating, roughly, from a point near the bottom center of the figure indicate radial streets upon which the principal car lines are operated. These naturally determine the directions of growth and they indicate that, other things equal, Detroit's area may be expected to preserve a roughly semicircular shape as it expands. The heavy dotted lines indicate the rights of way of the various steam railroads entering the city. Other things equal, these rights of way, combined with the river frontage, indicate the probable future loca-

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Buffalo Section.

tions of the larger manufacturing industries. Apparently, these industries may be expected to scatter all along these lines in the future, as they have in the past.

4 The small rectangular spots indicate the locations of the various substations of the Detroit Edison Company. These are so drawn that the length of a side indicates the relative capacity. The concentration near the center from which the radial streets diverge is due to the fact that this is the business center of the city and that it is served with direct current. This gives a very concentrated and very important direct current load at this point.

5 The two circular spots near the lower left hand corner of the

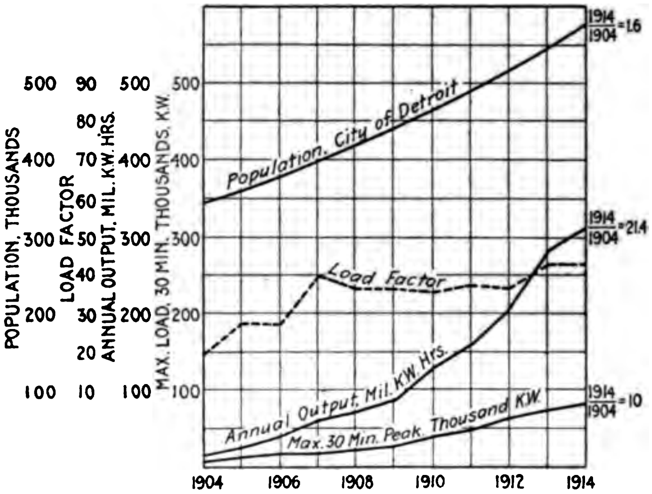

Fig. 1 Curves Indicating Growth of City and Load

figure indicate the location of Delray power houses No. 1 and No. 2. These have an aggregate maximum capacity of about 95,000 kw. and contained practically all the generating equipment of the company before the Connors Creek plant was built. They are adjacent to one another and operated as one plant.

6 When it became evident about 1912 that greatly increased capacity would soon be required, two possible solutions were available; a third power house could have been built at the Delray site or a new site could have been selected. Consideration of the direction of growth of the city; the rapidly increasing population and industrial development of the east side; the location of the heavy direct current load; necessity for continuity of service with an ever increasing com-

Fig. 3 after being weighed on the railway scales indicated near the upper left-hand corner of that figure. The cars dump into hoppers under the tracks in the train or coal shed, there being one hopper for each unit of one turbine and two boilers. A motor-driven, variable-speed flight conveyor with a capacity of 120 tons per hour, receives the coal from the hopper, carries it up a rather sharp incline and discharges it into a four-roll motor-driven crusher of similar capacity. This crusher breaks from 18-in. cubes, or smaller, to 100 per cent through a 1¾-in. ring.

12 The crusher discharges directly into a motor-driven, constant-

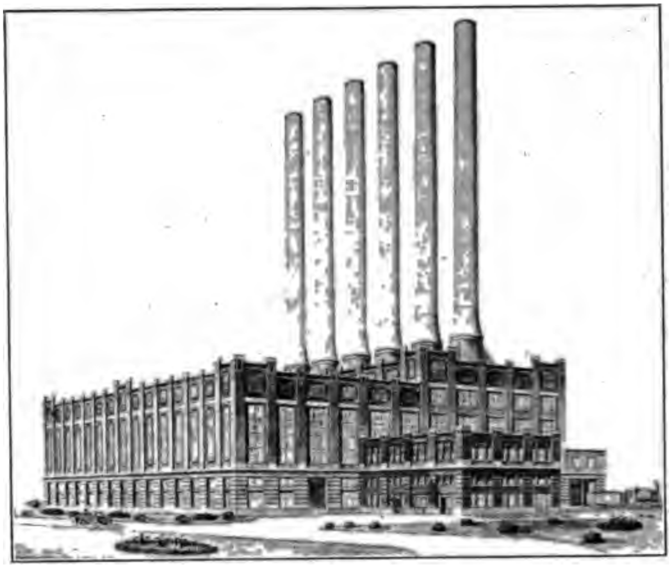

FIG. 4 THE NEW CONNORS CREEK POWER HOUSE, AS IT WILL APPEAR WHEN COMPLETED

speed, lapping bucket conveyor with 30 by 36-in. buckets. This conveyor forms an endless chain which entirely encloses the section of the boiler house, as indicated in Fig. 5. It carries the coal up on the side nearest the coal shed and discharges it into any one of the three coal bunkers which serve the two boilers of one unit. The hopper, pan conveyors, crusher and bucket conveyor for each unit of one turbine and two boilers are so located that they can deliver through chutes to one adjacent range of bunkers, serving thus as a spare for that range.

13 The entire coal handling equipment is electrically operated

houses at Delray combined. The first house was planned to accommodate six 25,000 k.v.a. units, but it seems probable that this unit size will be increased before the last units are installed. The plan of this house is shown in Fig. 3, together with the canals and tracks which serve it. The river is located beyond the left-hand side of the figure and runs in a direction roughly parallel to the left-hand end of the plate, and upward. The diagonal position of the plant was dictated by the shape of the site, the necessity of leaving room for a future power house, the curves required for railroad tracks, and the location of the river which determined the direction and location of

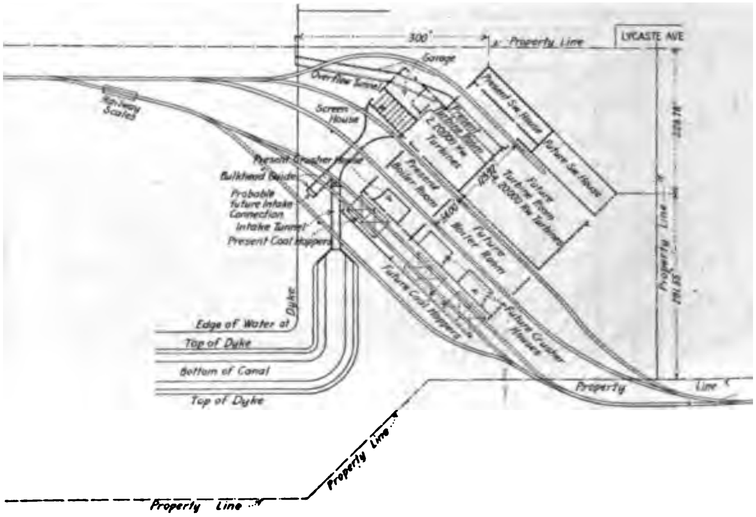

FIG. 3 PLAN OF SITE OF THE NEW CONNORS CREEK POWER HOUSE

the canals. One-third of the first power house on the Connors Creek site has been built and two of the proposed six units are installed. The building is closed with a temporary end, so that the machinery now installed can be operated until and while future extensions are being constructed. The intake and overflow canals are completed in full size for six units.

10 An architect's drawing of the finished power house is given in Fig. 4 and a sectional elevation in Fig. 5. The latter shows the construction to best advantage and should be consulted in connection with the description which follows.

11 The coal enters the train shed in drop-bottom cars (usually of the 50-ton size) which move on the lower group of tracks shown in

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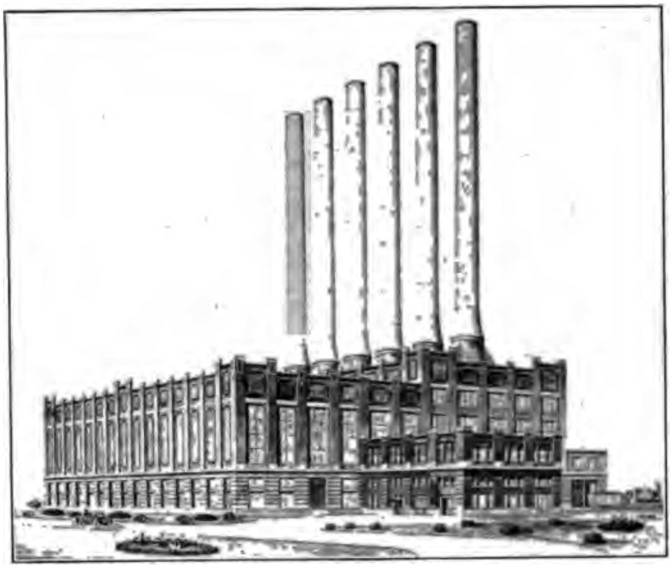

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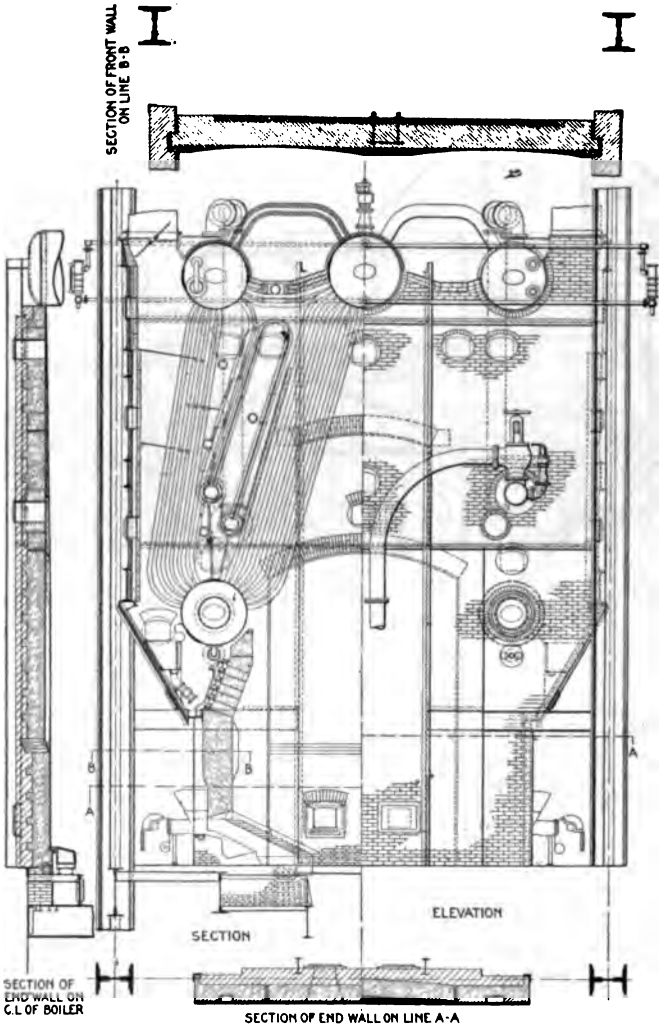

FIG. 6 SECTION AND ELEVATION OF ONE BOILER

boiler and holds the refuse until it is finally dropped directly into standard railroad cars which run lengthwise of the house directly under the ash hoppers.

15 The stokers are driven through chains by direct current

motors, the speed of which can be varied from 200 to 2000 r.p.m. This wide range of speed is obtained by means of armature throw over from 125 to 250 volts combined with field control. The motors are handled by means of drum controllers located at the gage board of the particular boiler which they serve and electric tachometers are used to indicate stoker speeds at this board.

16 Motor driven blowers supply the air required for combustion. There are three of these blowers per range of two boilers. One of the three serves as a spare and is located between the two boilers. It is arranged to discharge either way through a Y-shaped duct. All of the blowers discharge through expanding ducts which are designed to recover the greatest possible fraction of the velocity head. These large ducts lead into a plenum chamber with a horizontal section equal to the horizontal section of the furnace. The stoker wind boxes form part of the roof of this chamber, so that the air passes directly from the plenum chamber to the stoker and does not have to pass through small ducts of any kind. Each blower is designed to deliver at maximum 74,000 cu. ft. of air per minute against a static pressure of 6.5 in. of water pressure. The motors are shunt wound and are direct connected to the blowers through flexible couplings. Their speed can be varied from 300 to 750 r.p.m. The drum type controllers are mounted at the gage boards of the boilers served, directly under gages which indicate the static pressure in the plenum chamber.

17 The boilers are similar to those used in the older plant at Delray, but are set 3 ft. higher so that the height of the combustion chamber is 33 ft. These boilers each have 23,654 sq. ft. of water heating surface and are built to operate at 225 lb. per sq. in. The superheaters have about 2400 ft. of surface and are designed to give 200 deg. fahr. superheat at 200 per cent of rating.

18 The gases, leaving the dampers at the top of the boiler setting, pass upward through easy curved breechings into the steel stack near its base. This stack is supported entirely on the steel work of the boiler room structure and extends to a height of 325 ft. above the floor on which the stokers are located. It has a height of about 240 ft. above the roof of the boiler house. The stack is brick lined and has a diameter of 16 ft. inside the lining.

19 As already mentioned, there are only two boilers per turbine. When operating the turbine at full load, the boilers will operate at about 170 per cent of Centennial rating. In the ordinary sense, there are no spare boilers. Experience has shown, however, that these

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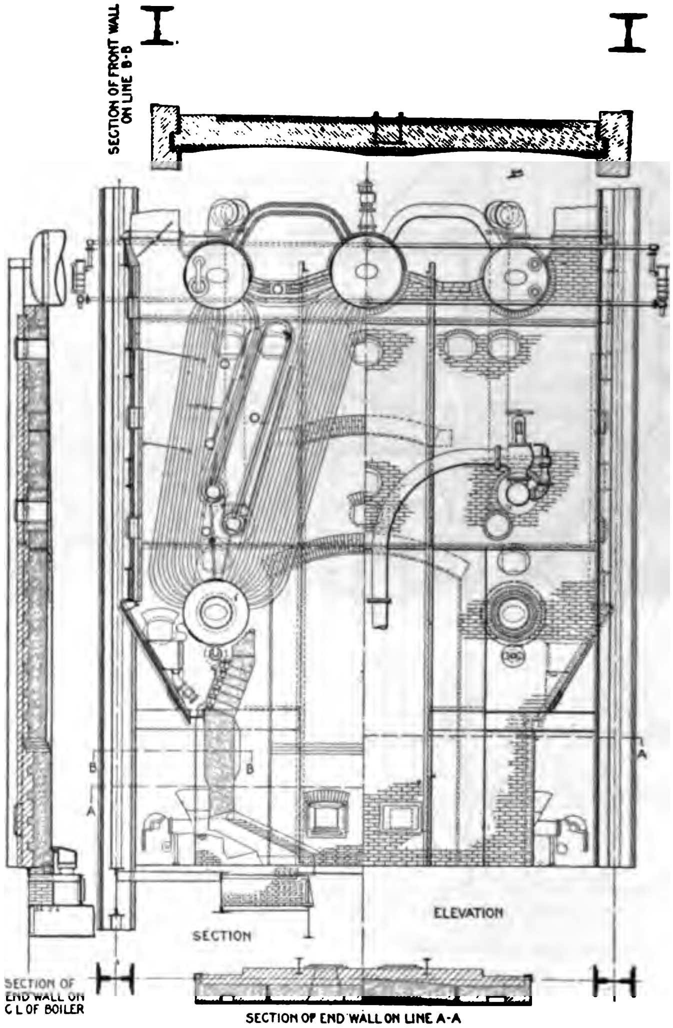

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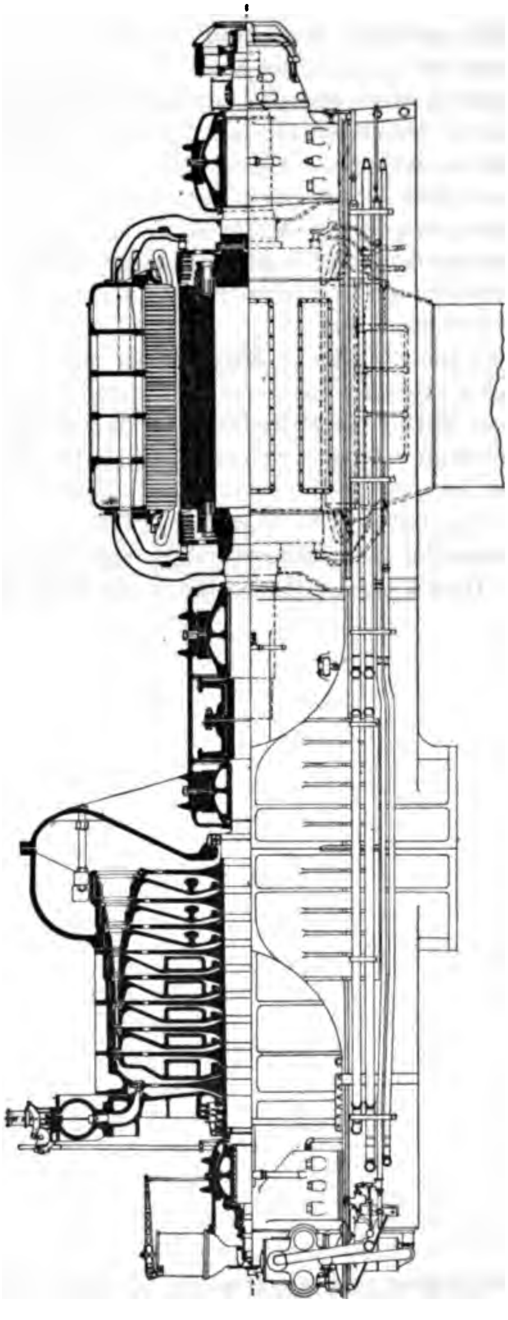

FIG. 8 SECTIONAL VIEW OF ONE OF THE 25,000 K.V.A. TURBO-GENERATOR UNITS

that shown in Fig. 7 was adopted. The steam, leaving the 10 by 14 by 10-in. Y-branch previously mentioned, passes through a cast steel expanding nozzle which enlarges to a diameter of 28 in. This, in turn, leads into a 28-in. cast steel side-outlet T or side-outlet cross. The 28-in. lateral outlets of the latter fittings are the connection points of the cross-over main. The velocity of the steam passing into the cross-over, or from the cross-over main to the turbine lead, is thus reduced to about one-quarter of its value in the 14-in. pipes, or roughly, a little less than 4000 ft. per min. under the worst conditions. The steam turns through the necessary right angle at this low velocity and therefore with small loss.

23 Steam leaving the 28-in. fitting on its way to the turbine is carried through a tapered reducer with a discharge diameter of 14 in. Similarly, steam leaving the 28-in. fitting on its way to the cross-over main passes through a tapered reducer which is cast with its longitudinal axis in the shape of a quarter circle. This reducer leads the steam into a 14-in. return bend, as shown in Fig. 7.

24 The steam for the auxiliary turbines, which will be mentioned later, is taken from a 6-in. outlet on top of the 28-in. fittings above described.

25 All superheated steam piping is full weight steel with welded flanges. The flanges are finished smooth and corrugated steel gaskets are used. All fittings are cast steel. At the Delray plant, Hopkinson-Ferranti valves with venturi throats were used; these have been found to cause an excessive pressure drop because of the short length of the expanding nozzle and were therefore not deemed desirable for the new plant if an equally reliable, full opening form could be found. Full throated gate valves made by Hopkinson of the same material as the Ferranti valves were therefore used in the Connors Creek plant.

26 The atmospheric exhaust from the main unit is made of riveted steel pipe and fittings. The auxiliary exhaust piping is lap welded steel with Van Stone joints and fitted with corrugated copper gaskets. All valves in the exhaust lines are steel gate valves of American make. All saturated steam piping is extra heavy steel fitted with steel flanges. The fittings are all cast steel and steel valves of American make are used.

27 The feed water piping is extra heavy, lap welded steel, with Van Stone joints and cast steel fittings. The use of steel pipe throughout for the feed water represents a marked departure from Delray practice in which brass pipe was used for the boiler leads and con-

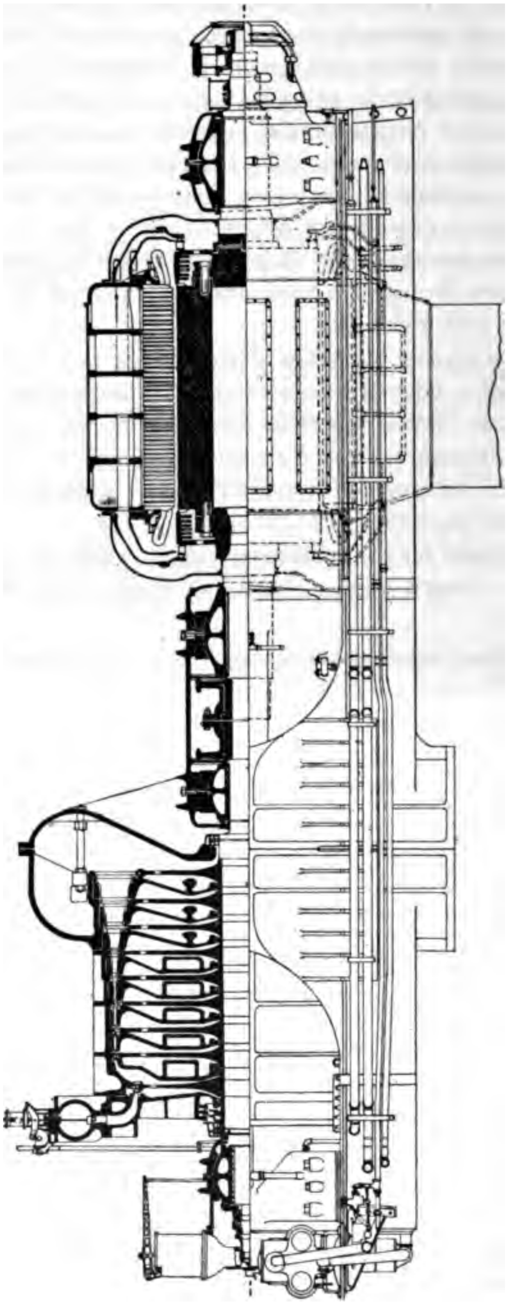

Fig. 8 Sectional View of One of the 25,000 k.v.a. Turbo-Generator Units

nections. In laying out the feed water system at Connors Creek, the design was so arranged as to permit of free expansion without the use of expansion joints of any kind.

28 The boiler blow-off valves are also made by Hopkinson. These were decided on after two of them had been in satisfactory operation for several years at Delray. The valve is a parallel-seat gate, on whose stem is cut a rack which meshes with a pinion. A 180-deg. turn of the handle turns this pinion through a like angle and fully opens the gate. The superiority of the valve appears to be due to the use of excellent metal and to a very high class of machine work.

29 The main units, as previously mentioned, are rated at 25,000 k.v.a. or 20,000 at 80 per cent power factor. The steam end consists of a horizontal, nine-stage, 1200 r.p.m. turbine; the electrical end generates three-phase, sixty-cycle current at 4800 volts. A section of the turbo-generator unit is shown in Fig. 8. The steam exhausted from the turbine passes down through a large expanding exhaust nozzle into the condenser, as shown in Fig. 5. Each condenser is built to contain 35,000 sq. ft. of heating surface made up of 1-in. tubes a little over 18 ft. long, but has only 32,500 sq. ft. installed. The tube heads have a diameter of 14 ft. 6 in.

30 The tubes are so arranged as to leave numerous lanes through the steam space, so that the vapor and air readily flow to all parts of the cooling surface. No baffles of any kind are used, excepting only those necessary to form an air box. The latter is located at the bottom of the condenser, as indicated in Fig. 9.

31 The condensate collects in a cylindrical hot-well which extends from the lower part of the condenser shell, and it is removed by either one of two hot-well pumps. These are motor-driven two-stage centrifugal pumps operated at 1200 r.p.m. and they are fitted with bronze impellers. The condensate is discharged into what is known as the cold end of the boiler feed tank, as will be explained.

32 Circulating water is supplied to each condenser by a motor-driven, single-impeller, double-suction, volute pump. The motor is three-phase, slip-ring type arranged for a 15 per cent speed variation, the maximum speed being 400 r.p.m. During the period of the year when the circulating water has a high temperature, the pump will be operated at the highest speed and will deliver about 40,000 gal. per min. When the water temperature is low, the pump will be operated at the lower speed and the quantity will be proportionately reduced.

The temperature of circulating water varies with the season from 34 deg. to 76 deg. fahr.

33 The circulating water enters the property through an open cut shown in the plan in Fig. 3 and thence flows through a concrete tunnel to the screen house. This tunnel has a free section 9 ft. 3 in. high by 10 ft. wide, and when it is supplying water for all of the units

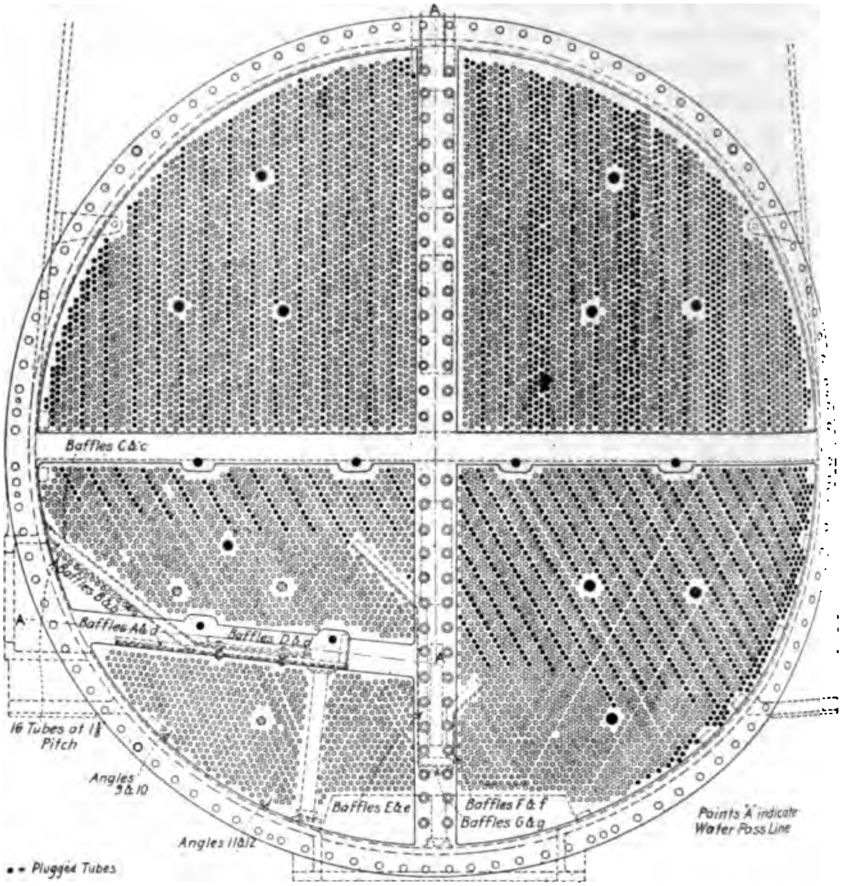

FIG. 9 DETAILS OF TUBE ARRANGEMENT IN THE CONDENSERS

in the finished plant under the lowest water conditions on record, the water velocity will be just under 8 ft. per sec.; in obtaining this, further units are assumed to be larger than those now installed.

34 The water flows from this section of the tunnel into a sort of funnel-shaped enlargement which leads it to the screens. These are

mixture is there picked up by the boiler feed pump and delivered to the boilers.

42 The barometric condenser is therefore the equivalent of an open feed water heater in which exhaust steam from auxiliaries mixes with and heats the condensate from the main units, the mixture passing to the boilers as feed water. This condenser, however, offers possibilities not possessed by the ordinary open heater in that it is capable of maintaining a very low back pressure, if this is desired, or can be operated at any pressure between this and atmospheric. This property, taken in conjunction with the great variation of the auxiliary turbine water-rate with variations of back pressure, gives great flexibility of control of the station heat balance.

43 This is shown diagrammatically in the upper left hand corner of Fig. 11. The straight lines represent total steam consumption of the house alternator turbines for the entire range of load at different back pressures. Assume that at some particular time with a given load and vacuum on the main unit, the load on the house alternator has the value indicated by the vertical line, and that the back pressure is that corresponding to the middle steam consumption line, as at *a*. If the feed water temperature is too high it can be lowered by decreasing the back pressure, reducing the steam consumption to some such value as that shown at *c*. In this way, the feed water temperature can be accurately controlled, just enough auxiliary exhaust being made available to give the desired temperature.

44 The variation of the back pressure is obtained by means of a back-pressure valve in the exhaust line from the boiler feed pump turbine. If, under given constant conditions, this valve is partly closed down, the back pressure on the boiler feed pump turbine is increased and its steam consumption for the same load is correspondingly raised. This, however, means that more steam enters the barometric condenser and, with a constant quantity and temperature of circulating water (main condensate), the temperature and pressure in the barometric condenser must rise. This, in turn, means higher back pressure on the house alternator turbine and it delivers more steam, still further raising the temperature.

45 Under certain conditions, this interchange may result in a cumulative temperature rise, ending only when the pressure in the barometric becomes equal to that of the atmosphere and the auxiliary exhaust blows to waste. To prevent such an occurrence, the flood valve shown in Fig. 11 is provided. This is automatically opened whenever

centrifugal pumps, driven by 350-h.p. steam turbines, at 2000 r.p.m.

40 The motor-driven auxiliaries may be operated, all or any of them, with power taken from the system bus, or with power taken from turbine driven, 1000-kw. alternators, known as house service alternators. In every case, that arrangement is used which, in conjunction with the other adjustments described below, will give the best heat balance for the station, under the conditions at the time existing.

41 What may be called the house service system is shown diagrammatically in Fig. 11. It is really a development of a method long used in both marine and stationary practice, in which a feed

FIG. 10 INTERIOR OF SCREEN HOUSE

water heater is operated under vacuum. In the present instance, the wet vacuum pump of each main turbine unit discharges into one end of a large tank, known as the boiler feed tank. A centrifugal pump draws its water from the same end of this tank and discharges it into the head of a barometric condenser, as shown in Fig. 11. The relatively cold condensate is picked up by the second pump before it has time to mix with the mass of water in the tank, and serves as injection water for the barometric condenser. The house-service alternator turbine and the boiler feed pump turbine exhaust into this barometric condenser, so that the condensate from the main unit takes up all the heat of the auxiliary steam. The foot of the barometric condenser is immersed in the other (or hot) end of the boiler feed tank. The mixture is there picked up by the boiler feed pump and delivered to the boilers.

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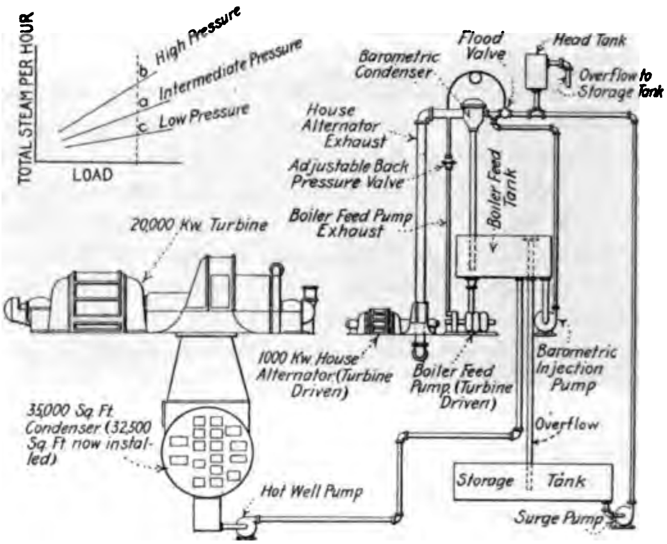

Fig. 11 Arrangement of the House Service System

engineer would make necessary re-adjustments before the valve had acted many times.

46 The house alternator, boiler feed pump, and barometric injection pump are all grouped under the gallery shown at the right-hand side of the turbine room in Fig. 5. The natural position for the watch engineer is at the same place, so that continuous observation of this apparatus is readily obtained. An auxiliary gage board containing instruments showing critical temperatures, pressures and quantities is located within easy view, and the results of all adjustments are readily seen.

47 This method of operating the auxiliaries appears very complicated when described on paper, but it has proved to be very simple

in practice, and easy to handle. It possesses the following advantages which should be compared with methods more commonly used:

First The one great advantage of steam driven auxiliaries is retained because exhaust steam is available for feed water heating

Second The quantity of auxiliary exhaust can be fitted to the ability of the feed water to absorb it

Third All the advantages of motor drive are obtained

Fourth Up to the ability of the feed water to absorb steam, the power used by the motor driven auxiliaries is produced at a thermal cost practically equal to that obtained in the case of steam driven auxiliaries

Fifth Since the feed water heater is normally operated under a vacuum, the feed water temperature is readily maintained at a value which will give ideal economizer operation

Sixth Because of the vacuum maintained in the feed water heater, air is readily removed from the feed water just before it enters the boiler feed pumps. In practice there is no reabsorption of air by the condensate between main condenser and boilers.

48 Another unusual feature is the provision for the distillation of all make-up water. Experience at Delray has shown that the quantity of make-up required is readily held down to 1.5 or 2 per cent. Under such conditions, the apparatus required for distillation is small and the cost is negligible in comparison with the cost of boiler labor saved. The evaporator installed has a capacity of 12,000 lb. of vapor per hour and is heated by high pressure steam. The vapor formed in it passes directly to the barometric condenser in which it mixes with the auxiliary exhaust and thus becomes part of the feed water.

49 Many cases of boiler pitting and corrosion which are recorded in engineering literature have been attributed to the use of very pure water and the presence of a small quantity of scale-forming material has therefore been regarded as desirable. Most authorities now seem to believe that pure water is not responsible for such damage to good boiler metal, but that the blame is to be laid on small quantities of atmospheric carbon dioxide dissolved in what is assumed to be pure water.

50 In the Connors Creek plant, the feed water heating and storing system has been so designed as to prevent the absorption of air, so far

as prevention is readily possible. Should experience show that the provisions made are inadequate to insure safe boiler operation, there are several simple remedies. The simplest is probably a very small quantity of an alkaline salt, experience having shown that if the water shows a mere trace of alkalinity it is apparently non-corrosive.

51 Another innovation in this station is the very complete provision for the recovery of radiation and electrical losses. Air enters the turbine room principally through louvres above the air washer shown in Fig. 5. This air, after being washed if necessary, passes down through the turbine room, picking up part of the heat lost by

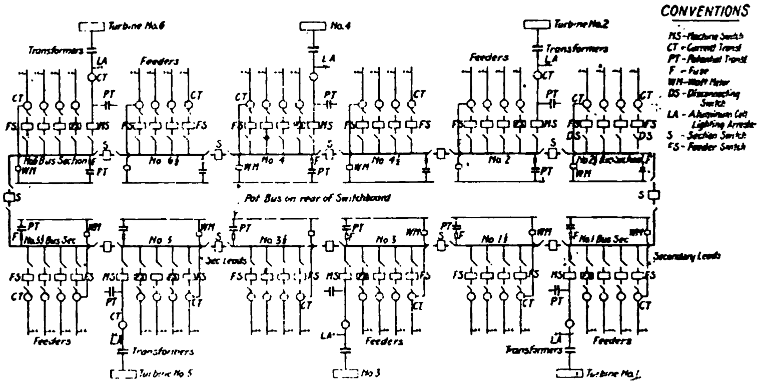

Fig. 12 Layout of the Complete Electrical System

the steam apparatus. Ultimately it is drawn into the generator, picks up the heat representing the losses therein, and is discharged through a duct below the turbine room floor. This duct carries the heated air into the pipe gallery in which the stoker blowers are located.

52 The electrical end of the plant also contains several unusual features. A semi-diagrammatic layout of the complete electrical system is given in Fig. 12.

NOTE. Since the presentation of this paper the Connors Creek plant has been put in operation and thoroughly tested out. Small troubles, such as would be expected with a new and radical design of this sort have, of course, been met, but, on the whole, the plant has operated very smoothly indeed.

The operating record shows that the thermal efficiency of the plant will be about as high as was expected when it was laid down. The coal consumption per kilowatt-hour during the first two months of operation was considerably lower than the record performance of the Delray plant, and it is natural to expect that it will be further decreased as the operating methods are improved.

No. 1520

A NOVEL METHOD OF HANDLING BOILERS TO PREVENT CORROSION AND SCALE

By Allen H. Babcock,[1] San Francisco, Cal.
Non-Member

There will be described in detail a radical departure from the conventional methods of handling boiler troubles due to corrosion. At present the method considered most highly scientific is to send samples of water to a chemist in order that a proper corrective may be prescribed. The composition of the corrective is usually a trade secret, jealously guarded, which the purchaser uses blindfold. On the other hand, the method to be described applies one corrective to all waters. A different composition of feed water demands not so much a different composition of corrective as a different quantity thereof. A simple chemical test, made by the engineer of the plant, or his clerk, gives the condition of the water *in the boiler, under steaming conditions*, and determines the quantity of chemicals to be applied. The purchaser now works with his eyes open.

2 It seems from all that can be ascertained at the present time, that at least there is a method of handling boilers, with a compound applicable to any water under any conditions of steaming, without a material change in the formula. The only requirement is that the boilers must be treated individually; it is not enough to treat a battery, or to treat the feedwater going into the boilers in general.

3 The first application by the author of this method was made at the Fruitvale power station of the Southern Pacific Company, the boiler equipment of which consists of twelve water tube boilers, each of 645 boiler h.p. It has since been used by the Southern Pacific Company in stationary plants in the oil fields and on locomotives in the same district.

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Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the San Francisco Section.

4 The method is in no respect original with the author. It has been his good fortune to be able to apply the work of another who was similarly situated, but who solved the problem for himself. The results have been such that it seems worth while putting them on record, particularly since a description of the method has lain in the technical literature of this country unnoticed for about three years. Apparently it has attracted almost no attention outside of a very limited circle.

5 The first inspection of the boilers of the Fruitvale plant after it was put in operation showed that serious corrosion began a very short time after the plant was started, although the boilers were not on regular load. The ordinary remedial measures produced no results at all. Bearing in mind that the plant was put in service to haul trains in the fall of 1911, it is significant that by June, 1912, the feed elements of the boilers were all in serious condition. With a total of 3360 tubes in all boilers, 252 tubes were requisitioned for replacement during practically the first eight months of operation. By the end of July, 1912, the trouble was extending to the second row of tubes, and 250 more tubes were requisitioned to take care of the trouble. In September, inspection showed that the drums were being attacked. In October, Boilers Nos. 1 and 3, after having been equipped with new tubes and placed in service in August, and worked entirely with water treated by the lime process, were found to be pitted worse than ever, and this was after the first serious effort to treat the water had been put into effect. In November, 1912, the piping was modified according to a plan sent by the manufacturers of the boiler, but in December of that year the inspection showed the pitting to be very severe, and in January 1913, 300 more tubes were requisitioned. This made a total of 1050 tubes in a year and a half, not counting about 90 put in by the manufacturers.

6 About this time began the use of a compound furnished by one of the best known companies in the business. The result was apparently to transfer some of the trouble from the tubes to another part of the boiler where there had been no trouble before. A sediment of black mud formed a scale which caused blisters and tube failures at those points. In June, inspection showed bad corrosion and pitting in all parts of the boiler, and 960 tubes were requisitioned. By October, 1913, the total work orders for boiler repairs on that plant amounted to $16,000 or nearly 12 per cent of the first cost of the boiler plant. Three different special formulae for compounds submitted by the same chemical company had been used, but as

things were going from bad to worse the use of the special compounds was discontinued.

7 In the fall of 1913, with a fine steam plant in a very critical condition, 1160 tubes out of 3360 had been replaced and there was no end in sight. A new expert was called in, who began by suggesting the conventional methods of zincs and paints, and that perhaps electrolysis was to blame, although everything had been kept carefully insulated. The result was no better than before.

THE NAVY STANDARD BOILER COMPOUND

8 About that time the author was handed the August 1912 number of the Journal of The American Society of Naval Engineers, which gave an account, by Frank Lyon, Lieutenant-Commander U. S. Navy, of what seemed to be a highly significant development in boiler trouble treatment.[1] In the course of his article, Commander Lyon said, "The writer has no hesitancy in saying that any boiler, using any water, can be kept from corrosion for any length of time." Having been brought up, to understand that whatever a Naval officer says is so, without qualification, the salvation of the Fruitvale boiler plant appeared to be within reach.

9 The result of Commander Lyon's investigations was the development of the Navy Standard Boiler Compound as described in Appendix 1 of the present paper. This compound is composed of sodium carbonate, trisodium phosphate, starch and tannic acid. The sodium carbonate takes care of any chemical reactions and renders the solution non-corrosive. The tannic acid and starch are added to prevent the formation of scale, the action being to hold the impurities in suspension in a colloidal state. The trisodium phosphate prevents the rise of the surface tension of the solution and consequent priming caused by the impurities in the water and by the application of the other ingredients in the compound. In using this compound to prevent corrosion a sufficient quantity must be added to each boiler to render the alkaline strength of the water in the boiler 3 per cent of normal or above.

10 The compound may be purchased from the Government Contractor for about one-half the cost of the ordinary compounds. Various proprietary companies also supply essentially the same material under their own trade names, but at about ten times the price of the real article.

[1]Navy Department, Washington, D. C. $1.30 postpaid. Reprints of the article are also available.

USE OF THE NAVY COMPOUND IN THE BOILERS OF THE FRUITVALE STATION

11 On the 4th of March, 1914, the first boiler was placed under treatment with the Navy Standard Compound. By the 26th of March it was in use in all of the boilers. As illustrating the results, the record of tube replacements month by month is most significant. In 1912, 690 tubes were replaced; 858 in 1913. In January and February of 1914 there were 229 tubes replaced in the two months, or at the rate of 1374 a year. The treatment, as stated above, was begun on the 4th of March, and was in all the boilers on the 26th. Thence the record stands: In February, 171 tubes replaced; March, 24; April, 15; May, 2; and from May 1914 to June 1915, (date of writing), not one has been replaced for failures. There have been 13 tubes replaced since May 1914, but they have all been replaced because they had to be taken out to get at something else, or by reason of obvious defects.

12 As to the money saved by the introduction of this method: In 1912 it cost in labor and material $5,790; in 1913, $10,256; in the first six months of 1914, it was $1,579, and since then the expense will hardly come to a hundred dollars a month for everything considered, that is, the ordinary wear and tear of the plant, washing of boilers, etc. In addition to this, extra labor in the plant, (made necessary by the failing tubes), to the extent of $181 a month, has been discharged. The treatment as practiced there costs about $80 a month, depending upon the amount of compound used. There is no scale in these boilers; they are absolutely clean. Below the water line they show clean black iron; above, a thin coating like whitewash.

USE OF THE NAVY COMPOUND ON LOCOMOTIVES

13 The first two months' operation at the Fruitvale power station with the Navy compound in use showed such very satisfactory results that the management ordered a trial of the method on probably one of the worst locomotive water districts in the company's service, namely, that part of the San Joaquin Division between Bakersfield and Mojave, known generally as the Tehachapi Pass. Mr. R. S. Twogood, assistant engineer in the office of the consulting electrical engineer, was detailed on this work and for nearly a year he has given practically his entire time to this demonstration. The following is an abstract of his contribution to the discussion at the December (1914) Meeting of the San Francisco

Section of the Society, when this subject came up for a general discussion.

14 On July 21, 1915, a compound test was started on freight Mallet Locomotive No. 4031 operating over the Tehachapi Pass. The locomotive was just out of shop. It had a new set of flues, was free from scale, with the exception of a thin deposit on the crown sheet and crown and stay bolts that were not removed during back shopping. The engine was left in chain-gang service.

15 At first an effort was made to carry the alkalinity at 3 per cent, but so far the best results have been from 0.5 per cent to 0.7 per cent normal alkalinity. Several things in the design of the locomotive, especially the Mallet, contribute to this result: for example, a very small steam space and steam dome; no perforated dry pipe to collect steam from all parts of the boiler; constant surging of the water due to the motion of the locomotive; sudden opening of the throttle and consequent great demand for steam, which reduces the pressure on the surface of the water under the throttle and causes it to rise and pass over into the steam pipes. Also, the water used for locomotive purposes in this district contains a large amount of solid matter which would tend to cause priming.

16 One point constantly in mind was to find some treatment in which one man could be made responsible, without having to depend on the engineers and firemen. This means that the boiler must be treated at engine turning points.

17 At first the chemicals (enough for the entire trip), were put into the tender tank, thence to pass with the feed water through the injector, feed-water heater and into the boiler. By this method the desired per cent of normal alkalinity in the boiler was not reached until one tank of water (10,000 gal.) had been used. Also, considering the fact that the tank capacity is much greater than the boiler capacity, the tank alkalinity was very low. This low alkalinity, though strong enough to bring out some solid matter, did not furnish sufficient tannic acid and starch with the standard proportion of chemicals to prevent this solid matter from forming scale. In the injector, where the cold water came in contact with the steam, and later where the warm feed water came in contact with the hot boiler water, some scale was formed. After this was noticed no more compound was added to the tank, but was pumped, in a concentrated solution, through the injector. This gave at once the desired per cent of normal alkalinity in the boiler. There has been no further trouble with scale.

18 The water from Bakersfield to Mojave averages about 3 lb. of scale-forming matter per thousand gallons. A Mallet locomotive in a helper trip from Bakersfield to the Summit and return will use from 30,000 to 35,000 gal. of water, which means about 100 lb. of scale per helper trip. With the present method of treating, 20 lb. of compound are pumped into the boiler before leaving Bakersfield yards. This gives an alkalinity of about 0.6 per cent, and as this percentage remains approximately constant between engine turning points, it is possible to treat the boiler for an entire trip before it leaves the roundhouse.

19 If too little compound is used the precipitate is fine and does not settle readily; but when the amount is sufficient the precipitate is large and flaky and settles very rapidly. This makes it easy to clear the boiler of some of the solid matter by blowing down after the engine has been standing on the siding a few minutes awaiting passing trains.

20 Present practice is to wash the boiler every second trip. A large percentage of the solid matter runs out with the water while draining the boiler and the rest is easily washed out. The solid matter has no tendency to cake, but will remain indefinitely in the form of a soft brown mud. It must be borne in mind that the frequency of washing is a function of the amount of solid matter in the water and the amount of water used.

21 Prevention of corrosion and scale formation was the main endeavor and yet another result was obtained, just as important. Absolutely no boiler troubles were encountered during the first four months of the test, and then the record was broken by only a minor leak, a record that gives some idea of the importance of keeping the boiler free from scale and thus prevent the high and uneven temperatures in the steel which cause the opening up of joints.

22 It is standard practice to carry from 1000 to 1500 lb. of sand on a Mallet locomotive to cut the soot from the inside of the flues. Why is it not just as important, for good steaming, to use some method to keep the outside of the flues free from scale?

23 The ingredients of the Navy compound originally were proportioned for use in stationary and marine boilers, and at an alkalin[illegible] strength of 3 per cent normal. The 10 per cent of trisodium phosphate was added to prevent light water. The locomotive type o[illegible] boiler, in general, shows a greater tendency to foam than other types of boilers and should, therefore, have a stronger non-foaming reagent. In cutting down the alkalinity from 3 per cent of normal to less than

1 per cent, the amount of trisodium phosphate has been greatly decreased with good results. The amount required seems to vary so much with different runs and conditions that as yet no decision can be made as to a definite amount for general use.

24 A new water was obtained at Bakersfield in January that is a great deal better than the old, as far as content of incrustating matter is concerned, but it is very light. Foaming and priming became so bad in one of the passenger locomotives that the water in the glass rose as much as 8 in. when the throttle was open. No compound of any kind had been used in this boiler. Six pounds of trisodium phosphate (no compound) was then put in the boiler through the injector just before the engine left the roundhouse for each trip. There has been no foaming on this locomotive since.

25 During the early part of the test on Mallet No. 4031 some light water and foaming were experienced. It was feared at that time that water, passing over, would loosen superheater units and high-pressure steam piping, and would cut cylinder bushings and packing, but these fears proved to be ungrounded. The engine has made more miles between shoppings for boiler repairs than any other Mallet operating out of Bakersfield.

26 On this division where the water is poor, practically all of the locomotives are back-shopped on account of the boiler and not the machinery. The present indications are that the use of this compound will make the life of the boiler longer than that of the machinery.

27 During the last month the use of this compound has been extended to all freight Mallets (seven) operating out of Bakersfield. Some of these had been in service for several months and large quantities of scale had formed. The cutting of this scale made very dirty boilers, causing frequent blowing and boiler washing. After two or three weeks, scale from $\frac{1}{8}$ in. to $\frac{3}{8}$ in. thick came off in large quantities, which proves that the compound will cut the old scale, but this action causes such dirty boilers that such procedure cannot be recommended. By the practice outlined above, a clean boiler can be kept clean and operated with success, using water classed by boiler people as very poor.

USE OF THE NAVY COMPOUND IN STATIONS OF THE ASSOCIATED PIPE LINE COMPANY

28 The results in the Fruitvale power station boilers caused prompt steps to be taken to extend the use of the compound to other

stations either operated or controlled by the Southern Pacific Company.

29 The Associated Pipe Line Company operates an oil pipe line from the oil fields in the southern end of the San Joaquin Valley to tidewater near San Francisco. On this pipe line are 30 steam pumping stations equipped, some with three and some with four 250-h.p. water-tube boilers. The feed water is of practically every known grade from good to very poor, and nearly every known type of boiler compound has been used from time to time in the worst of these stations. Mr. E. B. Partridge, Jr., superintendent of the pumping plants in the fields, came to Fruitvale power station for a study of the method. He began his investigation in the most skeptical frame of mind and he introduced the compound first into a station where, as he said, it could do the least harm. Eight months later, at the December Meeting of the San Francisco Section of the Society, where the subject was discussed, he gave an account of his experience which is here briefly summarized: In the thirty different pumping stations there were thirty different kinds of water, with a solid content running from 6 to 230 gr. per gallon. In the first station, a three months' trial of the compound showed that a turbine tube cleaner was no longer necessary, because the boiler was readily washed out and cleaning would not be required oftener than every three months.

30 These results were confirmed at the next station where the compound was tried. It was then introduced in a station which most of the feed-water experts in the State had turned down as impossible. The water carries 230 grains to the gallon. In these boilers 2½-in. extra-heavy boiler blow-off nipples corroded away in from three to four months and 3-in. extra-heavy feed-water lines pitted through in the same length of time. Nearly every known variety of boiler trouble had been experienced at this plant, but the Navy compound eliminated corrosion and precipitated the solids in a soft slushy form easily washed out. Owing to the unusually large quantity of solid matter in the water, however, it was necessary to blow the boilers every hour and to clean every ten days.

31 At another station where corrosion very similar to that in the Fruitvale boilers had been experienced from the start and all kinds of compounds had failed to relieve the trouble, the Navy compound was introduced. Up to that time they had used No. 9 gage seamless tubes in the bottom boiler rows, and No. 10 gage special spellerized tubes in the upper rows, but none of them lasted more

than three months. Feed lines, blow-off pipe lines, boiler drums and headers, together with the tubes, were all affected by corrosion; but the Navy compound straightened out all their trouble in this station precisely as in all the others. In order to have further verification pieces of new tubes were hung in these boilers and after three months they showed not the slightest signs of corrosion.

32 In general, Mr. Partridge found that when he commenced to use the compound there were a great number of tubes which had been badly pitted and that these failed gradually, but they held up surprisingly well, considering their condition. Only a very few of these old tubes have been lost and none that were put in just prior to the introduction of the Navy compound treatment. He found that to get results it is necessary to work intelligently and not at all in the old haphazard fashion. He detailed an inspector on the work who travels from station to station, not only to see that the compound is being properly used and in sufficient quantity, but also to instruct engineers and to quiz them as to the results they obtain.

33 At every pumping plant a printed form is filled out and mailed to the general office every morning. From these Mr. Partridge keeps a close supervision over all of the plants. The reports show how much compound is used; what percentage of alkalinity is carried; how often the boilers are blown, that is, a bottom or a surface blow; when any particular boiler is cut in or cut out, and why; and the condition of the boiler when it is opened for cleaning. The importance of these results is best understood by engineers who know the operating conditions of an oil pipe line, where all the pumping plants are in series from one end of the line to the other, and a delay or a stoppage at any one station affects the output of the entire line, and consequently, the earning capacity of the whole property.

FACTS REGARDING THE COMPOUND AND BOILER FAILURES

34 The use of the Navy compound was discussed in a preliminary way at a meeting[1] of the San Francisco Section of the Society after which it was suggested to the author that it would be well to go slow in using the compound, for the reason that some vessels had experienced such serious damage to their boilers from it as to cast discredit upon this method of boiler treatment. The substance of the statement was that for some years evidence has accumulated

[1]December 8, 1914.

to show that under certain circumstances the use of sodium carbonate or sodium hydrate in boilers makes the steel brittle and causes failure of the joints by cracks; that at present it is impossible to state how or why the action occurs, but it is possible to name the circumstances under which it is likely to occur, the characteristics of the cracking when it does occur, and the principal cases that have come under direct notice.

35 Briefly, the theory advanced to the author as to the trouble is as follows: That sodium carbonate is always partially converted into sodium hydrate when in solution under boiler temperature, so that it makes no difference whether the carbonate is used in the beginning, or caustic soda, or both, as sodium hydrate will always be present sooner or later. When either of these substances is used for the treatment of scale, no trouble from the cracking of the steel is likely to occur, because they react with the scale and there is little or no excess hydrate. Where cracking has occurred it has always been true that free hydrate has been present in the boilers in the absence of sulphate; so that the statement is warranted that whenever sodium carbonate or sodium hydrate is introduced into a boiler in the absence of substances reacting with them, brittleness will result sooner or later. Cases have been known in which the brittleness approaches that of neat cement of the same thickness as the plate. It occurs only in the seam of the boiler, either due to the fact that the joints are at higher stresses here than in the rest of the plate, or that the joints are the only place in which this solution can concentrate, and it is known that unconcentrated solution does not attack steel at all. The cracks that result are distinguished from ordinary forms of cracking by the following characteristics.

36 They are more fully developed at the surfaces of the contact of the joints than they are at other surfaces, so that a crack at a rivet hole will often measure ¾ in. on the contact surface of a ½-in. plate without showing at all on the other surface; in other words, the surface of the crack itself is usually triangular, with the base of the triangle on the contact surface. They usually originate at the rivet holes and show scarcely any tendency to follow the lines of stress. Running from adjacent rivet holes, two cracks will often pass each other and after passing, join, leaving an island in the plate; that is, the cracks show no strong tendency to join each other. They change their direction sharply and often, to an extent of 90 deg.; they are always unaccompanied by any elongation of the plate;

and finally, they are always, of course, below the water line.

37 The statement made to the author gave also a number of cases of such defective plates. Among others, were named the boilers of the U. S. Torpedo Boat Destroyer *Aylwin,* wherein occurred a fracture of one drum, and cracking of one or more drums of three boilers out of four; also the U. S. Coast Guard Cutter *Unalga,* where a cracking of one drum out of two in a period of operation of about fourteen months was noted. The statement concluded by a note to the effect that in the case of the Coast Guard Service, an order had been issued discontinuing the use of the Navy compound.

38 Because the treatment, at least in this form, originated in the Navy Department, because the *Aylwin's* boilers were cited among others as having failed from this cause, and because the Navy boilers are operated in general with water that is practically free from sulphates, the statements with reference to Navy experience were investigated as fully as possible, and the following facts were developed:

39 The use of Navy compound has not been discontinued in the Navy, but the compound is used ordinarily only in sufficient quantity to maintain the water in the boiler at a low alkaline strength, and not above ½ of 1 per cent normal, the point at which corrosion changes in character from general corrosion to local corrosion or pitting. In a steaming boiler no difficulty is experienced in maintaining the alkaline strength of the water at or above 3 per cent normal; but Naval boilers are idle during a large part of the time, and owing to the fact that the compound does not remain in solution in idle boilers, but settles to the bottom, the alkaline strength of the water in the upper part of the boiler frequently falls between the limits of 1 per cent and 2.5 per cent of normal. As is well known, the effect of this alkaline strength is to promote corrosion in its most dangerous form—local corrosion or pitting.

40 The early experiments with the boiler compound determined the facts that good steel will corrode almost evenly over its entire wetted surface when placed in distilled water, and that as boiler compound is added to give the water a low alkaline strength, the rate of corrosion is somewhat decreased, but the character of the corrosion remains the same. Therefore, low alkalinity inhibits all but a light general corrosion, which is so small in amount that its effect may be disregarded for practical purposes.

41 Further, since the Navy uses distilled water to a large extent, only small quantities of scale-forming substances are ever present in the boilers, and hence large amounts of boiler compound are not required to prevent the formation of scale.

42 In the case of the *Aylwin's* boilers the Naval Engineering Experiment Station conducted an extensive investigation of the rupture of the drum, and authority was granted for a representative of the manufacturer to be present during the tests. The conclusions of this investigation were that no experimental or other evidence indicated that the failure of the *Aylwin's* boiler was due to the use of boiler compound containing sodium carbonate or sodium hydrate.

43 It is needless to say that any condition of design or construction that causes leaky seams which can be made tight only by excessive caulking might possibly produce brittleness in the metal of the plate.

44 The White-Forster boiler installed in the *Aylwin* class has D-shaped water drums, similar to the Yarrow type. In practice, it was found impossible to keep the seams of the White-Forster boiler tight, owing to the tendency of the drum to assume a circular shape under pressure. The manufacturers of this boiler have since corrected this by placing struts in these drums across the long diameter. It is also understood that similar trouble has not been experienced with Yarrow boilers.

45 With reference to the *Unalga* as another vessel having had trouble of a similar nature, not enough is known at the present time upon which to venture a statement as to the cause of failure. It is known, however, that the cracks in the steam drum plates of these boilers are similar to those observed in the *Aylwin's* lower drum plates; that the manufacturer of the boiler has conducted tests of the material, and that at the suggestion of the company the Naval Engineering Experiment Station will soon begin an independent examination and investigation of the material with a view to determining the cause of the peculiar conditions existing. It has also been learned that boiler compound in considerable quantity was used in this boiler, and that another vessel with a duplicate boiler, but in which only small quantities of the compound were used, has not had trouble.

SUPPLEMENT

BOILER WATER TREATMENT

By Fred E. Geibel, Assistant to Mr. Babcock

46 The water used for boiler purposes at the Fruitvale Power Station comes from two bored wells driven on the inside of the station near the north wall. The chemical analysis of the water is as follows:

Silica	0.932 grains per gal.
Oxides, iron and aluminum	0.245
Carbonate lime	3.023
Nitrate lime	2.624
Sulphate lime	1.082
Carbonate magnesium	5.667
Chloride magnesium	1.769
Sodium and potassium chlorides	5.863
Loss, etc	0.169
	21.374
Total incrustating solids	15.342 or 2.19 lb. per thousand gal.

47 This water is ordinarily classed "Fair" and the first step taken to prevent corrosion was the installation of a water-treating plant. The water received the ordinary lime and soda ash treatment, being allowed to stand from 10 to 16 hours before being used. The scale-forming solids, however, did not settle out as was evidenced by scale forming in the transformer cooling coils, the feed-water heater and in the tube elements. Chemically, though, the treated water was pronounced excellent and was not considered the cause of the corrosion. Several boiler compounds were tried out with the treated water with no better results.

48 In the original installation the feed-water was introduced into the upper or economizer section of each boiler, having been partially heated before being discharged into the section. Operating in this way, it was practically impossible to keep tubes in the upper sections.

49 About September 1913 the feed-water connections were removed from the economizer sections and the water fed directly into the drums at the front end, the feed pipe being run into the drum for a distance of about four feet. Two months' operation in this way showed serious corrosion in the bottom of the drums and practically the entire destruction of the internal portions of the feed pipe.

50 At this point the matter was brought up as being a case of corrosion by stray currents from the railway system. To detect the presence of any such currents in the boiler, a copper terminal or electrode was placed through the shell of the boiler into the water, but insulated from the shell. A recording milli-voltmeter connected between this terminal and the boiler shell or feed-water connection would indicate the presence of a potential difference, and it was expected that if the curve drawn by the voltmeter followed approximately the load curve of the station, it would show that the cause of corrosion was undoubtedly stray electric current from the railway system.

51 It was found that a very small difference of potential did exist, but the curve drawn by the meter, instead of following the load curve of the station, was directly dependent on the boiler taking feed-water. The voltmeter reached a maximum value which was constant as long as the boiler was being fed. Closing the feed-valve reduced the potential to practically zero, but as soon as it was opened the voltmeter would again indicate the maximum value. This led to the conclusion that the potential difference was due either to the temperature or the chemical characteristics of the feed-water. Other experiments, therefore, were started with a copper-iron couple in samples of water from the wells, before and after treating, and from the boilers, condensers and hot-wells. Taking into account the varying contact-resistance between the electrodes and the water, the potential differences obtained with the several samples did not differ materially. With all the samples, however, the maximum difference of potential was obtained at temperatures between 170 and 190 deg. fahr.

52 While Mr. Babcock had seen the paper by Mr. Lyon describing similar tests and was watching results, he purposely allowed these tests to be carried on in ignorance of the paper. At the completion of these electrical tests the results were found to be practically identical with those described by Mr. Lyon, which gave an

unbiased check on the electrolytic theory of the corrosion of iron and steel accepted by Mr. Lyon.

53 This theory is now accepted by many leading physicists and embraces the fact that in the presence of a solvent the iron goes into solution as a hydrate before oxidizing. Considering that water is a universal solvent every metal has an inherent tendency to dissolve in water or water solutions. This tendency is called the solution tension of the metal. Opposing this tendency to dissolve is a pressure in the solution tending to resist the entrance into the solution of any more of the metal. This opposing pressure is known as the osmotic pressure of the solution. Accepting this theory, it is only necessary, in order to prevent corrosion, to raise the osmotic pressure of the solution, or electrolyte, above the solution tension of the metal. In the case of boilers, the osmotic pressure of the electrolyte, or boiler water, is raised by the addition of alkaline salts. The osmotic pressure being dependent on the concentration of the salts, the corrosive condition of the electrolyte is indicated by its alkaline strength. The alkaline strength to be carried in the boiler depends on the salts used.

54 After a few simple chemical tests on the Fruitvale boiler-waters and an investigation of the use of the Navy Standard Boiler Compound as described by Mr. Lyon, it was decided to adopt the treatment at Fruitvale. In starting the treatment the treating plant was discontinued and the raw well water was used. A quantity of the compound was added to the hot-well continuously just sufficient to take care of the chemical reactions necessary to render the feed-water neutral, or only slightly alkaline. One boiler at a time was boiled out with a strong solution of the compound. In boiling-out, the boilers were filled nearly full with 10 per cent normal alkaline solution of the compound and a pressure of 10 lb. gage was kept up for 36 hours. This boiling-out loosened up and brought out as a sludge practically all of the old hard scale and rust in the boiler. Formerly, this scale at regular cleanings was removed but only by chip-hammering methods. The boiler was then thoroughly washed out with clean water and put in service with sufficient compound in the boiler to render the water 3 per cent normal alkalinity.

55 The last boilers to be put in service at the high concentration were not boiled out as it was found that the small excess of compound that had been added to the hot-well had practically cleaned these boilers in ordinary operation.

56 Before this time, the boilers were regularly blown-off once

on the morning watch. However, after putting the boilers in service at the 3 per cent concentration they were not blown off until the concentration of salt ran too high. The boilers at Fruitvale have been run from four to five days before blowing-off. The salt or sodium chloride content has run as high as 2500 grains to the gallon. This high concentration of salt is to be avoided as, if for any reason the boiler should run below 3 per cent alkaline, serious corrosion would result. The presence of so much salt in the boilers at Fruitvale is due to there being some salt in the well water, and with the varying load carried it has seemed impossible to keep the condensers free from some small leaks. The circulating water for the condenser is from the tidal canal and is consequently salty. Recent changes in the condensers, however, have reduced the salt content to a very low degree.

57 In this method of boiler treatment it will be noted that each individual boiler must be considered and the water in each boiler be kept above the prescribed concentration. At Fruitvale Power Station samples of the boiler waters are taken each morning. Simple tests are made to determine the alkalinity of the water and chlorine or salt content. Any boilers low in alkalinity are treated with sufficient compound, pumped into the boiler, to bring the alkalinity up to the required point. Any boilers high in salt are blown down to rid them of the salt and then additional compound added to grind up the alkalinity, if necessary. The boilers are arranged with suitable connections and a portable pumping outfit has been provided for pumping compound directly into any particular boiler.

58 Shortly after beginning the high alkalinity in the boilers serious trouble arose with the man-hole gaskets blowing out. It was thought that the strong alkaline solution was eating out the gaskets which were of the asbestos type. It soon developed, however, that not only were the man-hole covers slightly warped, but the surfaces next to the gaskets were coated in spots with iron rust. The action of the compound was to eat away this rust and a consequent leak started, resulting in the eroding and blowing away of the gasket. The truing up of the man-hole covers and cleaning off the surfaces has entirely eliminated this trouble. This action may also be noticed in old boilers where the seams are rusty. With the first application of the compound the rust is eaten away and leaks are started.

59 There was also some trouble with the brass check-valves in the tube elements of the boilers corroding. This, however, was

expected as it was known that a brass-iron couple in a strong alkaline solution would result in the deterioration of the brass. At the time the treatment was begun malleable check-valves had been ordered and have since replaced the brass ones.

60 In the above described method of boiler-water treatment the amount of compound consumed per day is merely the amount required to take care of the chemical reactions in the make-up water, the water in the boilers having once been made the proper alkaline strength. If too little compound is used per day the alkalinity of the boilers will fall off, and if too much is used the alkalinity will gradually build up, as after the chemical reactions are taken care of any excess of compound begins to raise the alkalinity. Blowing-off reduces the alkalinity of the boiler as some of the excess compound is blown out. This must be made up by the addition of more compound. Therefore, boilers should be blown off as little as possible and the frequency is dependent on the amount of impurities in the make-up water. The present practice at Fruitvale is a short blow every other day.

61 One important difference in this method of boiler-water treatment and the commonly accepted theory as regards boiler waters is the high alkalinity carried in the boiler. It has been commonly considered that a boiler water high in alkalinity, or sodium content, has a decided tendency to prime. This is probably true if no anti-priming substance, such as trisodium phosphate, is added. However, while it was not known or expected by the boiler attendants, tests on the boiler-waters at some plants have shown as high as 22 per cent normal alkalinity. With this Navy Standard Boiler Compound it has been found that the tendency to prime is more a function of the solid matter in suspension rather than the per cent alkalinity of the water. The boilers at Fruitvale have frequently been run at 10 per cent normal alkalinity without any tendency whatever to prime.

62 The Fruitvale boilers have been run with absolutely no corrosion for more than a year under the high alkalinity treatment. Recently, owing to reports of damage to boiler steel due to concentrated sodium carbonate or hydrate solutions, the alkalinity was reduced to less than 0.5 per cent and kept at that point for a month or more. While it was not definitely proven that corrosion did again set in, there were some indications of it; and since the boilers were in a critical condition before the Navy compound was first used, it was decided to return to the high alkalinity. Before increasing the

alkalinity, the boilers were carefully examined for any defects that could be attributed to the high alkalinity treatment but absolutely none were discovered.

63 The case at Fruitvale is one of the most pronounced cases of localized corrosion on record. Such cases, differing from the ordinary cases of general corrosion, are found chiefly in boilers using a high percentage of condensate or distilled water. The design of a boiler probably has a noticeable effect on localized pitting, but it is very evident that the low osmotic pressure of distilled water is the direct cause of the corrosion and that the corrosion is electrolytic in its action. The proof of this is found in the fact that all other known methods of boiler water treatment failed to stop the corrosion at Fruitvale and also, after handling the situation for more than a year on the electrolytic theory, a return to the ordinary method of treatment gave indications of immediate trouble.

64 General corrosion, while electrolytic in its action, is generally attributed to the presence of some acid radical in the water and is found in boilers using raw water. The raw water may test slightly acid or may become so upon being heated. To stop this general corrosion it is necessary to add a sufficient quantity of alkaline salts to neutralize the acid radical. This is the ordinary method of treating boiler waters and the method has probably failed only in such cases where structural details were particularly favorable to localized pitting and the impurities in the water after being neutralized were not of the proper character or of sufficient quantity to bring the osmotic pressure to a point where pitting would stop. The Navy compound is being used with such waters but other salts in the boiler water which do not test alkaline in a clear sample must act to render it non-corrosive, as excellent results are being obtained at plants using raw water with alkalinities ranging from 0.2 to 2.5 per cent.

65 In the bad water districts, both on locomotive and stationary boilers, it has been more a problem of fighting the formation of scale rather than the corrosion. The impurities forming the scale in these waters are chiefly the carbonates and sulphates of lime and magnesium. In stationary plants undoubtedly the most efficient process is to bring down most of the scale, by the use of chemicals in an open type feed-water heater, and to supplement this treatment with a compound in the boiler. The Navy compound cannot be used for this outside treatment as the compound holds the scale in suspension. The most common chemical used for this purpose is

soda-ash. This outside hot treatment, of course, cannot be used with locomotives and is not always possible at stationary plants. In such cases it is necessary to handle all of the solids in the boiler. This is rather difficult in very bad waters, since to prevent scale the solids must be held in suspension and the concentration of these solids in suspension beyond a certain limit causes priming. This limit is not definitely fixed but depends upon the anti-foaming agents in the water or the compound.

66 The Associated Pipe Line is using successfully the Navy compound in 26 stations in the San Joaquin Valley, chiefly for the prevention of scale. Most of these stations have three 250-h.p. water tube boilers and use water containing solid matter varying in amounts from 5 to 231 grains per gallon. In places where the solid content is so very high it has been necessary to increase the quantity of anti-foaming chemicals above that ordinarily contained in the Navy compound. This has been done, not by changing the composition of the compound, but by adding separately a quantity of the non-foaming ingredient, which in this case is trisodium phosphate. Additional trisodium phosphate has also been necessary where the compound has been used on locomotives.

67 In handling the Navy compound a simple chemical outfit is placed in the hands of the engineer in charge of a station. A few minutes' use of this outfit each day gives him a positive indication of the condition and performance of each individual boiler. It shows at once whether all chemical reactions have been satisfied and also the condition of the boiler water with respect to scale-formation and corrosion. Records of the tests and of the compound used furnished the supervising engineer give him a ready check on the work of the power plant employes. These simple tests, described elsewhere in this paper, are easy to make, require but little time, and are practically a method of feeling the very heart-beat of the power station. They serve to relieve any feeling of uncertainty about the place and put the engineer in charge in a position to rely absolutely on his boilers.

APPENDIX 1

SYNOPSIS OF INVESTIGATIONS BY FRANK LYON, LIEUTENANT-COMMANDER U. S. NAVY[1]

68 From July 1896, to August, 1899, Commander Lyon was Assistant Engineer, U. S. S. Oregon and for the greater part of the time was in direct charge of her boilers. The feed water was kept in practically an open tank and was kept so stronglv alkaline as to render it unfit for ordinary purposes. In this vessel corrosion of boilers was almost unknown.

69 From May 1906, to May 1909, Mr. Lyon was Senior Engineer Officer of the U. S. S. New Jersey. Troubles from corrosion were experienced throughout the three years. The corrosion was most noticeable in the piping in which the water was heated more or less between its entrance to and exit from the system.

70 The Oregon had fire tubular boilers and the New Jersey Babcock and Wilcox boilers. The water in the latter was kept slightly alkaline, but owing to fear of priming in these small drum boilers, it was never kept as alkaline as that in the Oregon's boilers. It was felt that every effort to prevent corrosion and to keep the boilers clean had been made, yet destructive local corrosion was going on and increasing in effect in spite of the zincs, non-acidity of the water, cleaning and other efforts to stop it. The Navy regulations had been followed, and experiments made, yet Mr. Lyon had failed in every particular to stop corrosion of metals having their surfaces in contact with water, and he was detached from that duty and left, knowing that there was something woefully wrong in the methods he had pursued and with the general methods of treating corrosion on shipboard.

71 With this excellent grounding in a knowledge of the effects of corrosion, he began at the Naval Engineering Experiment Station an experimental investigation of the problem of preventing corrosion. Tests were made in distilled, sea and brackish waters, in diluted sea and brackish waters with jars open and closed; in steel pots, glass jars and in a boiler. Very soon after starting this investigation it was seen that in the untreated distilled or in fresh waters the specimens of steel corroded all over; that as the concentrations[2] increased, the rate of corrosion at first decreased slightly and then began to increase. In the concentration where the rate showed an increase there were always evidences of local corrosion or pitting, and as the concentrations increased the rate of loss increased to a maximum, then fell rapidly to zero and remained there, the areas of local corrosion becoming smaller and more pronounced until they disappeared altogether in the concentration in which the loss was zero.

72 The concentration whose strength is just below the one in which local corrosion or pitting first appears, he called the lower-limit concentration; the one in which the rate of corrosion is a maximum, the critical; and the one i

[1] The Journal of The American Society of Naval Engineers August, 1912. Navy Departm Washington, D. C. Price $1.30 postpaid. Reprints of this article are also available. Man the paragraphs above which refer to Mr. Lyon's work are abstracted liberally from his article.

[2] In this discussion, the concentrations mentioned are the alkaline salts in the water expresse percentages of "normal solution." Equal volumes of normal solutions will just satisfy a chem reaction completely without any surplus of reagents remaining on either side of the equation.

above the one in which the last signs of corrosion appear, the upper-limit one for the metal tested.

73 In general the fundamental fact was established that if the solution is kept sufficiently concentrated with sodium carbonate, corrosion is inhibited. That is the sum and substance of it.

74 Also, that the order of magnitude in which corrosion occurs in distilled-water solutions in the normal concentration of sodium nitrate, chloride and sulphate was in about the order of the strength of the acid radicals.

75 The upper and lower-limit concentrations in those solutions in which they are found, vary with the metals immersed, and with the conditions of the surface of the metal in contact with the solution, both with regard to impurities in it and to the physical treatment it has received.

76 The last two points are very important to remember as accounting for local corrosions; and also this vital point:

That the rate of loss in the critical concentration is greater than that in the untreated water. In other words, it is better not to treat the water at all than to treat it near the critical values, and this can be done very readily by pouring compound into a boiler without paying attention to what the concentration is in the boiler, a fact that demonstrates the necessity for the individual testing of the water in the individual boiler under steaming conditions.

77 The limit and critical concentrations of the following solutions for one grade of steel were determined very carefully. Expressed in percentages of normal strength in distilled water, they are:

SOLUTION	UPPER-LIMIT	CRITICAL	LOWER-LIMIT
Sodium carbonate (calcined)	2.6	0.8	0.16
Caustic soda	2.6	0.8	0.16
Lime	2.6	0.6	0.15

These are the three salts used extensively in boiler treatment. They have practically the same characteristics as regards solution pressure, which is the measure of the tendency to corrosion.

78 Further work along these lines with open jars showed that the concentration of lime solutions weakened very rapidly, while those in caustic soda, sodium carbonate, disodium phosphate and the chromates maintained or increased their concentration as the water evaporated. Solutions of lime water exposed to the air may be non-corrosive to steel one day and corrosive the next, due to the absorption of carbonic acid from the air, while if the upper-limit concentration of sodium carbonate is made, steel thrown in it will not corrode until the concentration is brought below the upper-limit by some external means.

79 The English of that is, use sodium carbonate if possible; do not use lime unless forced to, and when lime is used be careful not to work near the critical value.

80 Commander Lyon found that steel connected to copper by a good metallic conductor does not corrode in an upper-limit concentration of sodium carbonate, lime, caustic soda or disodium phosphate. In similar concentrations air, oxygen, carbonic acid gas, graphite, zinc oxide, mill scale, and other supposed exciters of corrosion have no effect upon steel or iron until the concentration is reduced below the upper-limit for that steel. Steel was suspended in a glass basin in, and above, the limit concentration of sodium carbonate, and pure oxygen was blown under it in such a way that sixty bubbles a minute impinged on the steel and passed up along its sides; this continued for eight days,

and no signs of corrosion were evident, and there were no losses of weight This was continued for fifteen days longer, using air instead of oxygen, with the same result.

81 Zinc, being of higher potential than iron, will corrode in an upper-limit concentration for iron or steel.

82 It was found that the upper-limit concentration for all irons and steels was about 2.6 per cent normal alkaline strength of sodium carbonate and of caustic soda solutions, in distilled and in sea-water. The highest concentration of any of these chemicals in which corrosion was found is 2.5 per cent of normal.

83 With pure feed water, sodium carbonate and trisodium phosphate, properly proportioned, will stop corrosion and priming if enough of the mixture is used. If impure water is used, then sodium carbonate, trisodium phosphate and cutch, (containing tannic acid), when used in the right amounts, will stop corrosion, prevent priming, and also prevent scale from forming unless the saturation of sludge gets too high. Such a mixture is the Navy Standard Boiler Compound; and if by the use of this compound, the water in the boiler is always kept at a concentration of or above 3 per cent normal alkaline strength, no corrosion will take place, no scale will form and the water will be no more likely to prime than it would be if it were untreated.

84 The ingredients of the Navy compound are intimately united by thorough digestion, dried, finely powdered, and well mixed. They are readily soluble in water. The compound must show on analysis at least 76 per cent of anhydrous sodium carbonate (Na_2CO_3), 10 per cent of trisodium phosphate ($Na_3PO_4.12H_2O$), 1 per cent of dextrine or starch, and sufficient cutch to yield at least 2 per cent of tannic acid, the remainder to consist of water and only such impurities as are common to the ingredients.

85 These investigations demonstrated why the boilers of the Oregon showed no signs of corrosion in three years, and why those of the New Jersey were considerably corroded in the same time. In the one the corrosion had been prevented by keeping the water strictly alkaline; in the other it had been materially aided by an insufficient degree of alkalinity. Mr. Lyon's conclusions are tersely expressed as follows:

86 "The writer has no hesitancy in saying that any boiler using any water can be kept free from corroding for any length of time if treated with soda, and if its concentration is maintained at or above 3 per cent normal alkaline strength. If the water is not to be kept sufficiently alkaline, it had better be kept neutral."—A statement that is absolutely borne out by the results of his treatment as applied at the Fruitvale power plant according to his method.

APPENDIX 2

DIRECTIONS FOR USING NAVY COMPOUND

87 The Navy compound is in the form of a dry powder. It is easily dissolved in warm water, but should be boiled with a steam jet for 30 minutes or more to digest the tannin thoroughly. Ordinarily this solution is fed continuously into the hot-well or feed-water line in just sufficient quantity to take care of the chemical reactions or to render the water neutral. When the boilers are

run at high alkalinity the excess of compound necessary to raise the alkalinity should be pumped directly into each boiler. In some instances where the water is bad there is a tendency for the sludge to choke the feed lines and feed pump and it is necessary to pump all of the compound into the boiler at intervals of 6 to 12 hours. The quantity pumped in at any one time must provide for the reactions until the next pumping.

88 Where injectors are used on the boilers the compound should not be fed into the water supply, but must be put into the boiler as above described. An injector, however, can be used for putting the concentrated compound solution into the boiler.

89 In cases such as the Fruitvale plant the practice has been to carry the water in the boiler at 3.0 per cent normal alkaline strength, or above. In plants where general corrosion or scale is found, the alkalinity of the boiler water has been held at a point at which it was found that no scale formed and no corrosion was evident. If in the necessary quantity of compound required for the above, the quantity of non-foaming chemicals was not sufficient to prevent light water with a reasonable amount of blowing-off, an additional quantity of trisodium has been added. Blowing-off, however, when properly done clears the boiler of a very great amount of the sludge and reduces the tendency to prime.

90 In the above methods of treating, the quantity of compound to be used in each case is determined by the alkalinity of the water. The test for alkalinity is very simple and can be performed by almost any one of the power plant employes. A knowledge of chemistry is not required but usually one learns to make the tests from reading the following instructions. The test for salt is not generally required except in plants using salt water for condensing purposes.

BOILER WATER TESTS

91 In general, tests of water show it to be alkaline, neutral or acid. An acid substance neutralizes an alkaline substance and vice versa. Therefore, to determine the degree of alkalinity or acidity of a water the opposite substance is added until the neutral point is reached. The quantity of substance added to bring about the state of neutrality denotes, when reduced to proper terms, the degree of alkalinity or acidity as the case may be.

92 *Indicators.* The state of alkalinity, neutrality or acidity is determined by the use of solutions which serve as indicators. A few drops of indicator No. 1 added to an alkaline solution will turn it a deep red. If added to an acid solution the color will be pink. If the solution is exactly neutral the color will be a faint yellow.

93 *Test Outfit.* The standard Boiler Water Test Outfit contains two standard solutions numbered No. 1 and No. 2; two indicators similarly numbered; one small bottle of soda solution; one dish; one stirring rod; one 50 cc. measuring flask, and two burettes numbered No. 1 and No. 2. The burettes are graduated from 0 to 25 cc. in 0.1 cc. and are used to determine the quantity of standard solution used.

94 *Alkalinity Test.* Draw off a sample of water from the boiler into a vessel which has just previously been washed out with water from the same boiler. Allow this water to cool to about room temperature. Fill burette No. 1 with solution No. 1. Open pet cock at bottom of burette and draw off a few drops of the solution until all air bubbles have been expelled from lower end of burette

and it is filled to tip with solution when cock is closed. Measure off 50 cc. of the sample of boiler water, using the 50 cc. flask, and pour it into dish. Both the flask and dish should be just previously washed out with other water from the same sample. The sample used should be clear, and contain only such impurities as remain in solution. If the suspended matter does not readily settle out the sample should be filtered.

95 Add four drops of indicator No. 1 into the sample in the dish and the color of the sample should turn immediately to a deep red. Now read the graduation at the top of the solution in the burette No. 1; then from pet cock at the bottom drop solution No. 1 into the sample in the dish, stirring continuously with the glass rod, until sample turns clear, then yellow, then a very faint pink. Close the pet cock and read the graduation on the burette at the top of the solution. The difference between the two readings indicates the number of cc. of solution No. 1 required to neutralize the alkali in the sample. The strength of solution is so proportioned that the number of cc. of solution used to neutralize the sample represents directly the degree of alkalinity in per cent of a standard normal alkaline solution. Thus if 2.3 cc. of solution is used, the alkalinity of the sample is said to be 2.3 per cent.

96 Note: The above instructions require that the solution be added until the sample turns a faint pink. This indicates that the alkali has been a little more than neutralized and the sample has become slightly acid. The error, however, will be negligible for the purpose of testing boiler water if the test is carefully made and the pet cock closed at the instant that the faintest pink color is attained in the well-stirred sample. After the sample becomes clear the solution should be added drop by drop and the test conducted in a good light.

97 *Chlorine Test.* Fill burette No. 2 with solution No. 2. Using the same sample used for the alkalinity test, add a drop or two of the soda solution to bring the sample back to neutrality or slightly alkaline. Add four drops of indicator No. 2. Now drop in solution from burette No. 2, stirring continuously, until the sample turns from yellow to a reddish-yellow color throughout. The number of cc. of solution used multiplied by 10 represents the number of grains of chlorine per gallon of water. That is, if 2.2 cc. of solution is used it means that the water contains 22 grains of chlorine per gallon. If the solution is added until the sample is a deep red, erroneous results will be derived. The stop cock on burette should be closed as soon as the change in color from yellow to reddish-yellow occurs in the well-stirred sample.

98 *Chlorine Content.* Where boilers are blown down regularly the chlorine test may be omitted, as the amount of chlorine does not reach a dangerous point. At plants running condensing the chlorine should never be allowed to run higher than 500 grains per gallon. Blowing-off decreases the chlorine content.

99 *Caution.* Keep dish, measuring flask and stirring rod clean. Never pour standard solutions from the burette back into the bottles. When a fresh supply of standard solution is received pour out all of old solution and rinse bottles and burettes several times with distilled water. On reading height of liquids in flask or burettes always read from bottom of meniscus.

BOILER WATER TEST CHEMICAL

100 *Soda Solution.* The soda solution is an N/2 normal solution of caustic soda made by dissolving 20 grams pure caustic soda ($NaOH$) in distilled water to make 1000 cc. of solution.

101 *Solution No.* 1 is an N/2 normal solution of sulphuric acid. Take about 950 cc. of distilled water and to it add slowly 24.5 grams of c.p. sulphuric acid (H_2SO_4) or 13.3 cc. of acid of 1.84 specific gravity. (Never add water to the acid). To the solution now add sufficient distilled water to make 1000 cc. of solution. To determine the exact acid strength of the solution it should be checked against the soda solution. A given quantity of the acid solution should exactly neutralize an equal quantity of the soda solution.

102 *Solution No.* 2 is a solution of nitrate of silver made by dissolving 41.01 grams of silver nitrate ($AgNO_3$) crystals in distilled water to make 1000 cc. of solution. Such solution when used with a 50 cc. sample of water represents for each cc. of solution used 10 grains of chlorine per gallon of water. If it is desired that the results be in terms of grains of sodium chloride (common salt), the solution should be mixed 24.85 grams silver nitrate per 1000 cc. of solution.

103 *Indicator No.* 1 is the neutrality indicator and is made by dissolving 1 gram of methyl orange powder and 5 grams of phenolphthalein powder in 500 cc. of alcohol and adding enough distilled water to make 1000 cc. of solution.

104 *Indicator No.* 2 is for the chlorine or salt test and is made by dissolving a gram potassium chromate (K_2CrO_4) in distilled water to make 100 cc. of solution.

DISCUSSION

Walter M. McFarland. It is unpleasant to have to say that a number of the statements in this paper (as relating to marine boilers) are inaccurate.

It is stated that the White-Forster boilers were of such design that it was impossible to keep the joints tight ordinarily. That statement is incorrect. The company with which I am connected has built all the White-Forster boilers constructed in this country, and I know of a large number used in the British Navy; and the boilers have given satisfaction generally.

A great deal might be said about this whole soda question, which is a very complicated one. I do not propose to go into it at length now, but some facts should be brought out. The *Unalga* was mentioned as having had boiler troubles, and it was said it was not known whether the trouble was due to soda or not, although it was mentioned there was some difference between the conditions on this vessel and a similar vessel.

The facts are that two vessels, the *Unalga* and the *Miami*, were built. They were sister ships, with identical boilers, and everything else,—and were put into commission at the same time. After the boilers of the *Unalga* gave trouble, the details of which I will not go into now, the suggestion was made to look into this soda question. It was found that in the preceding period of about eighteen months, since the boat had been commissioned, the *Unalga* had used 3300 lb. of soda in her boilers, and the *Miami*, had used only 300 lb. and had never had any trouble. The officials of the Coast Guard decided that it was undoubtedly soda which caused the trouble. They rescinded at once the order requiring soda to be used so that the strength of the solution should not be less than 3 per cent of a normal alkaline solution; and they ordered the present practice, which makes the water slightly alkaline, so that it is only a little more than neutral,—one-half of 1 per cent of a normal solution.

The drum plates of the *Unalga* were badly cracked and the drums were condemned by the Coast Guard officials, removed, and replaced by new ones. Very shortly after that, this matter was called to the attention of the Navy Department; and after the Navy officials looked into the story of the *Unalga* and what had happened, they also rescinded the order that not less than a 3 per cent of normal solution should be used. Since that time they have been using the one-half of 1 per cent solution.

There is another point in the paper to which I object. (I understand, of course, that the author and his associates got their information as best they could, but it is unfortunate that these erroneous statements should be made.) The statement is made, in the case of the *Aylwin*, that an investigation was made by the Naval Experiment Station at Annapolis, at which the manufacturer was permitted to have a representative present; and it was settled that the trouble with her boiler was not in any way due to soda.

As a matter of fact, there was a preliminary report from the physicist of the Experiment Station, but the matter is still under investigation at the Experiment Station and has not yet been settled. There is a great deal to be said about the matter; it is very complicated, and requires the most careful study and investigation. The reason for this is not because there is any doubt in the minds of those who have studied the subject, but the deleterious action of soda comes as such a surprise to those who encounter it for the first time that they want to investigate it carefully before endorsing this explanation of the phenomena.

I am authorized to say that later on, at the proper time, men who have made a very exhaustive study of this whole effect of soda solutions on boiler seams will prepare a paper, probably for this Society, which will give the latest possible information on the subject.

Geo. H. Gibson said that he believed that the return to treating feed water inside the boiler, as advocated in this paper, is a step backwards, at least for stationary plants. That corrosion will not occur in a solution of proper alkalinity was originally demonstrated, he believed, by the German investigators Heyn and Bauer. Where water is treated externally to boilers, a greater or smaller excess of reagent is employed, which accumulates in the boilers, and the concentration of alkalinity can be controlled by periodic blowing down. If the water is treated cold, however, a considerable amount of scale and sludge forming solids fail to be eliminated in the softening system, as pointed out in the paper. On the other hand, if the water is treated above 170 or 180 deg. fahr., the reactions are practically instantaneous and the precipitate formed is coarse and settles rapidly. It is certainly better to remove these solids before the water enters the boilers, as practiced in modern hot process softening systems, wherein the water is first heated and de-aerated by spraying through a steam bath before the chemical reagents are added. The proportions of the chemical should be based on an analysis of the water, and lime, as well as soda, is necessary for the correct treatment of many waters. In this way practically all of the scale or sludge-forming substances can be eliminated, as will be seen from the following analyses of water before and after treatment, as obtained in large steam plants.

	Grains per U. S. Gallon	
	Before Softening	After Softening
Total dissolved solids	21.49	18.25
Total incrusting solids	18.28	1.35
Total dissolved solids	19.44	21.01
Total incrusting solids	9.04	0.32

The first analysis quoted shows that the water before treatment contained over eighteen grains per gallon of incrusting solids. After treatment it contained 1.35 grains and the excess of reagent used is 0.88 grain of sodium carbonate and 1.22 grains of sodium hydrate. The second sample was a water which not only deposited scale but caused corrosion, the corrosion being due to the presence of free sulphuric acid. Before treatment, this water contained 9 grains of incrusting solids and after treatment 0.32 grain. The excess of treating reagent used was 3.18 grains of sodium carbonate and 0.33 grain of sodium hydrate. He could give further examples, but these illustrate the point at issue.

This elimination of scale and sludge forming solids is found more important in preventing foaming and priming than is the keeping down of alkalinity in the boiler, as pointed out in the paper. As a matter of fact, boilers can be operated successfully at heavy overloads with high alkalinity, but not with high alkalinity combined with a large amount of sludge in the water.

A good example is presented by analyses obtained at one of the boiler plants of the U. S. Navy Department. Before the water softening system was installed at this plant, a great deal of trouble was experienced, but whether the Navy compound was tried or not he did not know. The total incrusting solids in this water were 23.29 grains. It was impossible to operate the boilers up to rating; there was severe priming, and it was found necessary to shut down part of the plant and turn off a number of lights and use less steam in order to carry the balance of the plant. It was the practice to wash the boilers out every 24 hours. After the water softening system was put in, treating the water hot, the incrusting solids were reduced to 1.75 grains, and the boilers operated successfully, although there is present in the water 67.63 grains of soluble salts.

The elimination of sludge is also desirable, because of the fact that sludge is carried over by even slight priming, and its accumulation may cause burning of superheating surface, lubrication troubles, and filling up of turbine blades. Such removal is doubtful where the precipitation of solids is brought about in the boiler itself, as advocated in this paper.

M. F. Newman (written). The author's comparison of the difference in operating conditions between the use of hard, corroding feed water, and when subjecting the feed water to treatment with compound within the boilers is merely an illustration of changing the lime and magnesia salts into precipitates of such a form that they do not adhere to the surfaces within the boiler and the effect of introducing an alkali to combine with acid radicals that, under ordinary circumstances, would be free to attack the boiler metal.

The action of boiler compounds for the precipitation of lime and magnesia and the neutralizing of harmful acids has long been thoroughly understood, and it is well known that the addition of starch and tannin to a compound consisting principally of sodium salts tends to prevent the adherence of the precipitated lime and magnesia.

While the method described by the author may be considered novel, it is by no means rational, since to impurities already present in the boiler feed there is being added more impurities, so that the final effect is to increase the impurities in the water in the boiler without any compensation in the way of removing or freeing the water from suspended matter.

The application of precipitants to water within the boiler is irrational and at the best is a mere makeshift. In the process of generating steam nothing but pure water is evaporated so that there is a constant building up of impurities in the water remaining in the boiler. This applies to the soluble salts as well as the suspended matter.

When the reactions for the precipitation of lime and magnesia are carried on within the boiler, the result is to befoul the water with suspended matter and increase its density with soluble salts, so that more fuel is required for the generation of steam and a more frequent changing of the water to eliminate the soluble solids, as well as the suspended matter collected.

The logical method of overcoming scale and corrosion in steam boilers is to properly soften and purify the water by removing all permanent hardness, eliminating or neutralizing all acids, and reducing the remaining scale-forming substances to less than 3 grains per U. S. gallon, maintaining the alkalinity in the purified water between 3 and 4 deg. (1 deg. equivalent to 1 gr. calcium carbonate per U. S. gal.) with the efluent clear and free from suspended matter. Such a softened and purified water constantly supplied to boilers will not form scale or cause corrosion.

As an illustration of the results accomplished by this rational method with a properly designed apparatus, the analyses in Table 1 of raw and softened water are submitted to show water from widely differing sources in close agreement as to alkalinity and softness.

William Kent said it seems the author has established his point in regard to the particular water dealt with, but the paper contains no reference to water treatment in general.

He thought several terms in the paper should be made clearer, for instance, "3 per cent normal," "back shopping," "engine left in chain gang," "light water," etc.

Regarding the composition of the Navy compound, he thought the expression "sufficient cutch to yield at least 2 per cent tannic acid" should be translated so that engineers could go to the chemical manufacturer and buy the materials for making the compound.

TABLE 1 ANALYSES OF RAW AND SOFTENED WATER

	A	B	C	D
	Grains per U. S. Gallon			
Volatile and Organic Matter	1.45	1.85	.75	.65
Silica	.65	.45	.35	.45
Iron and Alumina Oxides	trace	trace	trace	trace
Calcium Carbonate	17.75	1.75	10.50	1.50
Calcium Sulphate	3.06	....	....	....
Magnesium Carbonate	....	....	3.99	....
Magnesium Sulphate	12.42	....	5.07	....
Magnesium Chloride	....	....	.96	....
Magnesium Nitrate	....	....	.21	....
Magnesium Hydrate	....	.28	....	.52
Sodium Sulphate	25.77	43.59	....	6.18
Sodium Chloride	16.15	17.63	.18	1.32
Sodium Carbonate	....	.80	....	1.06
Sodium Hydrate	....	.68	....	.08
Sodium Nitrate	....	....	.21	.24
Total Solids	77.25	66.48	21.96	12.00
Suspended Matter	.15	trace	.15	trace
Free Carbonic Acid	.44	none	.44	none
Alkalinity	17.75	3.75	15.25	3.50
Incrusting Solids	33.88	2.43	21.08	2.47
Non-Incrusting Solids	41.92	62.70	.13	8.88

Analysis A Water from Shallow Creek Bank Well, Enderlin, N. D.
Analysis B Same water as A after softening
Analysis C Spring water from Clarkdale, Ariz.
Analysis D Same water as C after softening

Howard Stillman (written). Concerning the use of the 3 per cent normal alkalinity and the Navy compound on locomotive boilers in main line service, I quite differ from Mr. Babcock as to results obtained on the locomotive boilers he has experimented with. In the first place Commander Lyon states distinctly that very rapid local pitting and corrosion occurs with alkaline solution under 2.6 per cent, placing the concentration of 3 per cent as the upper limit at which corrosion will positively stop. This concentration is equivalent to 92.7 gr. sodium carbonate per gallon, or 12.2 lb. per 1000 gal., an amount of soluble matter locomotive boilers cannot carry without excessive foaming and priming. With the use of the Navy compound on locomotives Mr. Babcock is maintaining less than 1 per cent normal alkalinity, and there seems to be no reason to expect a decrease in corrosion under these circumstances, inasmuch as the potential of the liquid is considerably less than that of the metal with which it is in contact. The evidence brought out by Commander Lyon's paper indicated that corrosion would not stop until the potential of the liquid was in excess of the metal, and that 2.6 per cent normal alkalinity was required to produce this.

On many occasions I have inspected locomotive boilers using the Navy compound and as yet do not find any evidence of decrease in pitting and corrosion in bad water districts.

The Mallet Consolidation engine 4031 to which Mr. Babcock refers, was first placed in service with new firebox on Dec. 20, 1912. The Navy compound was first used in July, 1914, and maintained until June, 1915, when the engine was withdrawn from service and condemned for new firebox. During the interval the engine ran 19,790 miles. I made a thorough inspection of the firebox in the boiler shop after it had been removed, and found the crown sheet badly pitted and corroded on the water side. The Navy compound was used for a period covering but 43 per cent of the engine's mileage, but the extent of the pitting and corrosion appeared to exceed that shown by the firebox of an engine of the same class in like service on the same district, that had used Navy compound only 4 per cent of its mileage, practically none. The evidence from engine 4031 does not appear to bear out Mr. Babcock's statement regarding the effect of the compound in lengthening the life of the boiler. This engine was shopped, not for machinery, but for boiler repairs, with a firebox mileage of but 45,501.

Mr. Babcock states that the new water at Bakersfield is very light. The engineers in this district state that it appears to be light only when used with Navy compound. On engines not using Navy compound no trouble is experienced with "lightness" of this water.

Nothing new is here brought out regarding the specific action of carbonates or phosphates of soda, of which the Navy compound is composed chiefly, on such solids as produce permanent hardness. The compound is effective in breaking up and decreasing scale formation, throwing out such matter within the boiler as insoluble carbonates or phosphates in definite proportion.

The tendency for boiler water to foam and prime is a direct function of the rate of evaporation demanded. The evaporation of the Fruitvale boilers was shown by official test to be approximately 3.7 lb. equivalent per sq. ft. of heating surface per hour. My test records of the Mallet locomotive boilers show the corresponding equivalent to be at the rate of 7937 lb., or an increase over the stationary practice of 114 per cent. There is a great deal of complaint from foaming and priming from our mechanical department on the district over which the Navy compound has been used.

Mr. Babcock makes reference to the use of tri-sodium phosphate as a specific against foaming. The evidence in operation of our power does not show this to be true. He states that tri-sodium phosphate prevents the rise of the surface tension of the solution and consequent priming. I do not find any verification of this statement in principle.

The Navy compound is being applied by Mr. Babcock most generally to superheater locomotives. Superheaters are placed in boilers at considerable expense, and if, owing to circumstances, their superheating surface is converted to evaporating surface, as with priming, their efficiency is impaired in direct proportion.

In order to determine the loss of superheat efficiency from this cause, tests were made with Mikado engine 3239 in passenger service between Los Angeles and San Luis Obispo, a run of 223 miles. This engine is one of our latest type Mikados, 26 x 28-in. cylinders, 210,400 lb. on drivers, and 63-in. wheel. It has used the Navy compound and tri-sodium phosphate since July, 1915. Full details of these tests are on file, and a summary of them is here given.

TABLE 2 EFFECT OF NAVY COMPOUND IN LOCOMOTIVE OPERATION

	Out Trip—L. A. to S. L. O.		In Trip—S. L. O. to L. A.		Totals and Averages	
Date	9-23-15	9-28-15	9-24-15	9-29-15		
Navy Comp. and Tri-Sodium Phos.	with	without	with	without	with	without
Gal. fuel oil per 1000 ton-miles	17.59	16.32	11.26	10.78	13.55	12.74
Average superheat, deg. fahr.	191	265	242	249	221	254
Per cent more fuel per 1000 ton-miles	7.78	....	4.45	....	6.35	....
Per cent less superheat	27.92	....	2.81	....	12.99	....

The excesses shown for the outward trip are due to the treatment with 5 lb. Navy compound and 9 lb. tri-sodium phosphate, given at Los Angeles. With no treatment at San Luis Obispo on return trip the losses were less.

Regarding the instructions that engineers operating engines which are being treated with this compound should, as often as necessary, avail themselves of the opportunity to use the blow-off cock, heat losses from blowing off are directly proportional to the volume of water wasted and heat units therein at 200 lb. boiler pressure.

F. F. Vater contributed a written discussion, in which he stated that the water used at Fruitvale is not a bad water; as far as the analysis shows, the only corrosive elements are nitrate of lime and chloride of magnesium. In any properly operated treating plant, these two substances would have been changed to sodium chloride and sodium nitrate, both of which are inert. The analysis says nothing of the dissolved oxygen, which may or may not have been present; if present, it would have a material bearing on the trouble.

If there were no dissolved oxygen in the water after passing through the softening plant, absolutely nothing of a corrosive nature remained in the water after treatment. This being true beyond argument, he suggested that the cause of the pitting and corrosion lay entirely outside the quality of the water, and that an intelligent search would have developed the cause.

J. H. Andrew has definitely shown that a saturated solution of caustic soda will embrittle iron and steel plates by occlusion of hydrogen. At boiler temperatures it is well known that soda ash yields up its carbonic acid, changing to caustic soda. We have, therefore, in the boiler caustic soda in a dilute solution, knowing that caustic soda in saturated solution crystallizes steel and iron.

A score of plants use in their boilers well water containing sodium carbonate, and have had trouble with cracked plates, tears, ruptures and boiler explosions. It cannot be proven that soda ash was the cause, because a wide gap exists between a saturated solution, such as used by Mr. Andrew, and a dilute solution, which is all that could exist in the boiler. We know that in saturated solution the crystallization of the metal takes place in short periods of time, numbered by days. We know further that several years' time elapsed between the introduction of the water containing sodium carbonate and the culmination of the trouble.

It stands to reason that a dilute solution will bring about the same effect, given sufficient time, that a saturated solution will bring about in a short time. In none of the cases where trouble came, due to crystallized plates, did the alkalinity of the water entering the boiler equal 1/200 part normal. What then will occur in time to a boiler in which 3 per cent normal alkalinity is maintained?

He did not like the density of the water where Navy compound is used. Neglecting the starch, tri-sodium phosphate or tannic acid at Fruitvale, we have a saturation of 998.667 grains per gallon, corresponding to a density of 1.017. Sea water has densities ranging from 1.026 to 1.03. He had seen tube after tube burned out in cases where the water had a density of 1.0032. Rather than take any chances beyond 1.0032, he would use barium hydrate to precipitate the sulphate, and get rid of the sodium sulphate end product.

Mr. Babcock speaks of loss of tubes since the adoption of the Navy compound, from other causes than pitting and corrosion. To him, the cause is excessive density of the water evaporated, and the wonder is that the tube loss from density is not greater than the tube loss from pitting and corrosion.

The whole tendency of the times in boiler practice is to obtain pure waters. With pure water high evaporative capacities can be obtained without trouble, and this cannot be done by using Navy compound. With boilers constructed of pure metal, waters treated hot, and low excess of soda ash, much more economical boiler performance will be obtained, and no pitting or corrosion due to autoelectrolysis or anything else will occur.

L. M. Booth[1] (written). It is surprising that Mr. Babcock should have been content to follow, without modification, the error of Mr. Lyon of feeding to boilers a substance like carbonate of soda which contains over 41 per cent carbon dioxide,—itself a fruitful cause of corrosion. It is safe to assume that, by the use of such a large quantity of soda ash in boilers, he has transferred some of his corrosion troubles to steam lines and other surfaces beyond the boilers.

The record of the reduction of corrosion is important, although the author would have had greater success by a judicious use of caustic soda instead of sodium carbonate. It is a fact that sodium carbonate changes to caustic soda in the boiler, so why add the corrosive carbon dioxide which must necessarily be promptly expelled

[1]President, L. M. Booth Co., Hudson and Morris Sts., Jersey City, N. J.

with the steam to get in its corrosive work elsewhere? Besides, the water in question is one which requires a caustic treatment instead of a carbonate treatment.

All of the chemical information available on boiler water treatment is none too much to command when tackling a corrosion problem, so it seems that the author's "one corrective to all waters" is an endeavor to make the task more easy and simple than it really is. In the first place, let it be understood that by the elimination of the lime process—assuming that it was efficiently carried out—there was fed to the boilers of this power plant, every day of service on the rating basis, 668 lb. of sludge which might just as well have been diverted to the sewer. The presence of this suspended matter, 1 lb. per 1000 gal. of water evaporated, increases the tendency of foaming and necessitates more blowing off than if Mr. Babcock had used the lime process on the feed water as preliminary to the excess soda method in the boilers.

While no data are given to show the quality of the water delivered by the water treatment plant which was abandoned for the Navy compound process, it is to be assumed that the water was not properly treated, since in Mr. Geibel's supplement are recorded unfavorable results in transformer cooling coils, etc., and also that there was hard scale in the boilers.

I wish to endorse the general plan of testing the boiler blow-off water, although I recommend tests be made to determine the amount of suspended matter, in addition to the alkalinity tests described. The additional information is valuable and it is but slight trouble to obtain it, the process being much quicker and simpler than the alkalinity test.

E. N. Trump thought the paper implies that the cracking is due to excess of caustic soda, whereas it has not been proven that caustic soda is the cause of the cracking of steel. It may be the cause of cracking of some kinds of steel, but not enough is known about the subject to make exact determinations. He had undertaken many investigations and performed various kinds of experiments to find out the cause of the cracking of steel in contact with caustic soda, but so far without result.

He thought the statements made in the paper in this connection are not based on adequate scientific investigation. From his experience there certainly existed no cracking because of carbonate of soda. An excess of caustic soda appears in some instances to give trouble, but only in the case of certain kinds of steel.

Frederick E. Geibel said, in reply to a question by E. P. Bates, it was found in experiments that corrosion was most rapid with a temperature of 190 deg. fahr., and results with the boilers in service checked with experimental figures.

J. F. Walsh explained the shop terms questioned by Professor Kent. He said "back shopping" refers to putting a locomotive in what is termed the "back shop" after it is practically worn out. In this shop the flues are ordinarily removed, and the entire engine, boiler and parts thoroughly overhauled.

The term "chain gang" is used in the practice of running engines with other than regularly assigned crews, where the crews vary from trip to trip. A chain gang is such a crew.

"Light water" is water which foams easily. Such water is likely to be carried over into the cylinders and cause much trouble.

The Author. In answer to those advocating pre-treating of the boiler water, I might say that any method of purifying the water before it enters the boiler is certainly preferable, all things considered. The hot treatment is believed to be more effective than the cold, but is necessarily not applicable to water for locomotives. The cold treatment has been used very extensively, but unless water is given ample time to settle, the desired results are not obtained. Recent experience with lime-treated water by the cold process bears out the statement made by Commander Lyon—that lime water under certain conditions is highly corrosive.

Mr. McFarland promises some further data on the destructive action on boilers carrying water at high alkalinities. Since our experience has failed to develop a single case of trouble from this source, his paper will be received with much interest. It is to be hoped that he will be sufficiently explicit, and that his statements will be sufficiently supported by unbiased evidence to justify his charges that many of my statements are inaccurate.

It is quite evident that Mr. Stillman has not followed personally the tests on locomotive boilers using Navy compound. No attempt has been made to carry a 3 per cent solution in locomotive boilers, but only a sufficient quantity to prevent scale and corrosion. The mileage on Engine 4031 given by Mr. Stillman may not be considered as a very good record, but it is 59 per cent greater than the previous record of this same engine not using the compound. His statements in regard to the effect of the compound on the superheat are based only on one round trip. Fuel records on eleven superheat engines over a period of twenty months show a 5 per cent decrease in gallons of fuel per 1000 ton-miles in favor of the engines treated with Navy compound. And he persists in a total lack of appreciation of the well-known fact that a boiler can better be blown off by a series of quick, short blasts than by opening the cock wide for several minutes. Furthermore, he fails apparently to comprehend the meaning of the upper and the lower limits fixed by Lyon's investigation, beyond which *in both directions* corrosion does not occur. The zones between neutrality and the lower limit, and between the upper limit and any higher degree of alkalinity yet observed, are those of no corrosion. (See "Effect of varying degree of concentration of alkaline solutions," in paper by Lieut-Com. Frank Lyon.[1])

The remainder of Mr. Stillman's discussion is founded on evidence derived from sources we have long since found to be unreliable.

[1]Jour. Am. Soc. Nav. Engrs., August, 1912.

No. 1521

AUTOMATIC MECHANICAL CONTROL OF LATHES AND SCREW MACHINES

By Luther D. Burlingame, Providence, R. I.
Member of the Society

This paper has been prepared as a companion paper to that on Electric Control of Machine Tools presented before the Society at this meeting by L. C. Brooks, Jun.Am.Soc.M.E., and also to the paper on Automatics presented by Ralph E. Flanders, Mem. Am.Soc.M.E., before the International Engineering Congress, San Francisco, September, 1915. It will be confined to a discussion of the automatic control of lathes and screw machines for the reason that these arts show the most highly developed examples of automatic control,—naturally so, as the attention of designers and inventors has longest been directed to developing such features in these lines of machines.

2 It is the purpose of this paper to supplement that of Mr. Flanders by illustrating with specific cases the general principles and organization of devices for automatic mechanical control. There is another fertile field for investigation and discussion not dealt with here, that is, the use of fluid control, including pneumatic control, a field which the Patent Office records show to have already received a great amount of attention and which will no doubt be further developed in the near future.

3 While it is true that the automatic control of machine tools began when the first power feeds, reversing mechanisms, etc., replaced hand operations, these features have become so well known that at the present day something further is meant when we speak of automatic control. In a general way the term "automatic control" now applies to the organization of a machine so that *all* operations required to complete the work are automatically performed, and the object is to have these operations so performed as not only to secure

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Machine Shop Practice.

in large measure the advantages of hand work guided by human intelligence, but also to insure a uniformity and quantity of product beyond that which can be commercially obtained by hand work,—uniformity, in that the same conditions prevail for producing each successive piece, and increased quantity, in that the element of fatigue is absent and that the workman can often keep in operation more than one machine.

4 While as a general rule automatic control means added complication of mechanism, many "single purpose" machines having full automatic control which are even simpler than the corresponding machines designed for general use but without the features of full automatic control are in use today.

5 In determining whether the employment of automatically controlled "single purpose" machines is warranted, the vital question is whether the product is wanted in sufficiently large quantities and whether the design is sufficiently well established to justify the investment in such special machines. The same question presents itself in considering the use of automatically controlled machines for general work as compared with semi-automatic or hand machines.

6 It must be determined whether the added original cost and greater cost for repairs are justified when to this must also be added the more expensive tool equipment and longer time required for setting up.

7 In considering the cost for repairs, the conditions to be borne in mind are the greater danger of a breakdown and the greater skill required to keep the machine in running condition. It must be borne in mind that while repairs are in progress a more expensive equipment is standing idle and the output is curtailed to a greater degree.

8 In spite of such drawbacks, each year sees an advance in the successful use of automatically controlled machines. The tendency to manufacture in larger quantities, including the growing demand for large numbers of duplicate parts, provides a legitimate field for many kinds of automatically controlled machine tools, both general and special.

9 The fact, already mentioned, that in operating automatically controlled machines the human factor is less in evidence than is the case in hand operated machines, makes it possible generally to employ less skilful workmen, or workmen who have had a shorter term of training, without lowering the quality of the work. On the other hand, the use of automatically controlled machines increases

the need of skilful supervision and of skilled men for their construction and repair.

10 The unit plan in the design of automatically controlled machines makes it possible often on a "general purpose" machine to add attachments to the simpler or basic design so as to perform automatically the more intricate special operations without adding new machines to the equipment.

11 The features most prominent and essential in the automatic control of machine tools can be classified as follows:

I Spindle drives
II Means for inserting, feeding and removing the work
III Tool feeding mechanisms
IV Indexing mechanisms
V Controlling means for the various mechanisms

I SPINDLE DRIVES

12 Features of automatically controlled spindle drives may be classified as follows:

a Speed change
b Reversal of spindle
c Stopping of spindle

13 *a Speed Change.* In automatic turret machine work it is often important to have more than one spindle speed available during the operation on a given piece of work, in order that time may be most fully economized. At the present time such machines usually have a constant speed drive, not only to secure power on the slow speed where most needed, but also so as to be convenient for applying a constant speed motor drive.

14 The automatic change of spindle speeds in such cases is usually provided for by gearing controlled by the mechanism of the machine. Fig. 1 shows the front view of an automatic turret lathe embodying this feature, and Fig. 2, a section through the spindle. In this construction any one of eight changes of spindle speed can be automatically obtained, the changes being made by means of an intermittently revolving drum carrying dogs which shift levers 1, 2 and 3 as desired. Levers 1 and 3 control clutches to engage gears giving four speeds, and the number of speeds so obtained can be doubled by the operation of lever 2 which either clutches direct to the spindle for the fast range of speeds or connects through differential gearing for a slower range of speeds. The dogs on the

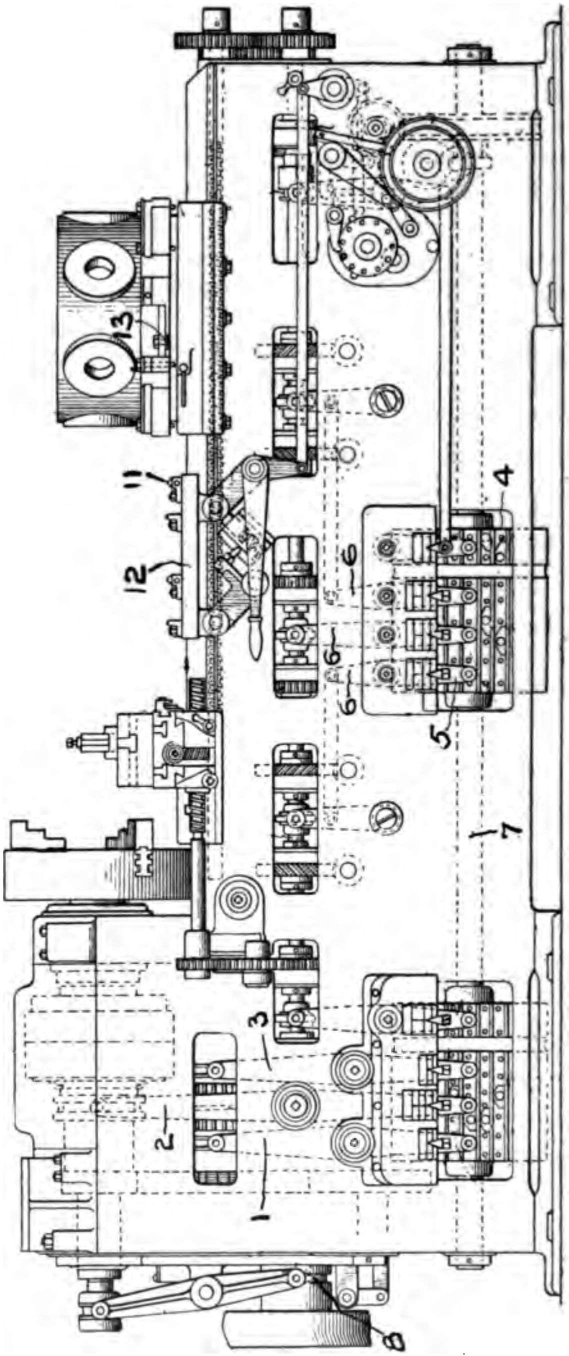


FIG. 1 AUTOMATIC LATHE WITH VARIABLE SPEEDS AND FEEDS CONTROLLED BY DOGS ON DRUMS. GISHOLT MACHINE CO.

drum may be so set as to bring the levers into idle positions, thus disconnecting the gearing.

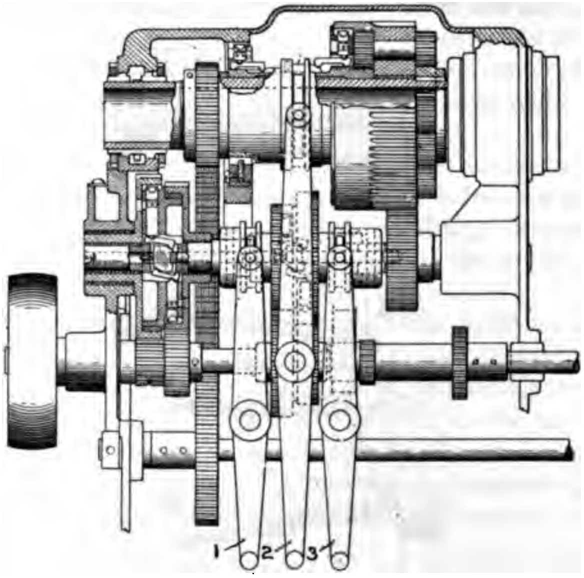


FIG. 2 VARIABLE SPEED SPINDLE DRIVE OF AUTOMATIC LATHE MADE BY GISHOLT MACHINE CO.

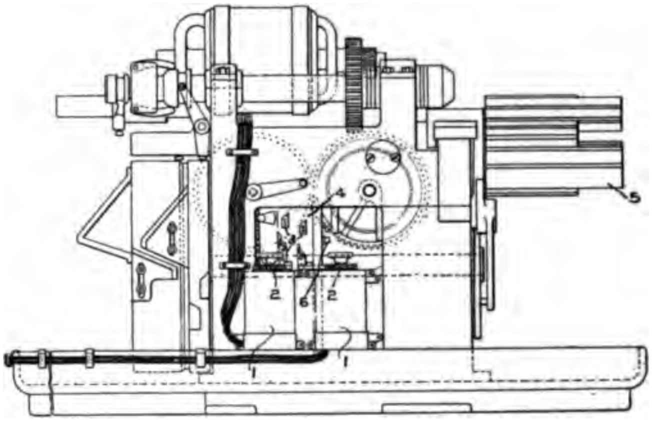


FIG. 3 MOTOR-DRIVEN TURRET LATHE OF THE GRIDLEY TYPE MADE BY THE WINDSOR MACHINE CO.

15 Another typical illustration of automatic control of spindle speeds is found in the machine shown in Fig. 21 where dogs on the

disc 4 cause the shifting of clutches engaging gears to vary the speed.

16 At the risk of encroaching on the field covered by Mr. Brooks

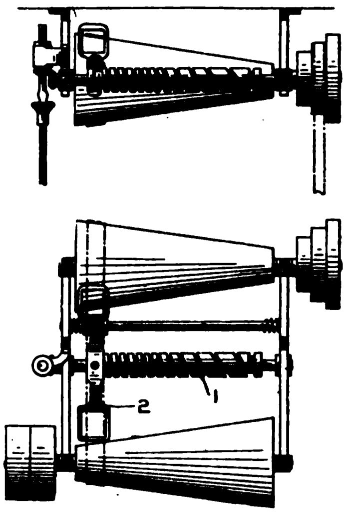


Fig. 4 Countershaft for Cutting-off Machine to Give Constant Cutting Speed

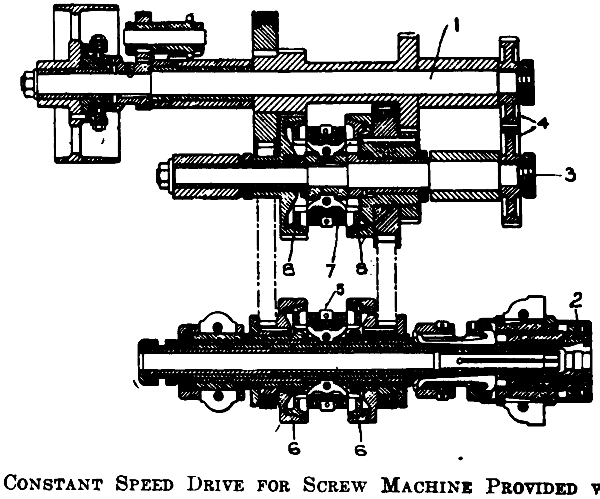


Fig. 5 Constant Speed Drive for Screw Machine Provided with Automatic Reverse and Speed Change. Brown & Sharpe Mfg. Co.

in his paper, an example of electric control of the spindle speeds and feeds is shown in Fig. 3. The changes of speed are entirely automatic, the controllers being operated by cams on the operating cam drum, this being an illustration of the combination of mechanical and electric control. In this machine the rheostats 1 are connected with independent motors. Rheostat adjusting pinions 2 are operated by lever segments which are engaged by dogs 3 on the drum 4, and by this means both the spindle speeds and feeds can be automatically varied as desired.

17 In types of machines such as cutting-off machines or those on which squaring-up operations are largely performed, a gradual and continuous change of speed is desired so that a constant cutting speed of the tool will be maintained from the periphery to the centre of the work or vice versa.

18 In order to obtain a constant cutting speed for cutting-off, the requirements are that there should be an accelerating speed for the work. A means for automatically accomplishing this is shown in Fig. 4. The automatic shifting of the belt on the tapering cones gives the desired change of spindle speed. Cam 1 is connected by gearing to the tool carriage of the machine and controls the traverse of the belt shifter 2, so that by providing the required accelerating lead to the cam, a constant cutting speed is obtained.

19 *b Reversal of Spindle.* In designs of machines where the threading or other operations require a reversal of the spindle, this is automatically accomplished in various ways, an example being shown in Fig. 5. From the constant speed shaft 1, the spindle 2 is driven in either direction and at various speeds, the control being automatic for direction and for one change of speed. Other changes of speed are by means of change gears 4.

20 The reverse is obtained by shifting thimble 5, engaging respectively friction clutches 6 connected by chain and sprocket with the oppositely revolving shafts 1 and 3. The shifting of the thimble is by means of a lever operated by a cam on an intermittently revolving shaft. The intermittently revolving shaft is in turn set in motion by a trip lever operated by dogs on a continuously revolving disk.

21 The automatic change of spindle speed is by means of thimble 7 engaging respectively clutches 8 also operated by lever connection to a cam on an intermittently revolving shaft.

22 *c Automatic Stopping of the Spindle* is required for various kinds of work. Some screw machines are so designed as to unthread the die or tap by stopping the work spindle to run the die or tap off after the threading operation. Sometimes the spindle is stopped to

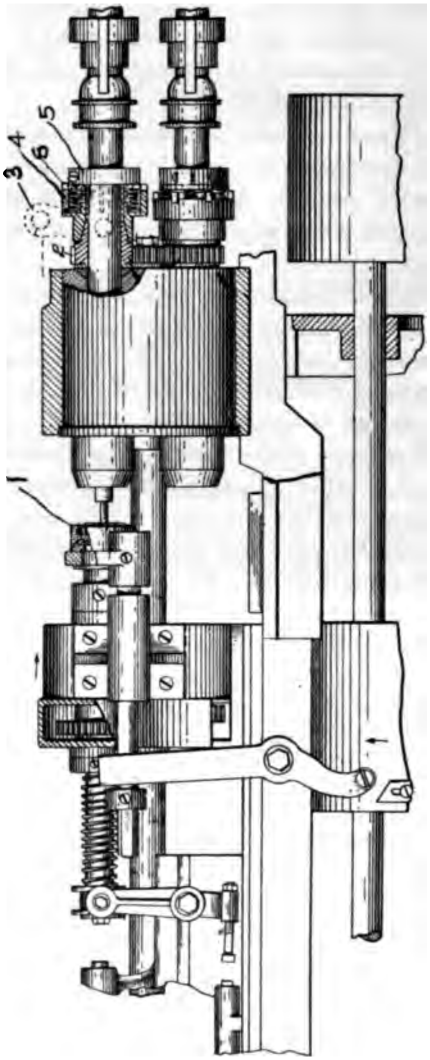


Fig. 6 Multiple Spindle Screw Machine having Means for Automatically Stopping the Spindle. National Acme Mfg. Co.

perform a milling, cross drilling or similar operation, and sometimes to save time in removing and replacing work when this is a hand operation.

23 An automatic screw machine having the feature of stopping the spindle for the purpose of threading is illustrated in Fig. 6. This is shown applied to a multiple spindle machine in such a manner that the bar of stock opposite the die spindle is stopped and held from revolving without stopping the other bars of stock which are simultaneously being operated on by other tools. After the forward movement for threading has been completed the stock is again revolved, and by this means the tap or die is backed off.

24 The revolving tap or die spindle is shown at 1, and when each successive work spindle is brought in line with it, a clutch on the constantly revolving gear 2 is disengaged by the lever 3 operated by a cam through a series of levers. This lever, 3, engages a sleeve 4 which is normally in spring pressed engagement with gear 2, disengaging same from gear 2 and engaging it with collar 5 which is fast to the spindle and which constitutes a brake to stop the spindle from revolving. For heavy work, where a positive stop is desired, a pin in collar 5 engages projections 6 on sleeve 4. The controlling cam can be so adjusted as to start the spindle revolving at the completion of the forward movement of the tap or die. Or, in the case of using an opening die the spindle can remain stationary when the die runs off.

25 A method of stopping the spindle of a single spindle machine for the same purposes as above is to apply a brake in place of the belt on the "reverse" driving pulley. When using such a brake the throwing of the clutch to connect with this pulley stops the spindle. The brake is so adjusted by the use of a spring as to give a cushioning effect in stopping, a feature of special importance in high speed machines.

26 As already pointed out, such provision for stopping the work spindles also facilitates the work of cross drilling, slabbing, etc. Another use is to stop the spindle at the end of a cut so that the cutting tool can be drawn off the work in a straight path, thus avoiding an irregular scoring of the work such as results when the tool is drawn off while the work is revolving.

27 A modification of spindle-stopping means is to revolve intermittently or index the work spindle for various operations.

28 An illustration of automatically stopping the spindle for the purpose of saving time when changing the work by hand is shown

in Fig. 21, where the clutch 6, can be thrown into engagement with the brake 7; also in the Fay Automatic Lathe, built by the Jones & Lamson Co., this lathe is provided with automatic spindle stopping means which act at the completion of each cycle of operations. The construction is such that the spindle can also be stopped by the operation of a foot treadle.

II MEANS FOR INSERTING, FEEDING AND REMOVING THE WORK

29 Means for inserting and removing the work will be considered under the heads:

a For bar stock

b For chucked work

c For transferring work for secondary operations

30 *a Means for Feeding and Holding Bar Stock.* Bar feeding devices have in a general way followed the lines of the hand operated Parkhurst feed, brought out in the shops of the Pratt & Whitney Co. about 1871.

31 The use of feeding fingers and roller feeds, in the latter case necessarily feeding against a positive stop, are forms of development which have since followed. In the former case graduated levers or scales determine the distance the stock is fed. An illustration is shown, in Figs. 7 and 8, of an automatic feeding device which can be adjusted so as to feed any required distance from zero to the full traverse of the machine. If desired, the feeding operation can be made to repeat so as to produce certain kinds of work which are longer than the normal capacity of the machine. By means of the crank 1, the screw 2 adjusts a nut carrying a block 3, so that the motion of the lever 4 operated by the cam 5 may give any desired feed to the slide 6 and thus to the feed tube 7, the setting being to a graduated scale. At the forward end of this feeding tube are feeding fingers to engage the bar of stock to feed it forward. After the stock is gripped by the chuck by means of the chuck levers 8, operated by the cam 9, the feeding fingers are retracted, ready for the next operation.

32 A device for stopping the machine when the bar of stock becomes exhausted is shown in Fig. 9; this result is accomplished by mechanism controlled by the disengagement of the feeding devices with the stock. This is so designed as to stop the machine with the jaws of the chuck open, so that a new rod of stock may be quickly inserted. It is also devised so that the machine is not stopped until the length of stock projected by the forward movement

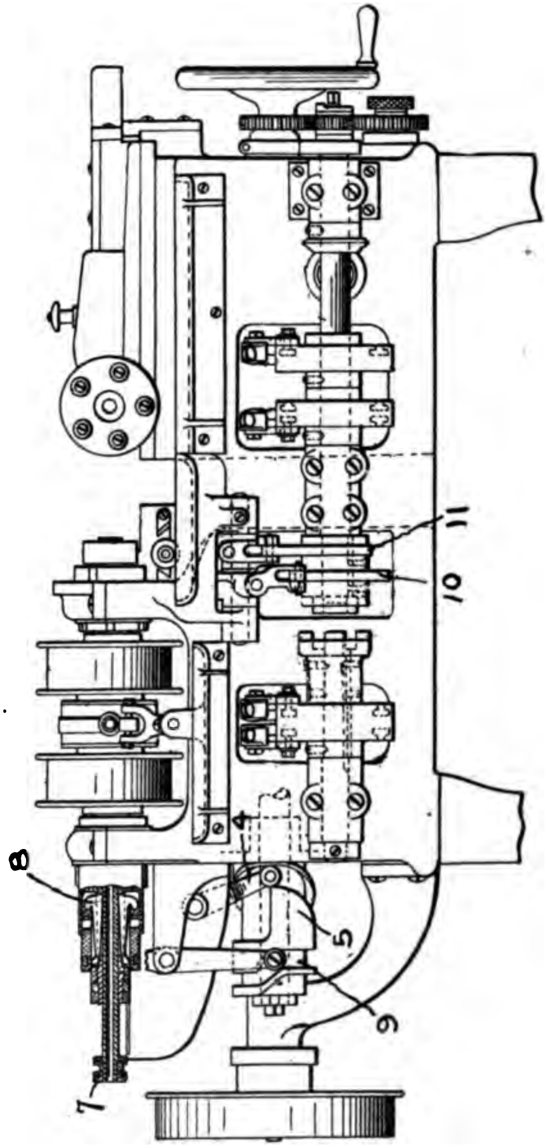


Fig. 7 Automatic Screw Machine showing Stock Feeding Mechanism and Method of Controlling Feeds. Brown & Sharpe Mfg. Co.

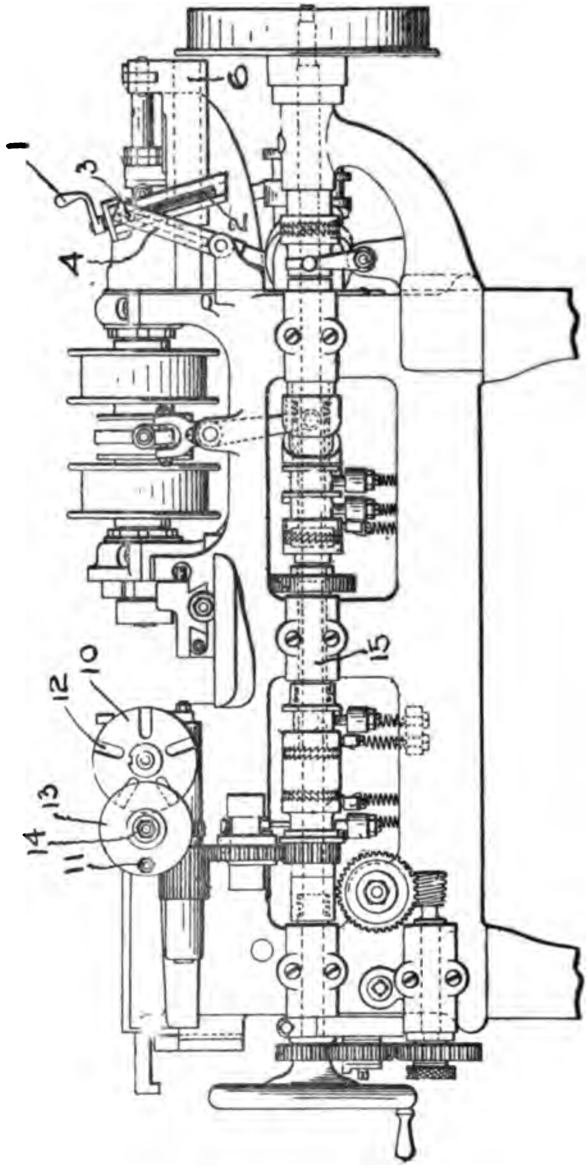


Fig. 8 Automatic Screw Machine showing Feeding and Indexing Devices, also Use of Controlling Shaft. Brown & Sharpe Mfg. Co.

of the feeding devices preceding their disengagement with the stock is acted on and severed from the stock; this is accomplished by so constructing the stop mechanism that it is thrown into operative position when the feeding devices are disengaged from the stock, but does not operate to stop the machine until the feeding devices are again advanced.

33 In this construction the slide 1, connecting to the feeding tube by the grooved collar 2, is drawn back by the spring 3 when the stock passes beyond the feeding fingers, there being then no friction to hold it. This rocks dotted lever 4, the movement being made possible by the widened space in the cam groove at 5. This allows the projection 6 to pass the latch 7 so that on the next revolution of

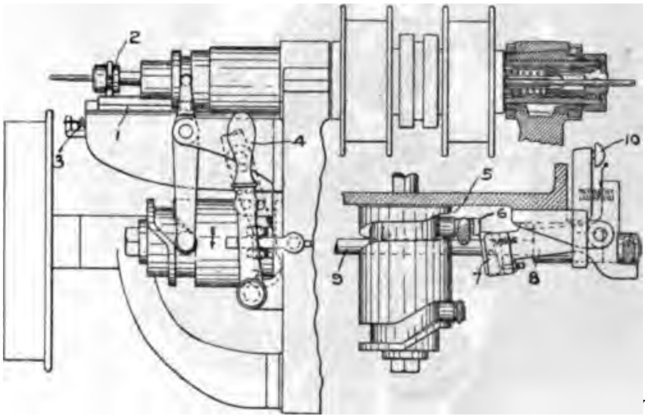


FIG. 9 DEVICES FOR STOPPING MACHINE WHEN BAR OF STOCK IS EXHAUSTED. BROWN & SHARPE MFG. CO.

the cam the lever 8 carrying the latch is rocked, this in turn rocking shaft 9 and throwing the driving mechanism out of operation, also sounding the gong 10, to notify the operator that a new piece of stock is needed.

34 A well-known design of the roller principle for feeding the stock is shown in Fig. 10, where the rollers 1 engage the bar of stock and feed it forward against a stop, the rollers slipping on the stock after it has been advanced the required distance and until the chuck jaws have engaged same, when by the release of the ring 2, which during the feeding operation has been held from turning, the rolls 1, with their driving mechanism, turn idly with the revolving bar of stock.

35 The driving means for the feed rollers consists of the circular rack 3, on the intermittently revolving ring 2, which, when held stationary, revolves the gear 4, which in turn revolves the worms 5 and the worm wheels 6, the latter being fast on the shafts carrying the feed rollers.

36 A deflector to separate work and chips is shown in Fig. 11 and operates as follows: While normally held out of the path of the falling chips by the spring 1 the deflector 2 can be swung into a position below the spindle at the moment the work is severed and thus

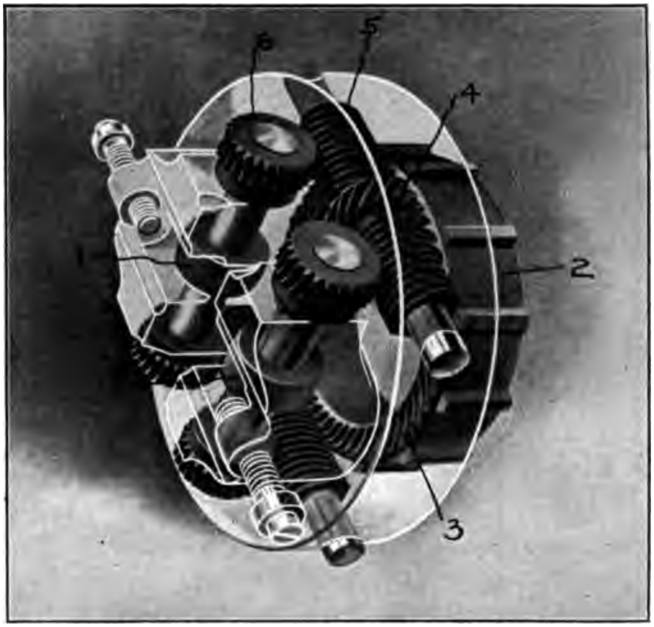


FIG. 10 ROLLER FEED. JONES & LAMSON MACHINE CO.

deposit the work in a receptacle separate from the chips. The deflector is operated by a dog 3 on the disk 4.

37 A counting device can also be attached to record the number of pieces made, and, if desired, this can be so made as to stop the machine automatically after the required number of pieces have been produced.

38 *b Means for Inserting and Removing Chucked Work.* This may be considered through successive stages from the hand-operated method of the Fay automatic lathe, where the work, when it

is to be finished on an arbor, is driven on one arbor by the workman so as to be ready to replace the piece being operated on when that is finished, the automatic feature in this case being the stopping of the spindle as already described.

39 Stopping the spindle for the purpose of changing the work is also a feature of value on automatic machines when operating on hand chucked work, an illustration being shown in Fig. 1, where the shifting of the levers 1, 2 and 3 to neutral positions disconnects the power and stops the spindle.

40 Magazines for handling work to be chucked automatically have developed along many lines, and a great number of ingenious

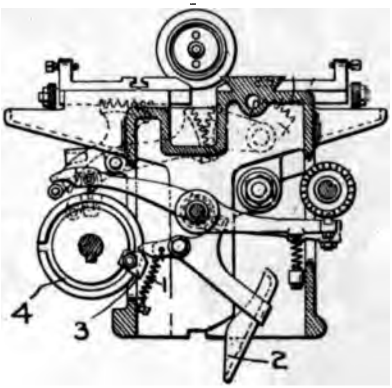


FIG. 11 CROSS SLIDE OPERATING MECHANISM AND DEFLECTOR TO SEPARATE WORK AND CHIPS. BROWN & SHARPE MFG. CO.

devices have been devised adapted to the various shapes and kinds of work to be operated upon. Several of these are shown in Figs. 12, 13 and 14.

41 Fig. 12 shows a tilting magazine attachment with the magazine 1 in position so that the conveyor 2 can advance to take a piece of work. After the piece of work is taken by the conveyor the magazine tilts up out of the way of the turret tools. The conveyor 2 then brings the piece in line with chuck 3 and deposits it in same. The conveyor is free to revolve so as to facilitate the pressing of the work into the revolving chuck. An ejector inside the spindle removes the work when completed.

42 A form of hopper for feeding studs into the rear end of the spindle is shown in Fig. 13. This might be called a reservoir magazine, as it has a widened upper portion to carry a large number of

pieces. An agitator 1, operated by a lever 2, makes the feeding sure.

43 The frame 3 in Fig. 13 is adjustable for different lengths of studs. The studs are fed positively into the back end of the spindle by the rod 4.

44 Another form of magazine is shown in Fig. 14 adapted to

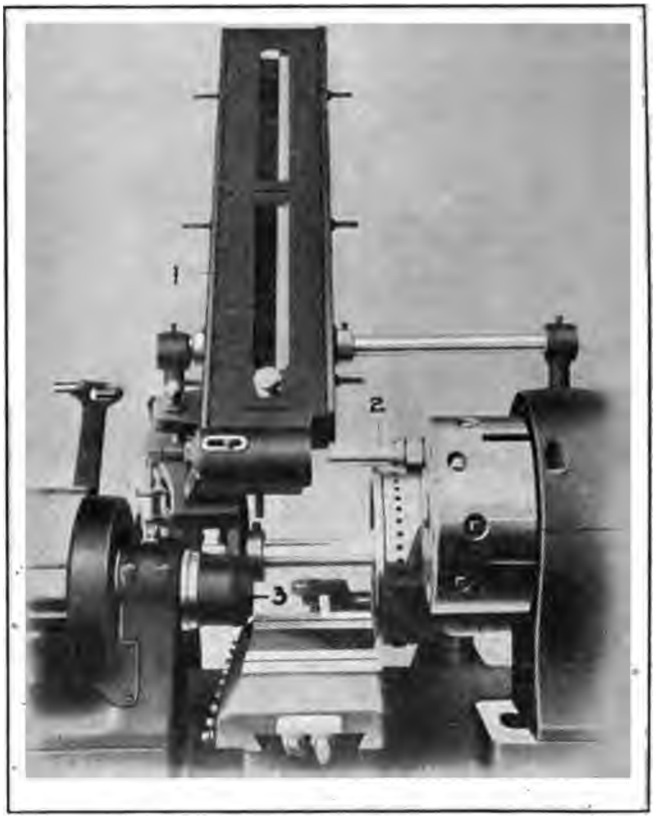
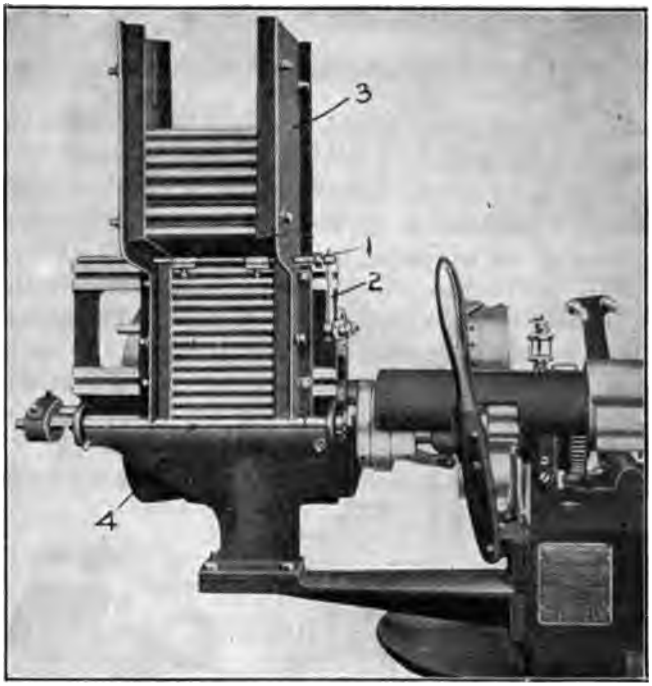


FIG. 12 TILTING MAGAZINE ATTACHMENT. CLEVELAND AUTOMATIC MACHINE CO.

handle more irregular shaped pieces. The pieces are placed by hand in bushings in the rotary magazine. In the illustration a piece of work 1 is shown as having been taken from one of the bushings 2 when the turret 3 was at its forward movement, and this will be placed in chuck 4 when the turret is indexed so as to bring them in



IG. 13 VERTICAL HOPPER MAGAZINE. CLEVELAND AUTOMATIC MACHINE CO.

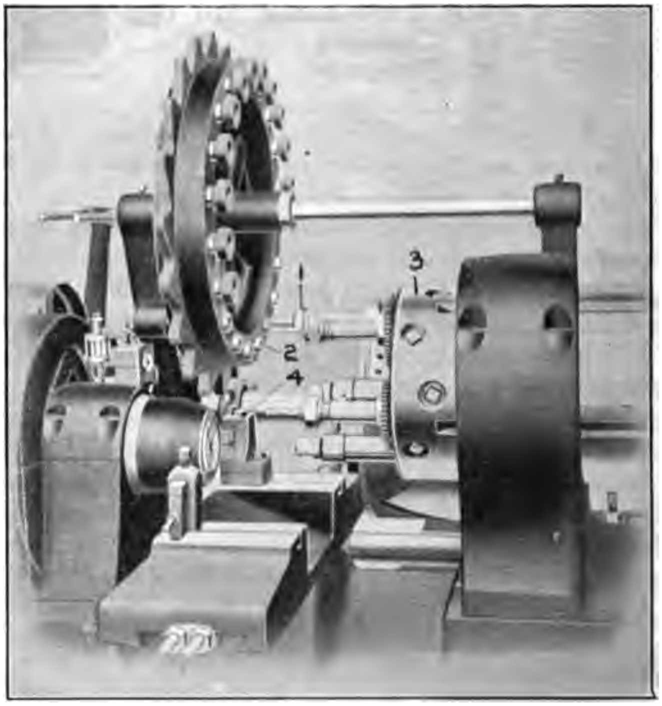


FIG. 14 INDEXING MAGAZINE ATTACHMENT. CLEVELAND AUTOMATIC MACHINE CO.

alignment. The magazine is indexed by a dog on a cam shaft in the rear and held in position by a locking pin after indexing.

45 A design by which blanks which are to be drilled and tapped are fed from a magazine to the chuck in one of the spindles of the rotary turret by the advancing movement of the drill itself is shown in Fig. 15. In this case the hole is partially drilled in the same indexed position in which the blank is chucked. The remainder of the drilling is done in successive spindle positions and the tapping is done in the final position. When the tapping is completed, the chuck releases the piece which is withdrawn by the tap itself, the tapped piece passing off over the curved shank of the tap.

46 In this case the spindles run at a suitable speed for tapping,

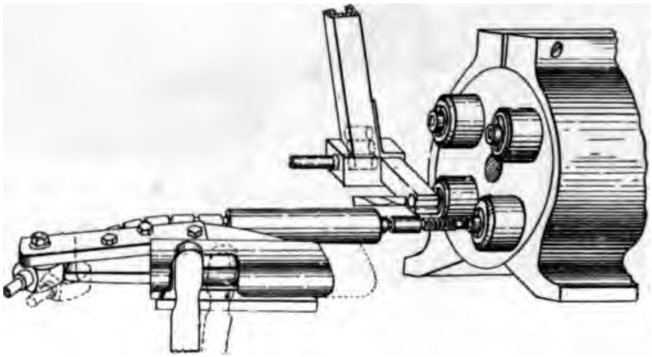


FIG. 15 DRILLING AND TAPPING ATTACHMENT. NATIONAL ACME MFG. CO.

the higher speed required for drilling being obtained by running the drill spindles in the opposite direction to the work spindles.

47 Large or irregular work presents many difficulties in automatic chucking, and hand methods are usually resorted to for such work. In some cases, however, automatic means have been devised for the chucking operation.

48 Fig. 16 shows a magazine for automatically feeding sewing machine handwheels to the chuck of a turret lathe. The slide 1 is reciprocated by means of levers 2 and 3 and connections to roller 4 which is operated by cam 5. Attached to slide 1 are gripping jaws 6 which, when in the upper position, engage a handwheel in the magazine and, on the downward movement of the slide, bring it in line with the chuck as shown in the illustration. The work is then gripped in the chuck and the gripping jaws of the feeding device are retracted.

49 While this device has been in use for many years it is doubtful whether such saving of time as results is sufficient to pay for the expense of keeping it in working condition. It is illustrated mainly

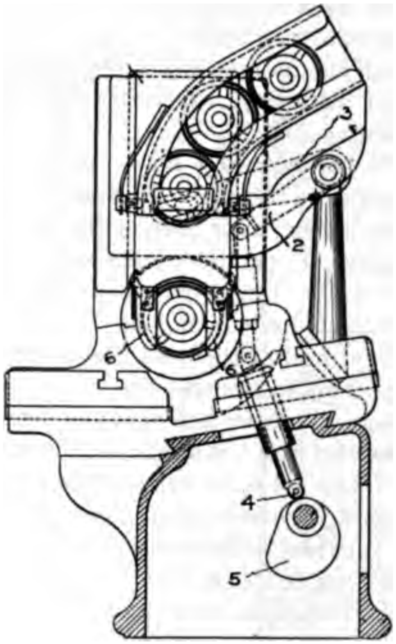


FIG. 16 MAGAZINE ATTACHMENT FOR IRREGULAR SHAPED PIECES

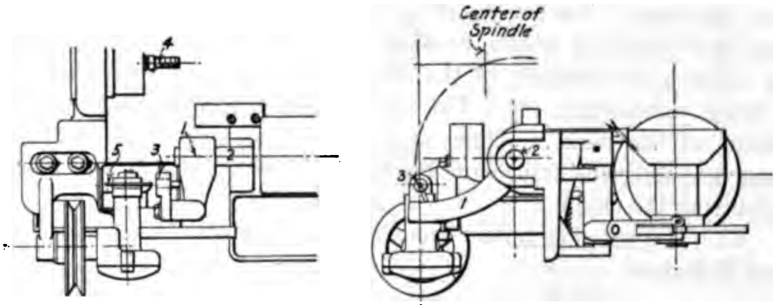


FIG. 17 TRANSFER ARM FOR SECONDARY OPERATIONS. BROWN & SHARPE MFG. CO.

to show the limitations of devices for this purpose and to emphasize the fact that in such cases it is a question whether automatic means can successfully compete with hand work.

50 Even when placing the work in the chuck by hand, automatic devices can be made to assist in ways additional to that already mentioned, of stopping the spindle ready for the insertion of the work. Fluid or pneumatic means can be employed for gripping the work, and this can be so applied as to reduce the pressure for the finishing cut, still maintaining sufficient to hold the work securely but without risk of distortion. Means for automatically ejecting the work can also be applied.

51 *c Means for Transferring Work for Secondary Operations.* It is a common practice to transfer work from the main spindle to an auxiliary spindle or holder, after part of the operations have been performed, for additional operations such as milling, cross drilling, etc. Such operations are thus performed without requiring additional time and without extra handling. This is a substitute for the use of a hopper for feeding the work into a magazine in a secondary machine to perform these operations.

52 Fig. 17 shows a transfer holder in use for slotting the heads of screws. The transfer arm 1 is swung by the rock shaft 2 so that the hole 3 is in line with the work 4 in the main spindle and engages same before the work is severed from the rod. After the work is severed the arm 1 is swung to the position shown in the illustration. The rock shaft 2 is then fed longitudinally towards the saw 5, both this motion and the rocking motion being imparted by cam action. The cam 7, Fig. 18, (using the same transfer device), serves to rock the shaft 2 while cam 6 feeds it longitudinally. In this latter illustration the transfer holder is used to transfer the work to an index drilling attachment by which the work can have a series of holes drilled in it, or a modification of the attachment provides for index slab milling, castellating, etc. The attachment is timed with the movements of the main machine and three cams control its motions: cam 3 feeding the drill; cam 4 controlling the indexing, and cam 5 operating the chuck.

53 The transfer holder can be made to turn the work end for end if desired.

III TOOL FEEDING MECHANISMS

54 Feeding mechanisms may be classified by the methods of controlling the movements as follows:

- *a* Controlled by removable strap cams
- *b* Controlled by permanent cams, and adjustable cams and dogs

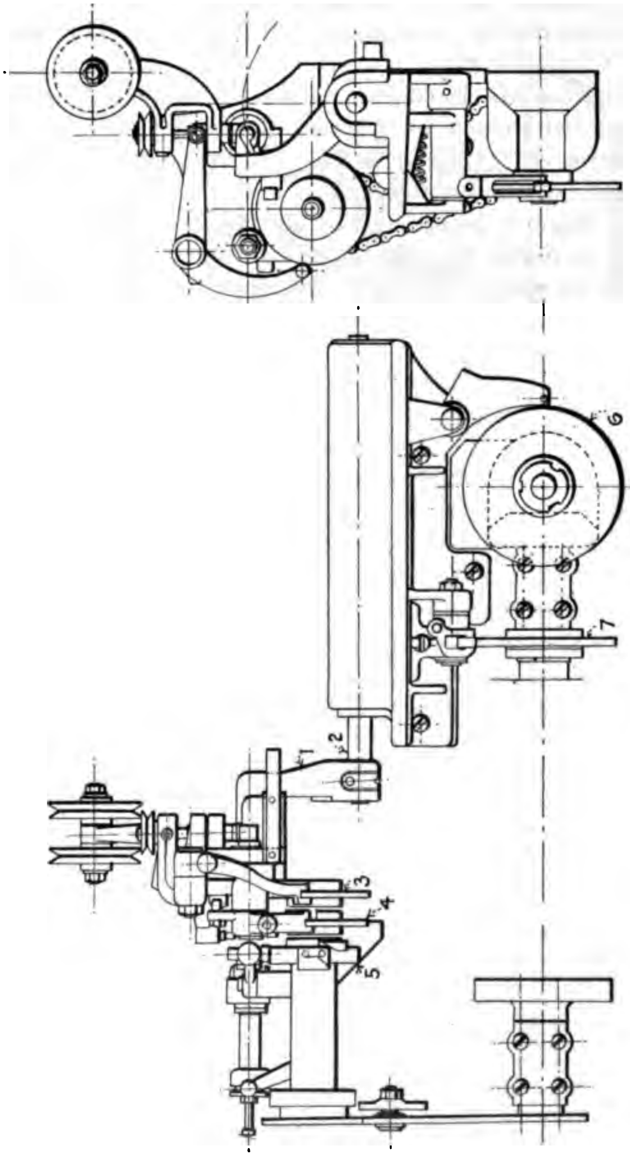


Fig. 18 Index Drilling Attachment. Brown & Sharpe Mfg. Co.

c Controlled entirely by adjustable dogs on drums

d Controlled by cams specially formed for each job

e Controlled by permanently set dogs in combination with adjustable stops.

55 Following Mr. Flanders' analysis, there would be a further classification determined by whether the cam 8 acted direct or through screws, etc. Gearing is also used in some of these various types to vary the speed of the cams.

56 The object to be attained by all these means is to save time by speeding up during the idle movements, and to provide the most efficient feed for each operation of cutting by providing a change of feed which can automatically be made effective during the operation of cutting.

57 *a Controlled by Removable Strap Cams.* This type had its origin in the Spencer machines, first brought out in the early 70's, and one of the first successful automatic screw machines placed on the market. A recent development of this type of machine and one also designed by Mr. Spencer is shown in Fig. 19. In this machine the stock feeding mechanism, the tool feeding mechanism and the cross feeding mechanism are all operated by adjustable strap cams. In the case of the two former, the straps are on the periphery of drums shown at 1 for the stock feed and at 2 for the tool feeding mechanism: in the case of the last, as well as for indexing and locking the turret, they are on the faces of disks (shown at 3 for the cross slide and at 4 for the indexing).

58 The machines of the Hartford Automatic Screw Machine Co., in this country and of the Alfred Herbert Co., abroad, are among the best known examples of using this method of camming.

59 *b Controlled by Permanent Cams and Adjustable Cams and Dogs.* An example of the combined use of permanent cams with adjustable cams and dogs is shown in Fig. 20, where the feed of the individual tool holders 1 is controlled by the permanent cam 2. The rate of revolution of this cam 2 is controlled for varying the feeding movements by an adjustable cam 3 which through roll 4 governs the position of friction wheels 5 between the disks 6. Besides this variable feeding movement, a quick movement of the cam 2 is obtained by action of the cam 7 which operates a double clutch 8 to connect either direct, giving a quick movement, or through the reduction gears 9 for the feeding movement.

60 The strap cam 10 regulates the cross feed, and this cam also partakes of the quick and slow movements controlled by clutch 8;

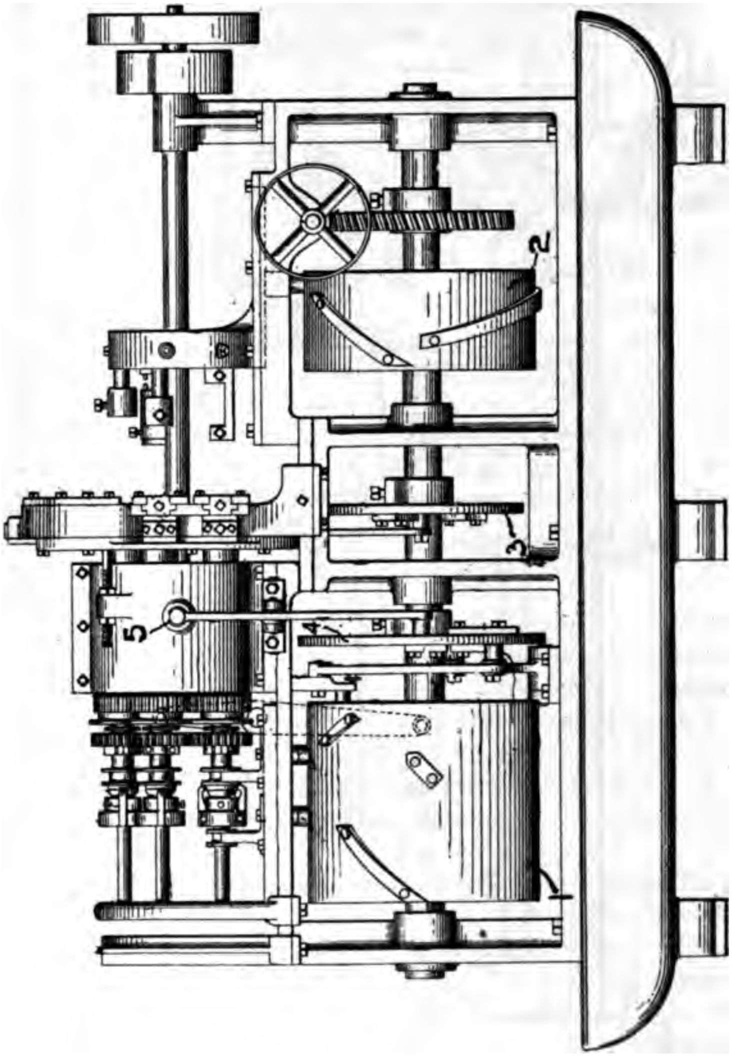


Fig. 19 Spencer Type Multiple Spindle Machine showing Use of Strap Drum Cams. New Britain Machine Co.

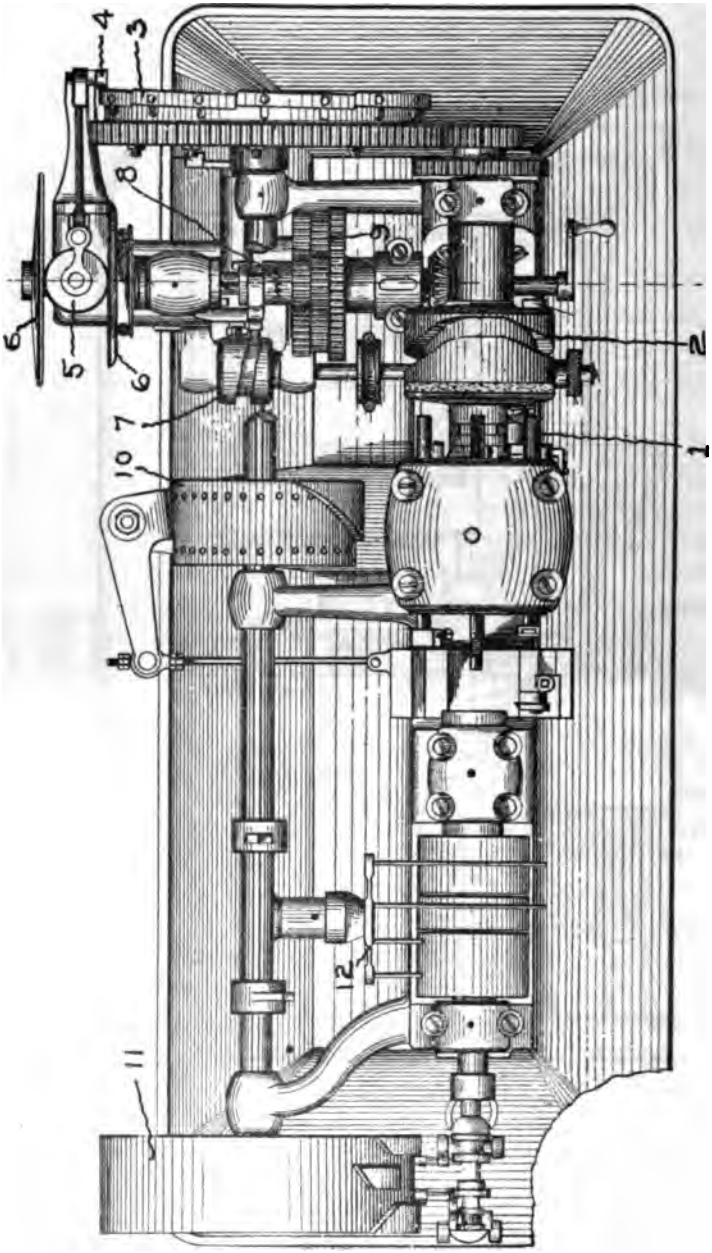


Fig. 20 Showing Control by Permanent Cams and Adjustable Cams and Dogs. Cleveland Automatic Machine Co.

it thus provides for moving the cross slide forward quickly to the point of cutting and then reducing to the required cutting feed, after which it may be quickly returned so as to bring the rear tool into cutting position following which the feeding movement is again engaged.

61 In the illustration shown, the feeding of the stock and the operation of the chuck are controlled by the strap cam 11, which also partakes of the quick and slow motion of the driving mechanism already described.

62 The speed and direction of the work spindles can also be changed by dogs (not shown) controlling the belt shifters 12.

63 Another example of the use of permanent cams and dogs is shown in Fig. 21, where the turret feed is operated by the permanent cam 1 and the cross slide by the cam 2. On this same cam 2 is a face cam operating, through a connection rod on the back of the machine, the gear segment 3 which in turn operates a back facing device. The spindle speeds are controlled by dogs on the disk 4. The revolution of the cam 1 is variable, being controlled by gears 5.

64 Still another illustration of this type is the Gridley machine shown in Fig. 3. Here block and strap cams are used for the stock and work feeds, the control of the spindle speeds and the operation of the chuck, while adjustable pins or dogs 6 are used to index the turret 5.

65 *c Controlled Entirely by Adjustable Dogs on Drum.* The machine shown in Fig. 1 has its movements controlled entirely by dogs on intermittently revolving drums, the only exception being the back facing device which is controlled by a permanent cam 8. For the turret the feeding is controlled by means of dogs 11 on the hinged carriage 12 engaging adjustable tripping blocks 13 on the turret, providing an independent tripping point for each tool of the turret.

66 It might be noted in this connection that devices for providing an independent stop for each tool of the turret have been in general use for many years as an automatic feature of hand operated machines. The typical form is a turret of adjusting screws, this turret being automatically indexed in time with the indexing of the main turret, thus bringing each screw which has been adjusted for that particular setting in line with a stop pin. Warner & Swasey machines illustrate this, as shown in Fig. 22.

67 *d Controlled by Cams Specially Formed for Each Job.* The advantages aimed at by this method are the securing of the ideal

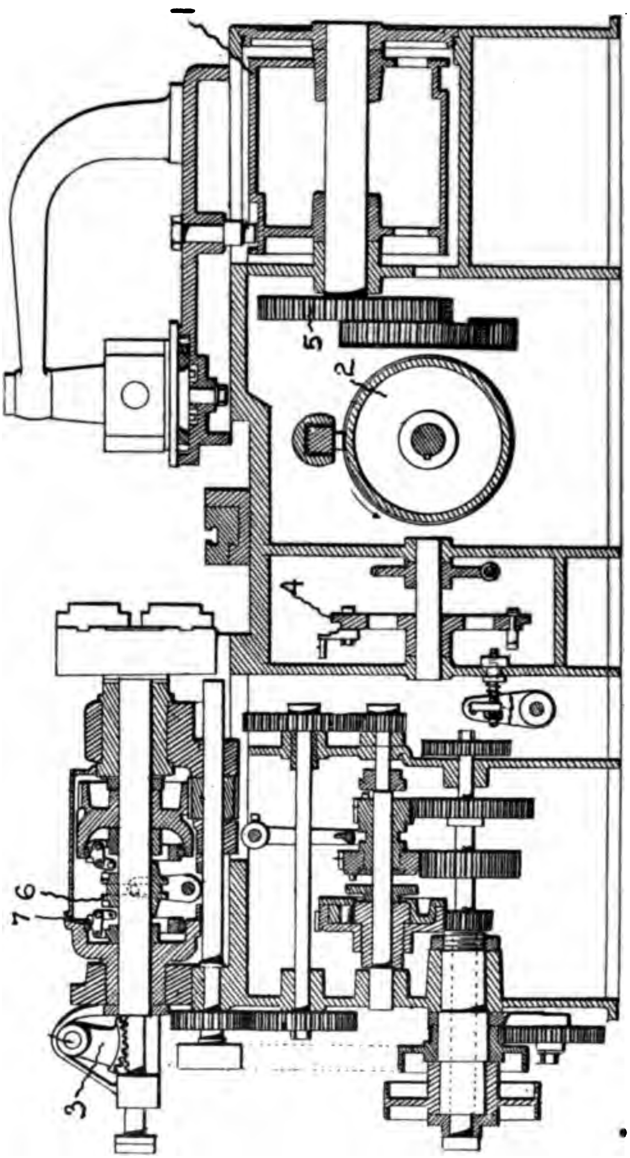


Fig. 21 Showing Control by Permanent Cams with Varying Speeds. Potter & Johnston Machine Co.

conditions as to rate of feed, etc., for each operation, and the minimum time for idle movements, and being able to duplicate readily these results for the same job at any future time, the cams being marked and preserved for this purpose.

68 The cam operated turret feed mechanism of such a machine is shown in Fig. 23. The advance feed is obtained by the cam 1 operating through the segment lever 2 to feed the turret slide 3. The return motion is accelerated by the revolution of the crank 4 bringing the turret back quickly an amount equal to the throw of the crank.

69 The cross feed slides, which are independent of each other,

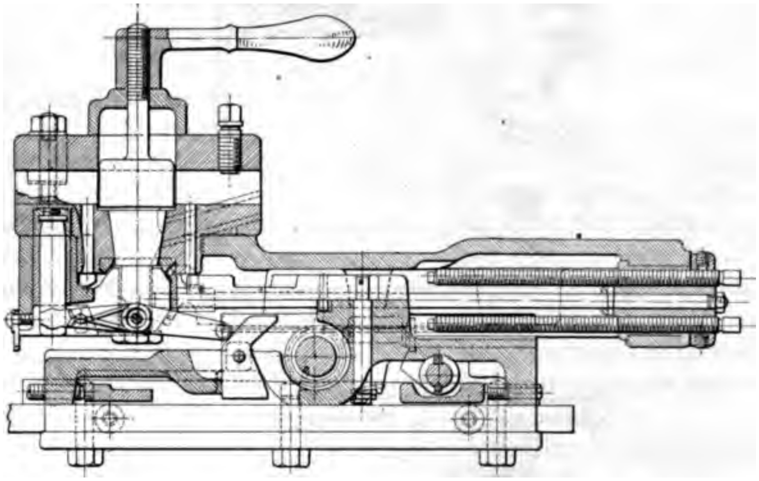


FIG. 22 INDEPENDENT STOPS FOR TOOLS IN TURRET. THE WARNER & SWASEY CO.

are also operated by special cams adapted to each particular job. These are shown in Fig. 7 at 10 and 11. The racks and segments by which the cross slides are operated are shown in Fig. 11.

70 In a machine for high speed work it becomes important, both in securing the desired speed and in avoiding objectionable shock, to move and reverse the lightest possible parts. For this reason machines having turrets of the "revolver" or "barrel" type, in which each spindle can be fed independently, are especially adapted to high speed work. In such machines each tool carrier is connected successively with a reciprocating feed slide, and only the feed slide with one of the tool carriers connected therewith requires to be reciprocated for the feed and return movements. Fig. 20 shows a machine

of this type. In order to "speed up" still further this type of machine, the use of an auxiliary slide has been resorted to, this auxiliary slide alone being moved during that part of the quick return movements required to retract each tool and even this being disconnected for the remainder of the return movement, thus avoiding the shock which would result from the rapid movement of these slides.

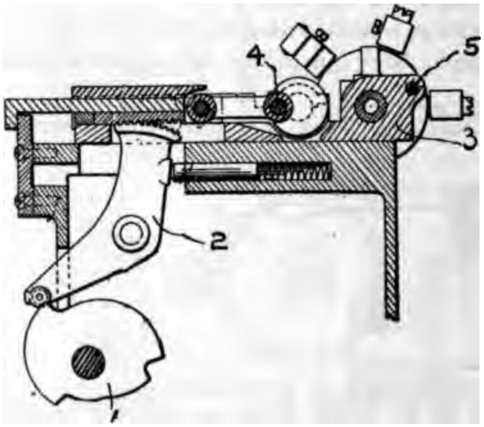


FIG. 23 FEEDING OF TURRET SLIDE BY CAM SPECIALLY FORMED FOR THE JOB—ADDITIONAL QUICK RETURN DEVICE. BROWN & SHARPE MFG. CO.

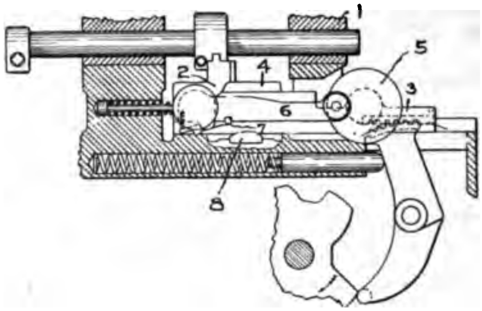


FIG. 24 DEVICE FOR OPERATING TURRET TOOLS FOR HIGH SPEED WORK. BROWN & SHARPE MFG. CO.

71 Fig. 24 shows an application of such an auxiliary slide with its disconnecting means. The turret 1 carries a series of tool spindles which are successively indexed to come into operative positions and be engaged by the block 2. A main slide 3 on which is an auxiliary slide 4 is mounted on the bed of the machine. A crank 5, also on the main slide, is connected to the auxiliary slide by the two-part

connecting rod 6, one end of which is connected with the main slide and the other with the auxiliary slide. A latch 7 holds these two parts together except when, during the quick return motion which is obtained by revolving the crank disk 5, the latch is disengaged by passing over the cam 8 and thus breaks the connection with the auxiliary or supplemental slide for the remainder of the crank throw which gives the quick return movement and the quick advance movement up to the point of cutting.

72 *e Mechanism Controlled by Permanently Set Dogs in Combination with Adjustable Stops.* This is a feature of the Bullard Mult-au-Matic vertical lathe, Fig. 25. This lathe has been classed by the American Machinist as a "station type machine" because the workman inserts and removes the work at one station or indexed position of the machine, while tools in the remaining positions are performing successive operations on other pieces.[1]

73 In this machine the rods 1 and 2 carry dogs which engage stops on the frame of the machine and trip respectively the advance and return feed movements. The quick traverse motion, which, in addition to retracting the tools quickly, can also bring them quickly forward to the point of cutting, is operated through gears 3 and 4 controlled by clutches. The advance feed for cutting is through bevel gears 5 and change gears connecting shafts 6 and 7, the worm 8 on the shaft 7 driving worm-wheel 9. These two trains of mechanism give the desired advancing and retracting movements through connection with screw 10.

74 A novel feature of this feed is in providing means for automatically tripping whenever the pressure on the cutting tool becomes excessive. This is accomplished by providing a thrust bearing for the worm 8 which, through bell clank 11, is held in place by adjustable weight 12. When the pressure is sufficient to raise the weight, the mechanism operates to trip the latch 13 and engage the return motion the same as if the regular tripping point had been reached.

IV INDEXING MECHANISMS

75 Under indexing mechanisms will be treated:

a Method of revolving turret
b Method of locking and clamping turret
c Rectifying the indexing

[1]For description of this machine, see American Machinist, vol. 40, no. 5, p. 177.

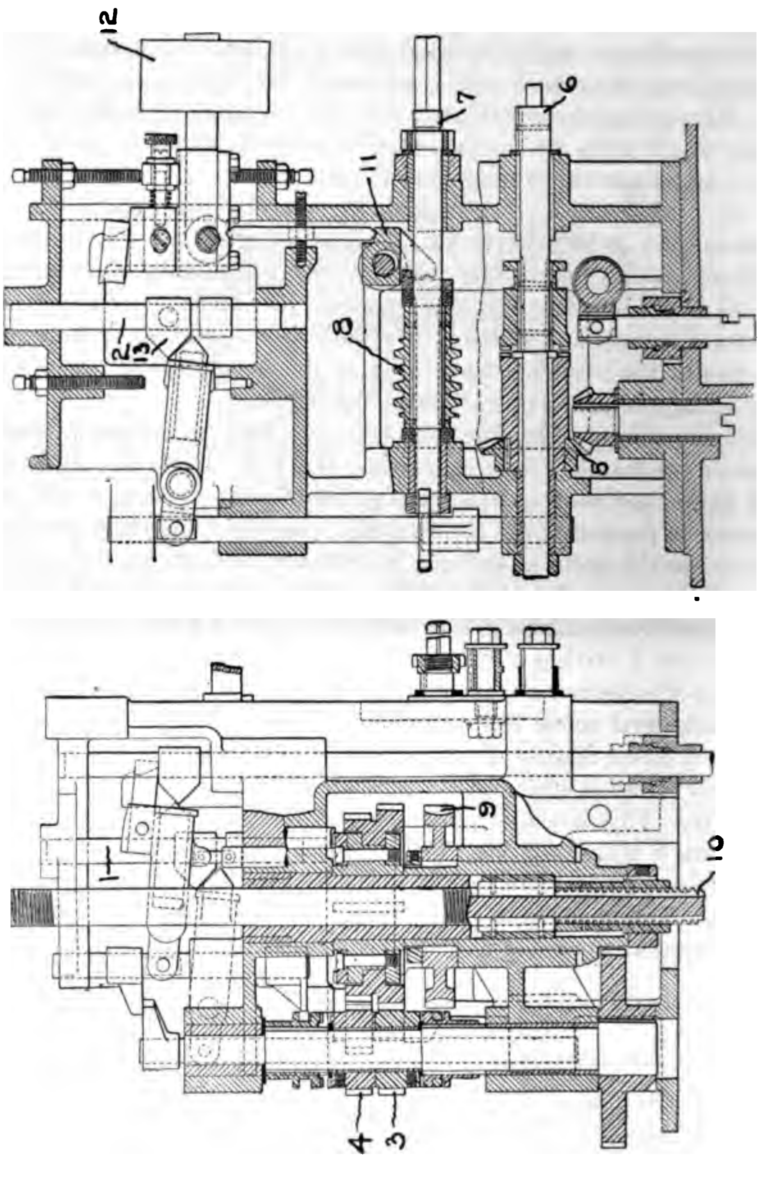


Fig. 25 Tool Device having the Feature of Tripping the Feed if Pressure on Tool becomes Excessive. Bullard Machine Tool Co.

76 *a Method of Revolving Turret.* A well-known method of lexing automatic turret machines is by the use of the principle the geneva stop. This has the advantage of giving a slow start-g movement gradually accelerating and slowing down before aching the stopping point, thus securing rapid indexing and at the me time avoiding shock. An illustration is shown in Fig. 8 where

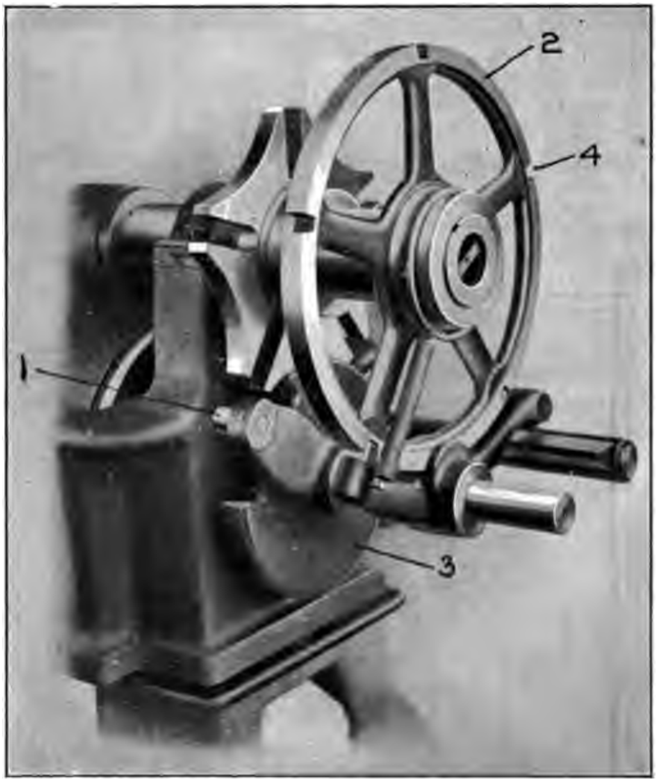


FIG. 26 DEVICE FOR INDEXING AND LOCKING TURRET. NEW BRITAIN MACHINE CO.

e turret 10 is indexed by the engagement of a pin 11 in the slots 12, e disk 13 carrying the pin being intermittently revolved on the aft 14.

77 When it is not required to use all the indexed positions of the rret, the idle positions can be passed by automatically tripping the dex as many times as there are idle positions to pass.

78 The geneva stop principle is also used in the indexing device,

Fig. 26, where the pin 1 engages the slots in the disk 2 to index the turret. The cylindrical portion of the pin carrier 3 engages concave portions of the disk 2 to hold it in approximate position, the locking being done by engagement of pawls in notches 4.

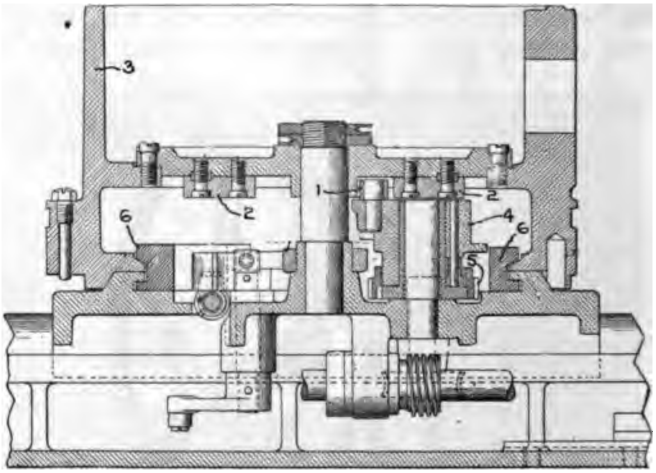


FIG. 27 INDEXING AND CLAMPING DEVICE FOR TURRET. GISHOLT MACHINE CO.

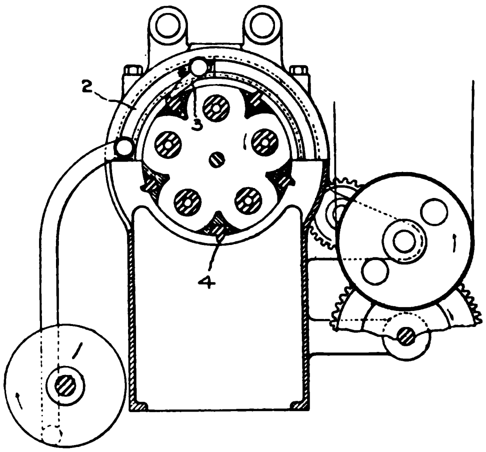


FIG. 28 INDEX FOR TURRET OF MULTIPLE SPINDLE MACHINE. DAVENPORT MACHINE CO.

79 Fig. 27 shows another application of the geneva stop, in which the roller engages slots formed between blocks 2 for indexing the turret 3. Roller 1 is carried on a sleeve 4 which is intermittently turned by gear 5.

80 Another method securing the same advantage of slow starting and stopping is shown in Fig. 28, where the revolution of the crank 1 operates the curved sliding block 2 on which is mounted the pawl 3 engaging teeth 4 on the turret.

81 *b Methods of Locking and Clamping the Turret.* It has been

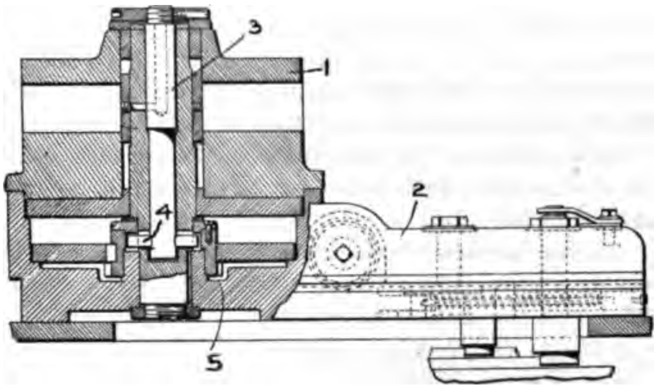


FIG. 29 CLAMPING DEVICE FOR TURRET. POTTER & JOHNSTON MACHINE CO.

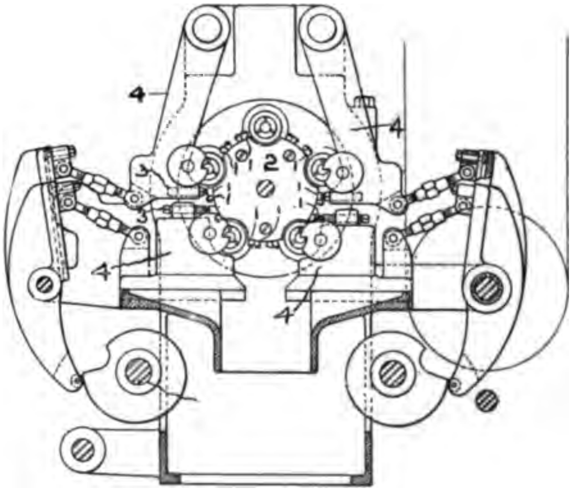


FIG. 30 MEANS FOR RECTIFYING THE INDEXING OF MULTIPLE SPINDLE MACHINE. DAVENPORT MACHINE CO.

the general practice in turret machines to have a locking pin to automatically engage to insure the accurate alignment of the turret with the spindle and to hold the turret firmly in position while the tools are operating. Such locking pins are shown at 5, Fig. 19, and at 5, Fig. 23, and another method employing notches in a disk of comparatively large diameter is shown in Fig. 26. In addition to the use of a locking pin, a further clamping device is often used which automatically clamps the revolving turret securely to the slide or bed and unclamps it before the next indexing.

82 An example is shown in Fig. 29 where the turret 1 is clamped to the turret slide 2 by the central stud 3 which is forced downward by a camming action on the projecting lugs 4. This camming action is produced by the revolving of gear 5, this gear in turn being acted on by a rack controlled by dogs acting at the desired point of the return motion of the turret slide. The locking pin for the turret is also operated from this same rack, and thus is timed with the clamping device.

83 Another method of clamping is shown in Fig. 27 in which the shoes 6, by means of a wedging action, hold the turret rigidly to the slide during the cutting operations.

84 *c Means for Rectifying the Indexing.* In multiple spindle machines it is more difficult to secure accurate indexing and thus produce accurate work than in single spindle machines, because of the mechanical difficulties of constructing the machine. It is difficult to bore the spindle carrying head and mount the spindles in it so that they will be equally spaced and equidistant from the centre. To overcome such inaccuracies as are due to this cause, rectifying stops have been employed as shown in Fig. 30, one for each cross tool carrier. These consist of a series of pins 1 projecting from the disk 2, which disk is secured to the front end of the spindle head. These pins are engaged by coöperating stops 3 on the tool carriers 4 and are made so that the cutting edges of the tools carried by the tool carriers 4 will be exactly the same distance from the centre of each spindle when the pins and their coöperating stops are engaged, thus tending to counteract the inaccuracy above referred to, or any which may result from wear in the machine.

V CONTROLLING MEANS FOR THE VARIOUS MECHANISMS

85 Aside from the ordinary practice in directly controlling the various movements of an automatic turret machine, Mr. Flanders in his paper has pointed out that the use of a controlling shaft or, as he terms it, a "lay shaft," constitutes a separate type of control.

86 A machine with this type of control is shown in Fig. 8, where the shaft 15 drives the various mechanisms of the machine, except the spindle, and by means of clutches controls their operations.

87 The shaft 7 in Fig. 1 may also be said to be another type of controlling shaft, as through the interposition of dogs and levers it controls the various operations of the machine although it does not drive them. It is an application to what Mr. Flanders terms the screw feed type of machine. The Bullard machine, Fig. 25, is also of this type.

NOTE

88 It will be understood that in the limits of such a paper as this it is not possible to describe all makes of machines and all successful mechanisms that have been devised for performing the various functions here discussed. It has been necessary to select a few only and the selections have been made as far as possible of those which are typical and in the main of such as are in actual use.

DISCUSSION

Ralph E. Flanders. One of the things in favor of the hand machine for large work is that with it higher speeds and feeds can be used in many cases than in an automatic machine. On an automatic machine the speeds and feeds have to be kept down to such a point that nothing serious will happen while the operator is away, attending to one of his other machines. The operator running the hand machine has his eye on the work constantly, and it is safe to run at speeds and feeds which are not safe for an unattended machine.

Furthermore, at times the automatic machine has to be so set that the slow feeding movements will start a reasonable length of time before the tool actually strikes the work, losing a little time there. This does not mean that the automatic machine will not occupy a larger field every year in certain kinds of work.

One further type of mechanism should be added to the classification of spindle drives. This is the type where a change is made in the relative movement of work spindle and tool carrier. Each one of the author's speed classifications comes under this head of change in the relative movement, strictly speaking; but the particular application I have in mind is used, if I remember correctly, on the Gridley multiple spindle automatic. In that machine the tap or die may be run at a slower rate of speed than the work for threading on, and then at a faster rate of speed than the work for threading off, without changing the rate of speed of the spindle. Perhaps, therefore, this feature does not come in the classification of spindle drive at all, though it has the effect of some of the other methods Mr. Burlingame has mentioned under this classification.

ELMER H. NEFF. Mr. Burlingame's paper would appear to use the word "control" with a different interpretation from the paper presented by Mr. Brooks. The latter has used the word "control" in the ordinary sense relating to the method of conveying to the machine tool its power for operation, while this paper covers the field of devices applied to screw machines and lathes for automatically performing the work for which they are designed. In other words, it relates to the details of design of the tools. A great deal of interesting information has been collected to cover the elements of the machines as listed in the paper.

Referring to the paragraphs on speed change, I would suggest as a still stronger reason for having incorporated in an automatic screw machine the possibility of changing its spindle speeds automatically during the series of operations going to make up a finished piece, that the presence of this possibility enables the designer of the tools to secure a larger output from the machine. The reason for this larger output is that some pieces of work, especially in castings, will have cutting operations on diameters quite different in size from each other, so that if the small diameter has to be machined at the same rate of rotation as is demanded by the limitations of cutting speeds on the large diameter, the product of the machine is cut down to a low ebb. A further illustration is in the fact that threading must be done at a relatively slow speed as compared with turning operations.

In former years it was a common idea among those who were prospective purchasers of automatic screw machines, particularly among those who had not used such machines, that a cheaper class of labor could operate them. Experience has shown this idea to be a fallacy. As a matter of fact, if there is any difference at all, it requires a higher degree of skill to operate an automatic screw machine than it does a hand screw machine. The skilled operator attending from four to seven or more machines is sometimes assisted by a helper who can slip a bar of stock through the feeding finger, and throw the starting lever, but that is not operating the machine. These screw machine operators are not necessarily machinists by trade, but they should be highly skilled specialists in this particular work. The statement with regard to greater skill required to keep the machine in running condition is correct, and evidently is contrary to the statement that less skilled labor can be applied.

Automatic screw machines are successfully performing the work indicated by the following classification:

a Producing pieces from bar stock (bar brass, machinery steel, drill rod, etc.)

b Second end operations on pieces produced by classification *a*. In this section the work is fed into the chuck automatically from a magazine or hopper.

c Machining operations on blanks such as punchings and small castings. The work is automatically fed into the chuck in such cases usually from a chute, the only requisite being that if they are castings they shall not vary in roundness or size more than the elasticity of the chuck will allow.

d Performing the machining operations on pieces inserted by hand in the chuck. These are usually castings of such irregular shape or size that they cannot be handled or inserted from a chute automatically.

I have recommended and installed a multitude of automatic machines during the past 18 years, covering operations along the classifications suggested. I cannot agree with Mr. Flanders in his discussion of this paper that automatic machines must run at a slower speed than hand machines unless his statement should be limited in its application to castings. The machining operations on castings are liable to be considerably curtailed in speed on account of the variations in their hardness, and also on account of the liability that they may have incorporated in them slag or other hard substances that will destroy the cutting tools, and spoil considerable work, before the operator discovers the injury to the tools.

There is no direction in which the general machine tool industry is growing more rapidly than in the field to which automatic screw machines as analyzed above can be applied. The reason for this growth is readily understood by those who have used automatic screw machines because their installation, almost without exception, has shown very large savings in cases of producing manufactured goods. At the same time it should be noted, in passing, that practically all the gears used in this country are produced on automatic gear cutting machines. Also that the cylindrical grinding machine, which has developed very rapidly, is a semi-automatic machine, that is, one which produces automatically in many cases the finished operation after the piece has been inserted in the machine by hand. Furthermore, some progress has been made in the development of completely automatic grinding machines, which insert their work automatically from a chute.

Norman Marshall inquired whether it was necessary to install automatic stopping devices on these machines to take care of accidents, and what was the state of their development.

H. K. Hathaway said that the contrast between what has been done in the way of automatic control in machine shops and in other lines of industry, has struck him very forcibly in the last few years.

In the textile and other allied industries, particularly, there has been much done in the way of the automatic stopping of machines when anything goes wrong, as well as other features of automatic control. In machine shops, however, the use of automatic control has, comparatively, been very limited.

The Author. As to the differential rate of speed of spindle and work, which Mr. Flanders mentions, it is interesting to note that the patent records show that practically every imaginable combination of running forward and backward, starting and stopping, has been patented at some time, probably without the claims being of any great value. All combinations for control of spindle speeds might be classed under one heading, whether they are for stopping the spindle entirely or for slowing it down thus giving a differential speed.

There are some factors of automatic control to which study has been given but which are not included in this paper. One is the supplying of oil to the point of the drill, in a turret machine, so that the oil is shut off as soon as the turret is indexed to another position, and restarted whenever the drill is in line.

I agree with Mr. Neff that the use of automatically controlled machines increases the need of skilful supervision. In fact this statement is made in the paper. It is simply that less skilled men can be used as helpers on automatic machines; but there must be somewhere up and down the line someone with a high degree of skill for the purposes of supervision.

Men who are not fully trained mechanics can sometimes be instructed so as to become skilful in operating automatic machines. These are men adapted to this work, who become skilled in a sense, although not classed as skilled "all around mechanics." I do not wish to convey in the paper, however, any thought but that skilled supervision and skilled men are needed to operate these machines.

In reply to Mr. Marshall, it is a plan in many types of machines to have a breaking or friction point which will break or yield first and thus prevent breaking any of the important parts. The machine so constructed can after failure be readily started up again without waiting for expensive repairs. This feature might be very aptly classed as one of automatic control, but not in the usual sense in which the mechanism for operating the machine is meant.

On the Bullard machine (Fig. 13) is a thrust for the worm which will allow it to give if the pressure is exceeded, and will trip the machine so as to stop its feeding. This illustrates in another way the point made by Mr. Marshall.

No. 1522

ELECTRIC OPERATION AND AUTOMATIC ELECTRIC CONTROL FOR MACHINE TOOLS

By L. C. Brooks, Schenectady, N. Y.
Associate-Member of the Society

The application of electric control to individual machine tools is considered to be one of the most important forward steps in the improvement of machine shop efficiency. The economies resulting from substituting electric drive (either individual or group drive) for the steam engine and long line shaft drive were very effectively outlined before this Society six years ago.[1] Prof. W. F. M. Goss too has stated: "I am convinced that the machine tool of the future is to be an individual motor driven machine, a machine in which we shall not see pulleys, belts or gears." This prophecy is surely being fulfilled.

2 Six years ago it was pointed out that for work requiring adjustable speed motors, the direct current motor alone was applicable. The same is true still, if the desired speed increments are small. However, very satisfactory results are being obtained with alternating current in large installations in steel mill and other heavy duty work of large capacity.

3 In the early application of electric motors to machine tools, the motors were started by hand starters, either of the dial, or drum type, the dial type usually being unprotected. Some of us vividly recall the scene of an aged joiner attempting to start a high speed machine, how as the starting arm of the rheostat was near the running point, his trembling was so great as to cause the starting arm to break contact. The result—a heavy arc on the starter and a burned hand on the operator. However, these conditions are now entirely

[1]Trans. Am. Soc. M. E., Vol. 32, p. 219; also, Proc. A. I. E.E., Vol. 29, p. 621.

Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the Sub-Committee on Machine Shop Practice.

reversed by use of Safety First requirements and remote controlled automatic controllers.

4 During the past year, the American Institute of Electrical Engineers has held a series of meetings under the auspices of the Industrial Power Committee. The subjects treated included the characteristics of electric motors; factors involved in motor applications; fields of motor application and controllers. These articles contained a vast amount of valuable data and information relative to their subjects and to present practice, and it is understood that they will be published in a special volume. All who are interested are referred thereto, as any attempt to give an abstract here would be unsatisfactory.

CONTROL APPARATUS

5 The importance of control appliances can best be emphasized by quoting a very prominent machine tool builder: "In analyzing the various elements of machine tools, the question naturally arises, wherein can they be improved? There are many ways. First, let us consider the question of control. Control is the essence of machine tool operation. It plays the largest part in production work. Control is everything . . ."

6 While there are a number of instances and special locations where dial or drum type hand starters are more applicable, we have reached the time when automatic starters, remote controlled, (with the possible occasional exceptions of the reversing switch) are the most suitable. The principal advantages of automatic control are:

a The use of manually-operated controllers may cause undue stresses on the motor, especially on rapid reversing equipments

b The operating switch, or push button, is easily attached to the machine and the main panel may be located at a distant point, out of the way

c The manufacture of automatic appliances is now a well founded art and is no longer an experiment

d A considerable increase in the capacity of the machines is obtained

e The starting time is automatically regulated to suit the load conditions on the motor

f Accurate stopping points are obtained by the application of *dynamic braking*, which consists in connecting a resistance across the armature circuit in the *off* position of the starter, the stored energy of the armature being dissipated as heat in the resistance

g It permits the use of operators not specially trained. It entirely removes the element of thought from the mind of the operator who has simply to operate the master switch or push button in the desired direction and does not need to think about drift points, safety features, etc.

TYPES OF STARTERS

7 The three general types of automatic starters are:

Time element, wherein the starting resistance is cut out in a fixed period of time. This type is not adapted to conditions of varying starting load.

Counter E.M.F., wherein the starting resistance is cut out by

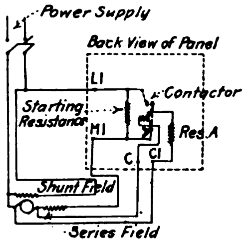


Fig. 1

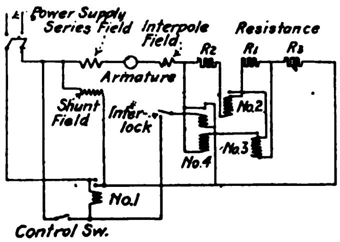


Fig. 2

Fig. 1 Connections of C. E. M. F. Automatic Starter. Line Switch in Motor Circuit

Fig. 2 Connections of Series Type Starter

magnets actuated by the counter e.m.f. of the motor. The reliability of this system is decreased by the fluctuation in line voltage, except in the case of motors of small capacity, though it has worked very well in elevator work. Fig. 1 shows a connection of a counter e.m.f. starter with one step of starting resistance. The contactor coil is connected in multiple with the motor armature and is so designed that the contactor will not close until the counter e.m.f. of the motor has built up to a predetermined value.

Current limit, wherein the starting resistance cut out is regulated by the motor current. The current limit type of starters are of two kinds: *First*, with shunt wound magnet switches, actuated through series relays which will not close until the accelerating current has reduced to a predetermined value; *Second*, with magnetic lock out switches (or series contactors). These series type switches consist essentially of a series wound coil actuating the switch, the coil being in the motor circuit. The magnetic path in the switch may be so adjusted that the switch will not close until the current has reduced below a certain critical value.

8 A simplified wiring scheme of a series type of starter is shown in Fig. 2. The explanation of this diagram is:

FIG. 3 STARTING PANEL FOR SMALL ADJUSTABLE SPEED MOTOR, WHEN STARTING RESISTANCE IS NOT NECESSARY

Contactor No. 1 is shunt line contactor.

Contactor Nos. 2 and 3 are series accelerating contactors.

Contactor No. 4 is compound series accelerating contactor. with shunt holding coil and interlock in the control circuit of the holding coil.

Operation of the control switch closes No. 1 contactor; motor starts with all resistance in circuit.

Contactor No. 2 closes as current falls to some predetermined value, short-circuiting R1.

As contactor No. 3 closes, due to falling off of current, R2 is short-circuited; this contactor also short-circuits contactor No. 2 with its coil, causing No. 2 to open.

In the same manner contactor No. 4 closes, short-circuiting resistance R3 and dropping out contactor No. 3; in closing, contactor No. 4 establishes a circuit through the interlock for shunt holding coil, thus insuring that contactor No. 4 will stay closed until the voltage fails or the control switch is opened.

9 All panels, whether for general starting duty, or for control

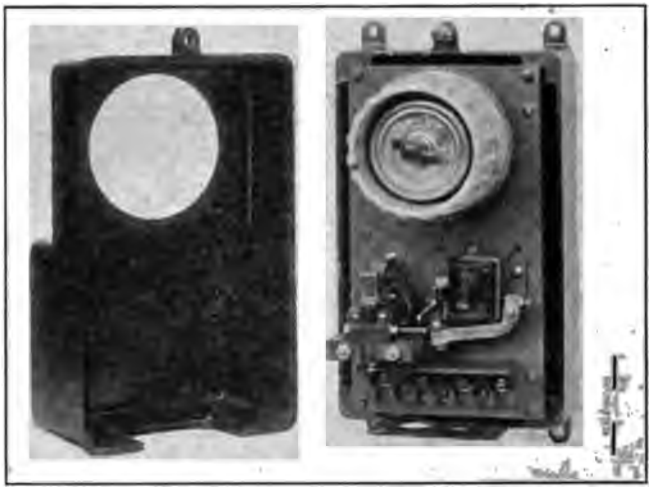


Fig. 4 Starting Panel for Small Adjustable Speed Motor, When Starting Resistance is Not Necessary

of special machines, should be provided with a suitable enclosing case, for the protection of the appliances on the panel against injury, also for the protection of the operator against accidental contact with current carrying parts. With starters for motors of small capacity, the enclosing case should be fool-proof and meet all the applicable Safety First requirements. Under ordinary conditions, the control panels for motors of large capacity should be provided with small openings to allow for the radiation of heat from the current carrying parts. In many cases, it will be found very desirable to make the enclosing case as a part of the casting of the machine frames.

STARTERS FOR CONSTANT SPEED MOTORS

10 A starter for small motors of ¾ h.p. and less, where starting resistance is not necessary, is shown in Figs. 3 and 4. It consists of a line contactor, an overload relay, a field rheostat and connection board, mounted on the insulating base and enclosed in a Safety First case, which is adapted for conduit wiring. This starter is arranged to always start with full field on the motor by simply turning the field rheostat handle to the extreme left, thus closing the line contactor, after which the rheostat handle may be turned to the point for the desired motor speed. If desired, an "emergency stop" push button may be located at any convenient point on the

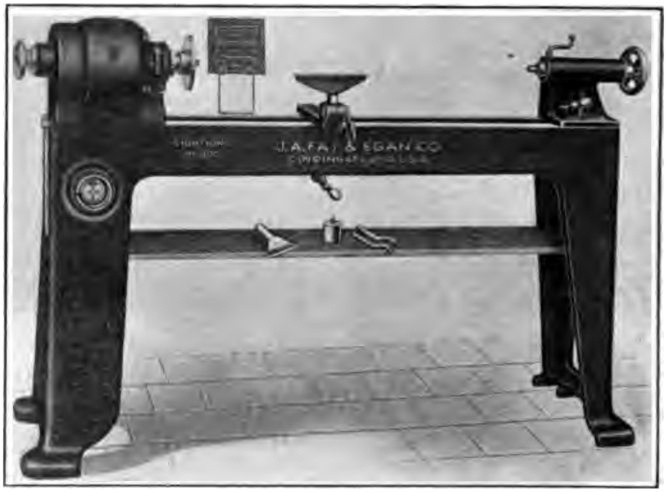


FIG. 5 WOODWORKING LATHE WITH STARTER IN HEADSTOCK

machine. This starter is especially adapted for headstock motors, wood-working machines, etc. Fig. 5 shows a wood-turning lathe with the starter self-contained in the lathe frame.

11 A starter for motors of ¾ to 3 h.p., where one step of starting resistance is necessary, is shown in Figs. 6, 7, 8 and 9. The appliances on the front of the panel consist of line switch and fuses, (1) contactor, (1) counter e.m.f. accelerating contactor, and connection board. The starting resistance is mounted on the back of the panel. The complete panel is enclosed in a Safety First case, the chief features of which are:

a The switch operating handle may be locked in position, thus preventing unauthorized operation *b* A hinged door is provided for the examination of the panel and renewing the fuses

c There is an interlock between the operating handle and

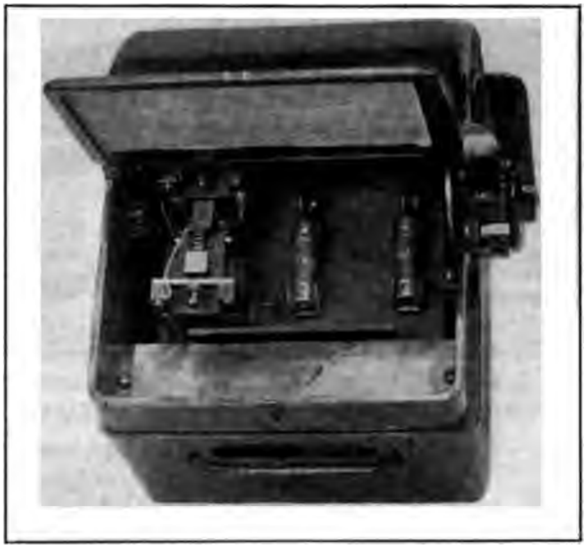


FIG. 6 D. C. STARTING PANEL. MAX. RATING 3 H.P. AT 230 VOLTS

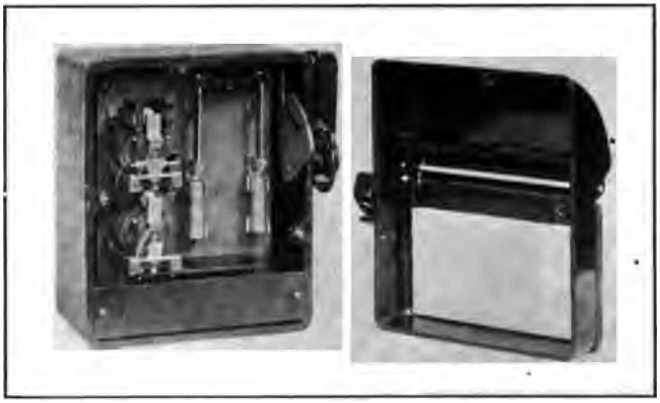


FIG. 7 D. C. STARTING PANEL. MAX. RATING 3 H.P. AT 230 VOLTS

the cover of the enclosing case, so that the cover cannot be opened until the switch is open and the switch cannot be closed until the cover is closed, thus preventing any injury to the operator as a result of accidental short circuits. This starter is arranged for push button operation and the enclosing case may be fitted for conduit connection.

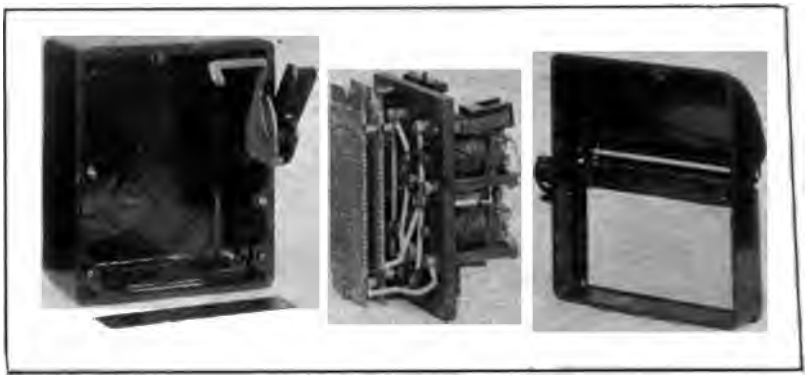


FIG. 8 D. C. STARTING PANEL. MAX. RATING 3 H.P. AT 230 VOLTS, SHOWING STARTING RESISTANCE ON BACK OF PANEL

FIG. 9 D. C. STARTING PANEL. MAX. RATING 3 H.P. AT 230 VOLTS

12 For starters of larger horsepower, arrangement *c* would be somewhat modified to accommodate the additional accelerating

contactors. A double pole line contactor should also be supplied. The questions of type of overload appliance and whether or not a dynamic braking contactor is necessary depend upon the application. The enclosing case should be of sheet steel, for appearance and light weight, and the interlocking features described above are not necessary.

STARTERS FOR ADJUSTABLE SPEED MOTORS

13 For general service, the starters for adjustable speed motors should be of the same general design as described above for constant speed service. The appliances on the panel should include:

- Double pole line contactor
- The necessary accelerating contactors
- Overload and no-voltage protection
- Field accelerating relay
- Provisions for starting the motor with full field
- Provision for disconnecting the motor fields when the machine is not in use
- Dynamic braking contactor, when required
- Protected terminal connection board.

14 The starting resistance should be mounted back of the panel, except in the case of large sizes of motors when it should be separately mounted. The field rheostat should be mounted separate from the panel, with the possible exception of small sizes of motors. The method of control should be at the panel or remote, as operating conditions require.

REVERSING SERVICE

15 For general reversing service, the starters should be of the same general design and requirements as given above for non-reversing, with the following modifications:

a Dynamic braking at the "off" position should be supplied to protect the motor

b Two double line contactors are necessary to give the reversing. In small equipments, a double pole, double throw switch may be used for reversing.

GENERAL SERVICE

16 The types of starters described above are applicable for all general service, as pumps, fans, drill presses, grinders, milling machines, and boring mills, except in cases where special features are desired.

PROTECTIVE FEATURES

17 All automatic starters and control apparatus should be provided with protection from low voltage on the line, also from excessive overload. In all cases, except possibly pumps and fans and similar machines, these protections should be of such a nature as to disconnect the motor from the line, and not to be restarted except by the operator at starting station. For motors of over 25 h.p., the overload protection should be preferably an overload relay, either hand or electrically reset, as operating conditions require. For motors below 25 h.p., the overload protection should be fuses, chiefly for economic reasons. The fuse capacity should conform to the requirements of the National Board of Fire Underwriters.

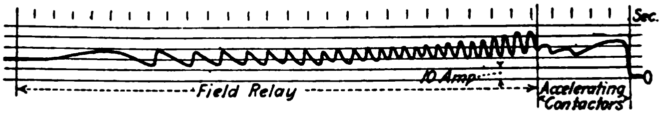


FIG. 10 CURRENT CURVE, STARTING A 5 H.P., 500/1500 R.P.M. MOTOR WITH FLYWHEEL LOAD

In many cases it will be preferable to have the line switch and fuses a part of the distribution system, separate from the control panel.

18 With adjustable field motors, it is desirable to set the field rheostat so that the motor will always operate at a predetermined speed. To accomplish this, connections should be such that the field resistance is short-circuited during the accelerating period, thus permitting the motor to start under maximum torque. (This connection to be re-established when the motor is stopped, thereby giving maximum torque for the dynamic brake cycle). After the armature resistance is cut out, the field is then automatically weakened to the predetermined point. In order to prevent the motor from accelerating too fast, thereby drawing excessive current from the line, the field accelerating relay should be supplied. This relay is connected in the armature circuit with auxiliary contacts which, when closed, short-circuit the field rheostat and as the relay operates alternately cut the field rheostat in and out as the motor increases from full field speed to the desired fast speed, thereby preventing the high current inrushes on the motor. This method also permits of quickly obtaining full speed and prevents severe field distortion, due to high armature currents. These relays are unnecessary on motors below 5 h.p., and on motors of less than 50 h.p., with a speed range of 2:1, except in special cases where the

time of bringing acceleration up to speed, with 150 per cent load, is many times that of the field constant.

19 Fig. 10 is a current curve when starting a 5 h.p. motor, showing the starting peaks, also how the field relay regulates the current.

20 With adjustable speed motors of a range of greater than 2 to 1, provision should also be made so that the field resistance is not cut in until the last accelerating contactor has closed, thereby cutting out the last step of starting resistance. Provision should be made so that the motor fields are not energized when the machine is not in use.

21 With adjustable speed motors, too, especially with a range of 4 to 1, arrangements should be such as to prevent a "pump back" when stopping the motor.

22 On work requiring quick reversing, dynamic braking should be provided at the "off" position of the starting switch. In many cases, as on planers and boring mills, a graduated dynamic braking of two or more points is necessary.

CONTROL APPARATUS APPLIANCES

23 The individual appliances of control apparatus should be of the best possible quality applicable to the work of long life, and of relatively strong, rugged construction. By this is meant, that with a contactor of 300 amperes capacity, it is reasonable to expect 1,500,000 or more operations without renewal of parts, while with a contactor of 25 amperes capacity, 500,000 operations would be satisfactory due to the relative lightness of construction and small parts.

24 The design of bearing pins and surfaces should be such as prevent "freezing" and produce minimum wear.

25 The appliances should operate satisfactorily under working conditions, with a voltage variation of 5 per cent above and 20 per cent below normal. They should also "wipe in" at or below the "pick up" voltage.

26 The temperature rise on contacts and coils should not exceed 65 deg. under continuous operating conditions.

27 All shunt wound coils should be treated with a waterproofing process.

28 Auxiliary contacts should have a positive action, to insure good contact and prevent accumulation of dirt.

29 In many cases where overtravel would be disastrous, as on

planers and slotters, arrangements should be made to be sure the line contactors will open when the limit switch or master controller operates to cut off the supply of current.

RESISTANCES

30 In connection with starting apparatus, one of the most important items for consideration is the starting resistance. Except in cases of small motors, cast iron grids are very satisfactory for this service. Cast iron has a temperature coefficient of about 0.0007 and by adding a small per cent of certain alloys, a very high resistance grid is obtained which is suitable for rather small size of motors. For small motors, a high resistance wire, or ribbon, is suitable, which may be in the form of enclosed coils, flat plates or other special forms as the various manufacturers develop the material. A motor starting a load which is subject to infrequent overloads of short duration should have a resistance unit of large thermal capacity, while a motor subject to frequent overloads should have a resistance unit of small thermal capacity in order to give more rapid

TABLE 1 NUMBER OF STEPS OF STARTING RESISTANCE

Horsepower of Motor	Below 1	1–3	5–25	35–50	60–100	110–200
Number of Steps	Directly on Line	1	2	3	4	5

cooling. In all cases, the resistance units and their mounting should be of rugged construction and properly supported to be conveniently accessible.

31 The starting resistance should be so laid out in the various steps as to give practically uniform acceleration peaks. In certain cases, as fans, where the load increases with the speed, it may be desirable to have the transition points (valley currents) tapered, the low points being at the beginning, thereby requiring less starting steps, and reducing the peak demands from the line.

32 The National Electric Light Association has proposed the following requirements for direct-current motors, as to maximum starting current, for 230 volts, which is the average factory voltage:

3 h.p., and below......12 amperes per h.p.
Above 3 h.p........... 9 amperes per h.p.

No motor may be connected without a starting resistance where the starting current exceeds 30 amperes.

33 Table 1 gives the number of steps of starting resistance which should be required for general service. Of course, there will

be instances where it will be desirable to depart from this schedule, to meet peculiar operating conditions.

APPLICATIONS OF AUTOMATIC CONTROL

34 With the foregoing general requirements for appliances, and with the following typical special examples described, it is not hoped to have universal agreement. In fact, it is hoped that the criticisms will be wholesome and that other suggestions will be numerous. All the data thus obtained would form a desirable basis for any future standardization of electrical equipments for machines of the various services.

35 The importance of standardizing cannot be too strongly emphasized. By this is not meant that the detail appliances should all be of the same design—the various electrical manufacturers can work that part out for themselves—but that the control equipment for a certain type of machine should contain a uniform set of appliances. The space required by every electrical manufacturer would thus be practically the same and the machine could be so designed to accommodate the electrical equipment, with the result that the complete installation would be much neater in appearance and more satisfactory in every way.

36 *Lathes.* At the present time, the problem of *purely automatic control* for general lathes has not been entirely solved. For many uses, the drum controller has been found to fill the requirements quite satisfactorily. An automatic starting panel with a drum switch for reversing has given good results for certain applications. Lathes for a special class of work have been controlled with safety and efficiency, so that it is believed the day is not far distant when the goal for general lathes will have been reached.

37 *Car Wheel Lathes.* The functions of a car wheel lathe require that electrical equipment be designed for especially heavy service and that the control be reliable and as simple as possible. The equipment should consist of the panel and resistances, a push button station with "start," "stop" and "slow down," and a pendant or foot switch for "slow down," the "slow down" feature being necessary when a hard spot is reached in the cut, and "slow down" being to approximately 50 per cent of basic speed at 100 per cent load. After the "slow" button is pressed, it should always be necessary for the motor to come to approximately full field speed before it is possible for slow down contactors to be closed again, to limit the "pump back."

38 The panel requirements should consist of the following:

One line contactor
The necessary accelerating contactors
Two dynamic brake contactors
One field accelerating relay
Double pole, double throw line switch for reversing.

39 The starting, brake, and slow down resistance should be mounted on the back of the panel, when the size will permit, the field rheostat being mounted separately. The enclosing case should

FIG. 11 WHEEL LATHE CONTROLLING PANEL. 200 AMPERE SIZE

be of sheet steel and provided for either wall or floor mounting. A panel for 50 h.p. is shown in Fig. 11.

40 *Boring Mills.* The control equipment for a boring mill presents many interesting problems, among which are:

- *a* Where the mill is operating upon heavy castings, it should be accelerated slowly on starting
- *b* Dynamic braking should always be supplied when stopping to insure efficient stops, also to prevent damage to the motor by improper manipulation of the starter
- *c* The control scheme must be such that no possible com-

bination of operating buttons will permit the motor to run on starting resistance when the load is decreased, or produce a "pump back" with consequent heavy sparking at the motor when stopping.

41 The type of control panel necessary to give most ideal operation on a boring mill consists of a straight series-type current limit starter, with provision for maintaining full field on the motor until the last point of armature acceleration is reached. A field accelerating relay should be used when adjustable speed is desired. Dynamic braking should be supplied in all cases, the resistance being adjusted for full load current at high speed. The panel should be controlled by a "start" and "stop" push button; on very large machines a pendant switch is also desirable to be used when setting up the work. An 18-ft. boring mill in operation is shown in Fig. 12 and the control panel is shown in Fig. 13.

42 *Planers.* The applications of electric motors and their control to planers, slotters, etc., is probably the most interesting, from an engineering point of view as well as that of economy in production, of all machine tool applications. The relative advantages and economies of the reversing motor drive for this purpose are now fully realized and have been set forth in bulletins issued by the various electrical manufacturers. They are also thoroughly appreciated by all machine tool men, so that a summation of them will be unnecessary. However, the main electrical points to be kept in mind are:

a Sparkless commutation of the motor
b Stability of the motor at all speeds
c Gradual dynamic braking (to prevent undue shock on the machine) in the shortest possible time
d Quick reversing
e Independent cutting and return speeds with maximum range of 4 to 1
f Provision for minimum drift when the motor is stopped or reversed, also when the power fails as a result of overload or low voltage on the line
g Auxiliary contacts on the contactors reduced to a minimum
h Provision for position operation of the main contacts
i "Time efficiency" for the complete cycle as high as possible
j All the appliances of the control equipment designed so as not to be influenced by reasonable vibration.

43 Fig. 14 shows a 72-in. planer that was changed from a belt

FIG. 12 WILLIAM SELLERS 18 FT. BORING MILL OPERATED WITH 7½ AND 50 H.P. MOTORS CONTROLLED BY PANEL. *A* IS RAPID TRANSVERSE AND FEED CONTROL STATION, *B* IS MOTOR PANEL AND *C* IS MAIN MOTOR CONTROL STATION

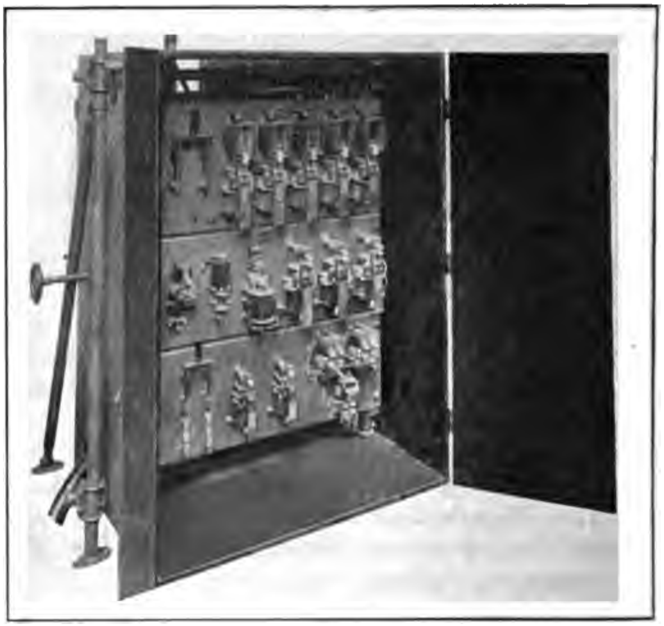


FIG. 13 CONTROL PANEL FOR 50 H.P. AND 7½ H.P. COMBINATION BORING MILL MOTOR

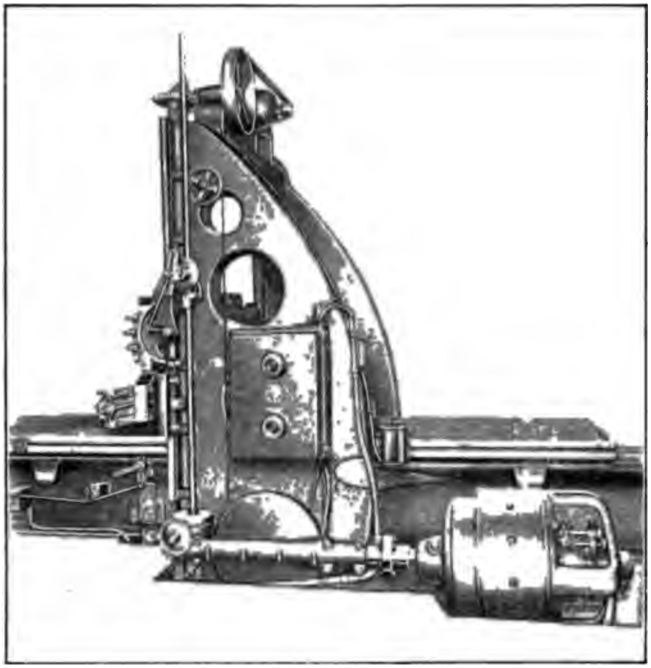


FIG. 14 REVERSING MOTOR DRIVE. 72-IN. BY 22-FT. BEMENT MILES PLANER. DRIVEN BY 35 H.P., 250/1000 R.P.M., 230-VOLT MOTOR

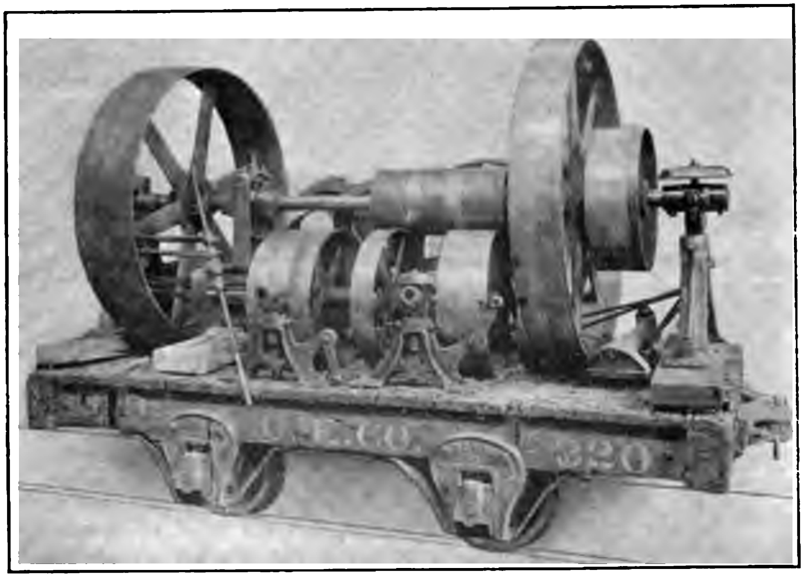


FIG. 15 MATERIAL DISCARDED FROM 72-IN. BEMENT MILES PLANER WHEN CHANGED FROM BELT DRIVE TO DIRECT CONNECTED REVERSING MOTOR DRIVE

drive to electric drive, and the discarded material is shown in Fig. 15. Fig. 16 shows a 25 h.p. control panel. Figs. 17 and 18 show a complete machine. Fig. 19 shows the results that may be obtained from a machine operated by this type of equipment.

44 Fig. 20 shows a diagram of connections of a planer control equipment. Contactor No. 6 is the accelerating contactor. Contactor No. 8 is 2nd point dynamic brake, operated by the field current. When contactor No. 6 is open, field current is made through an auxiliary contact which short circuits the field rheostats and half

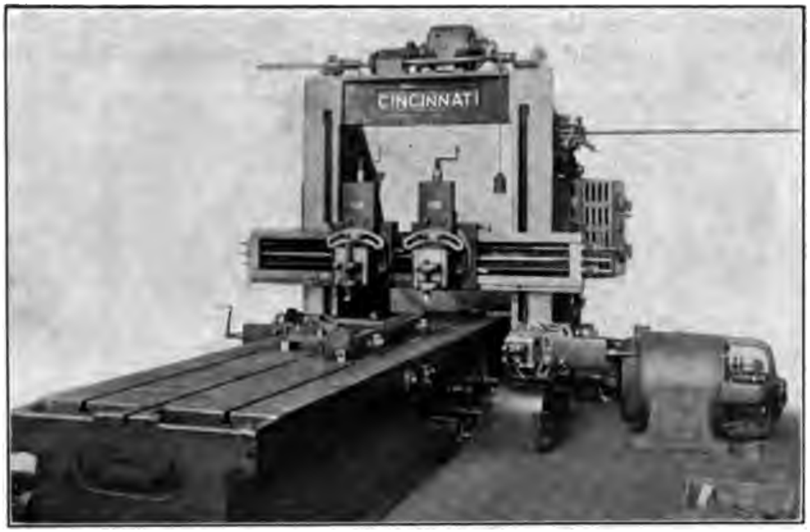


FIG. 18 48-IN. CINCINNATI PLANER DRIVEN BY 20 H.P., 250/1000 R.P.M., 230-VOLT MOTOR

of coil on No. 8 contactor. When field current approaches full field value, No. 8 contactor is so adjusted as to close.

45 At full field on the motor, which is the slowest speed, No. 8 contactor will close almost instantly, while on weak field (high speed) the contactor does not operate until such time as the field current has reached a predetermined value. This gives a definite time lag between closing of 1st point dynamic brake contactor (No. 4) and 2nd point (No. 8), which allows the motor field to be strengthened before increasing the dynamic brake load. When motor is accelerating contactor No. 6 closes, thus opening the auxiliary contact and allowing the field current to pass through the field rheostats and both sections of coil on contactor No. 8. The two sections of this

coil are wound in opposite directions and the field current passing through both sections de-energizes the contactor and it opens.

46 For emergency stopping, when the circuit breaker opens, an auxiliary contact on the circuit breaker connects the motor field to the armature, and the armature is connected across the resistance for dynamic braking.

47 To meet conditions of load where the full load cut is necessary up to the end of the stroke, a definite brake value on cutting speed is maintained independent of return speed. This is accomplished by contactor No. 7, which has two half coils, one being connected across the line and one across the armature and either coil being powerful enough to operate the contactor. When the

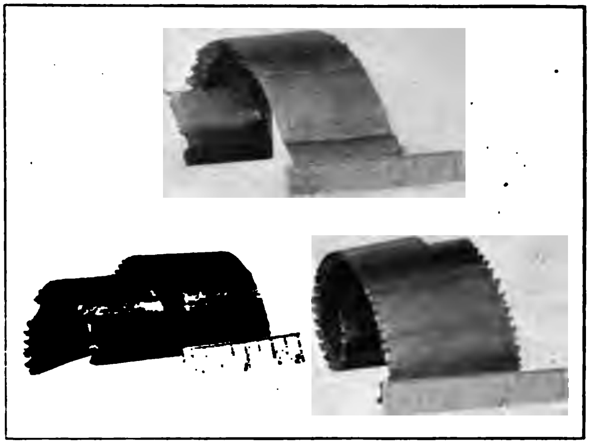
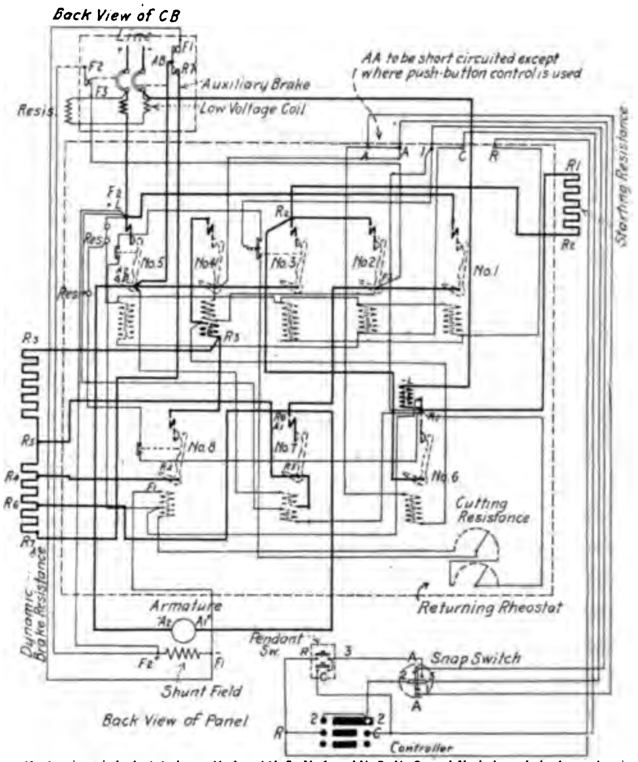


FIG. 19 CHIPS MADE ON 38-IN. HEAVY DUTY PLANER DRIVEN BY 50 H.P. GENERAL ELECTRIC CO. REVERSING PLANER EQUIPMENT

motor is running in the cutting direction, the coil across the line is in the same direction as the coil across the armature, so that the contactor operates and short circuits a part of the dynamic brake resistance, which can be adjusted to give a definite dynamic brake value on the cutting stroke. When the motor is running in the return direction, the coil across the armature is opposed to the coil across the line, and the contactor is inoperative until such time as the armature voltage has dropped to a predetermined value and the armature has decelerated to a definite speed on dynamic braking; the coil across the line then overpowers the coil across the armature and the contactor closes, giving a 3rd point dynamic braking on return stroke.

48 With the above arrangement, the maximum variation of cutting stroke between no-load and full load at high speed is about 1 in., and is practically zero at the slow speed.

49 The complete electrical equipment for planer drive consists of a reversing adjustable speed motor; a contactor panel and enclosing



Mechanical interlock between No. 4 and No. 5; No. 4 and No. 3; No. 2 and No. 1 Insulated mechanical interlock between No. 2 and No. 3.

Cutting Stroke { *Starting: No 2, No 5, No. 7 and No. 8 closed*
Running. No 2, No. 5 and No. 7 remain closed. No 6 closes and No 8 opens.
Brake Cycle: No 4 closes, followed by No. 8.

Return Stroke { *Starting: No. 1, No 3, No 7 and No 8 closed.*
Running: No 1 and No 3 remain closed No 6 closes, No 7 and No 8 open
Brake Cyles: No. 4 closes, followed by No. 8, followed by No. 7.

FIG. 20 CONNECTIONS FOR PLANER CONTROL CONTACTOR PANELS. 10 H.P. TO 25 H.P.

case with the field rheostats mounted inside of the cover, with external operating handles; the starting and dynamic brake resistance; a master controller of the drum type; a pendant switch for emergency operation, a snap switch and a special circuit breaker. A drum controller is used for controlling the cross rail motor. It is proposed to make the control equipment of the smaller horsepower sizes self contained to economize space and reduce the external wiring to be supplied by the user. An average time efficiency of 90 per cent (depending upon the speeds and cuts employed) is obtained with these equipments.

MOTORS

50 Not the least important factor for a successfully operating electrically controlled machine is the motor.

51 In a shunt motor which starts under constant field excitation, the torque is directly proportional to the armature current. This type of motor is applicable for machinery where constant speed is desired, as small printing presses, ventilating fans, small machine tools, woodworking machines, etc. For adjustable speed work it is applicable to planers, boring mills, heavy lathes, etc.

52 In a series motor, the field excitation will vary with the load, which results in a varying speed and a very powerful torque at slow speed. This type of motor should always have a certain friction load and be geared or direct connected to the load in order to avoid the possibility of the latter being thrown off and the motor accelerating to a dangerous speed. The type is applicable to centrifugal pumps, cranes and hoists.

53 In a compound motor, the combination of the shunt and series fields produces the heavy starting torque with small variation in speed. The working speed can also be increased by means of the shunt field. This type of motor is applicable to machinery requiring large overload capacity for short periods of time, as rock crushers, air compressors, shears, large printing presses, etc.

54 While the control apparatus for general use must be designed to operate with a standard motor, it is necessary that the designers of both the motor and control appliances be in thorough coöperation—especially if the customer is to obtain satisfactory results. One of the main points to bear in mind in the application of electric control to shop tools is simplicity, not only in the design of appliances, but in the control schemes as well. This is especially true in the application to special and heavy duty machines. This will mean that in many instances the motor will need to be designed for the purpose and have the desired characteristics as to stability, overload, commutation, etc. The control problem will then be very much simplified. Indeed, many of the troubles and probable failures of electric control in the past may have been due to the complicated control applied in an attempt to have the motor perform functions of which its design would not permit.

DISCUSSION

H. D. JAMES[1] (written). For starters for adjustable speed motors, the author specifies a field accelerating relay. The function of this relay is to prevent the operator from weakening the shunt field of the motor too rapidly during acceleration. The weakening of the shunt field causes the motor to take an increased current, and the relay operates in the same manner as an ammeter. When the current increases to a fixed value the relay closes the contact and increases the field strength of the motor momentarily. When the current drops the relay opens again, and this alternating closing and opening holds the field strength of the motor at the proper value during acceleration. The operation in this respect is ideal, and the relay has filled a long-felt want in many types of controllers. It has been found, however, that when the motor is operating at a high speed and the field rheostat is turned in the direction of reducing the motor speed, this increase in field strength causes the motor to act as a generator and return current to the line. This regenerative action may be very severe if the change in field strength is great. Under these circumstances the heavy current which flows causes the field relay to close the current, and still further strengthen the shunt field of the motor, causing an increased current flow. The operation of the relay in this respect is the opposite to what would be required for holding the current at its proper value.

The detrimental action of the field relay causes sufficient harm to more than counterbalance its good effects during acceleration, and better results can be obtained by omitting this field relay and using another device for strengthening the motor field during acceleration.

The author states that motors below 25 h.p. should have fuses for overload protection. The company with which I am connected is in favor of using overload relays on all sizes of motors, even down to 5 h.p., where such a relay can be adapted to the controller specified. The first cost of the relay is more than the fuse, but the continual replacing of fuses is a running expense and may often amount to more than the cost of the relay during the first year of operation. If fuses continue to open, there is a tendency to put in heavier ones, which is a further objection. Where an overload relay is used, it is so easy to reset the relay that very little annoyance is

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caused, and usually the calibration of the relay is left at a reasonable value.

The author states that a graduated dynamic braking of two or more points is necessary on planers and boring mills. Tests made with the G. E. graphic recording ammeter on planer equipments at the William Wharton Shops, Jenkintown, Pa., showed that the retardation of the planer bed was so rapid during dynamic braking that the ammeter recorded little or no difference between a single step dynamic braking and a graduated dynamic braking. These tests were made on Westinghouse, General Electric and Allis-Chalmers equipments; the G. E. equipment was of 50 h.p. size, and the others were 35 h.p. size.

In order to further analyze this condition we made oscillograph tests on 7½, 15 and 25 h.p. motors, both variable and constant speed, and with different maximum speeds and voltages. It was found that the graduated dynamic braking made practically no difference at the time of stopping, and merely added to the complication, the cost and the size of the control equipments. It is further found that a considerable variation can be made in the amount of resistance used in the dynamic braking circuit without affecting materially the time of stopping the motor. This investigation, which covers a period of over two years, has led us to abandon the graduated dynamic braking except in special cases where a considerable inertia load is stopped and the time of stopping is extended over a considerable period. The writer would be very glad to know of other tests made in this direction and what results they show. It is important to reduce controllers to a minimum number of parts, and no refinements which do not pay for themselves in actual results obtained should be introduced.

The safe temperature rise for any apparatus should be fixed by the materials entering into its design. The author states that the temperature rise on contacts and coils should not exceed 65 deg. under continuous operating conditions. This is high for some classes of apparatus and low for others. Recent improvements in insulation show that magnet windings can be operated at 125 deg. cent. actual temperature measured by resistance, and give good service over a long period of time. This comes under Class B insulation in the A.I.E.E. rules. The temperature of contacts is fixed by the fusing point of the material used; if a spring, by the temperature which draws the temper of the spring. The contacts subject to arcing, such as a line switch contactor, should be made of very refractory material, as the temperature set up in the contact on repeated operations is far in excess of any temperature which the contact may be subjected to by the passage of the current itself.

In criticism of Table 1, drum controllers we have had on the market for about ten years, having one step of starting resistance up to and including 15 h.p., and two steps of starting resistance above 15 h.p., and including 35 h.p., have proven very satisfactory. Further, we conducted a series of oscillograph tests extending over a period of six months on several hundred different motors, and we have adopted as our standard one step of starting resistance for motors up to and including 15 h.p. and two steps of resistance above 15 h.p. and including 25 h.p. Above that, we determine the number of steps by the motor used and the service conditions. Most machine tool motors start up light, as it is not the practice to start a machine tool with the tool cutting material. The resistance, however, is sufficient to start the motor under full load if the occasion should arise. In the above statement, by the resistance is meant that determined by dividing the volts by the amperes and adding sufficient external resistance to make the theoretical starting current of the proper value. As a matter of fact the self induction of the motor reduces the current at starting considerably below this value, particularly when the motor starts up at less than full load. This reduction in the number of starting notches reduces the size and cost of the controller, and if the control is rugged enough to stand the service it does not materially increase the wear on the resistance contactors. There are some types of control in which the switches used for short circuiting the starting resistance are light, and a larger number may be required in order to protect the switches.

In conclusion, it may be stated that the electrical equipment for both motor and controller is being reduced in complication and gives increased durability. For the best conditions to obtain, the controller should be selected with respect to the motor used, as some designs and types of motor require more refinement in control apparatus than others. It is the business of the electrical manufacturer to offer to the machine tool builder a complete electrical equipment having a maximum durability and a minimum complication.

The writer heartily agrees with the general statements made by the author with reference to the advisability of using electric drive for machine tools. A great deal of engineering work has been done and investigations made in connection with drives of this kind, and it is now universally recognized that the individual motor represents the best practice in this respect.

A difficulty now encountered in motor application is the mechanical one of attaching the motor and controller to the machine tool so that it makes a presentable appearance. This difficulty is being overcome by the manufacturers of electrical apparatus adapting their designs to this service and by the machine tool designers obtaining a better conception of the electric drive requirements. A great deal can be done by still closer coöperation between the designers of both classes of machinery.

The writer believes in standardizing the requirements as much as possible. There is always danger that a rigid specification made may hamper development in the art. It is better to standardize first on general requirements only, and allow the manufacturers of electrical apparatus as much leeway as possible in working out the details of their part of the equipment. The apparatus described in this paper is a good representation of controllers manufactured by one of the leading companies, and in the main is representative of the art.

H. F. Stratton (written). This subject interests me strongly, for to the best of my knowledge I made the first automatic controller generally applicable to motor driven machine tools. It is true that an automatic controller for motor driven planers was built thirteen years ago, but the reversing planer controller was then, and is now, an equipment specifically designed for use with only one tool.

Some six or seven years ago the series accelerating switch was discovered, and its obvious cheapness and simplicity at once suggested that the time had arrived for applying, in a broad way, automatic control to motor-driven machine tools. Accordingly, about five years ago, a controller was designed and built which included a train of series accelerating switches, and an operator's switch, by means of which the motor could automatically be started, reversed, or stopped by dynamic braking. This controller, in substantially its original form, has come into extensive use, and today there are thousands of them operating successfully on a large variety of machine tools.

The purpose of the automatic machine tool controller is simple and important—it is to increase production.

In Par. 2, Mr. Brooks touches rather lightly on several important matters, and possibly the following statements may merely affirm what he says. First, neither the synchronous motor nor the induction motor in its commercial form, is an adjustable speed motor. Second, if we except a few continuous running machines, fully 95 per cent of the machinery and cranes of the steel industry are driven by direct current motors. Third, the small speed increments obtainable is only one of several important advantages of the adjustable speed direct current motor over the alternating current motor when applied to machine tools. Certainly other important advantages are, first, a wide speed range which may easily be 4 to 1; second, with any given setting of the rheostat, the speed is constant regardless of the load; third, the motor can quickly be started or reversed without excessive currents; fourth, it is very simple to convert the motor temporarily into a generator and stop by dynamic braking. Certainly there can be no disputing the statement that the adjustable speed direct current motor excels the alternating current motor for the machine tool with individual drive.

In this paper, the author paints a word picture of a man manipulating a drum type or a dial type hand starter, and from this picture as a base, he passes on to the advantages of automatic control. I agree that the automatic controller possesses the advantages mentioned but am eager to put more emphasis on the main issue. What we are after is to keep the machine going the maximum amount of time and at maximum speed. Other seeming advantages are so in reality only when they yield tribute to this principle of increased capacity. It is true that automatic control protects the motor, but that is important chiefly because it saves delays; automatic control provides stopping by dynamic braking, but that too is important chiefly because unproductive time is transmuted into productive effort; it is true that the workman is relieved of much mental and physical effort, but that is important chiefly because he does more work. The problem is a matter of manufacturing economics rather than engineering technique.

As Mr. Brooks admits that the counter e.m.f. system of acceleration is unreliable, and as he mentions no compensating advantages for it, I do not see why he does not favor the universal use of current limit acceleration. In reality, there are thousands of small motors with current limit acceleration, and I see no good reason against its universal use.

Concerning the two types of current limit acceleration this point should be emphasized. Wherever possible, use series contactors in preference to shunt contactors and accelerating relays. The reasons are greater simplicity, almost total absence of fine wire coils and control circuits, absence of small control circuit contacts, less space, less money, and less apparatus.

Mr. Brooks very properly brings out the advantages of enclosing the electrical equipment.

In Par. 12, it is stated that "a double pole line contactor should be supplied." In some cases probably it should and in other cases it should not; it all depends on what you want. A single pole line contactor disconnects the motor from one side of the line but some time may refuse to open; a double pole line contactor disconnects the motor from both sides of the line, but is just as apt to stick closed as the single pole contactor; two single pole contactors disconnect the motor from both sides of the line and it is almost a certainty that both contactors will not stick at the same time. In this matter, the more you are willing to pay for, the greater safety will you get.

In several places in this section of the paper, the author states that contactors should be used to connect the motor to the supply lines, or to reverse it, or to establish dynamic braking. I disagree entirely with these statements. An enclosed type of operator's switch can be used which, when moved to different positions, establishes the motor circuits which provide starting, reversing or dynamic braking. This switch is easily mounted and operated, will last almost indefinitely, and is in extensive and satisfactory use up to 35 h.p. By its use considerable simplicity is gained and not one automatic feature is sacrificed.

Under the heading of protective features, it is stated that fuses and not overload relays should furnish overload protection for motors below 25 h.p. It ought to be apparent to anyone that this matter cannot be disposed of by such a sweeping generality. What is there peculiar about the 25-h.p. size of motor that calls for overload relays above it and fuses below it? Clearly this is a matter of service conditions. If the load is such on even a 5-h.p. motor that overload protection is needed several times a day, it is obvious that time and money will be saved by the use of circuit breaker protection instead of fuse protection.

Mr. Brooks states "in many cases it will be preferable to have the line switch and fuses a part of the distribution system, separate from the control panel," but he does not explain why. It may be preferable in some cases to, but in the majority of cases it is more convenient to have the knife switch mounted integral with the control panel. At least two good reasons for this are that there is only one equipment instead of two to install, and that the wiring between the knife switch and the control panel is eliminated so far as the user is concerned.

At the beginning of Par. 18 it is stated that a field accelerating relay should be supplied with motors above 5 h.p. and at the end of the same paragraph it is stated that on motors below 50 h.p., with a speed adjustment of 2 to 1, the field relay is not necessary except in special cases, and in Par. 20 it is stated that "with adjustable speed motors of a range greater than 2 to 1 provision should always be made so that the field resistance is not cut in until the last accelerating contactor has closed." These statements are somewhat conflicting but I take it that collectively they specify the following conditions: *First,* with adjustable speed motors having a speed range greater than 2 to 1, a contact should be provided on the last accelerating switch which short circuits the field rheostat until this switch has closed, and by inference this contact is not needed on adjustable speed motors of less than 2 to 1 speed range. *Second,* the field accelerating relay may be omitted between 5 and 50 h.p. if the speed range of the motor is not greater than 2 to 1, and by implication the relay should be used on motors larger than 50 h.p. even if the speed range is less than 2 to 1.

I prefer to look at this matter differently and to consider compound and shunt wound motors as calling for different specifications covering the field accelerating relay. With a compound motor it is generally satisfactory to omit the field accelerating relay entirely by the use of the following connections: Maintain the excitation of the series field, and full shunt field strength until the motor is directly connected to the supply lines and then at the same time short circuit the series field winding and introduce the field rheostat in series with the shunt field in one step. The damping effect of the short circuited series field delays the dying down of the field flux to such an extent that in nearly all cases the motor has ample time to accelerate to high speed without excessive armature current. In the case of the shunt motor I should recommend the use of the field accelerating relay for practically all motors of 5 h.p. or larger, and with a speed range of 1½ to 1 or more. I should recommend the omission of the contact on the last accelerating contactor to short circuit the field rheostat and instead of this connection to place the field accelerating relay directly in the armature circuit and allow it to function as soon as the armature is connected to the supply lines in series with all of the starting resistance.

It is stated that provision should be made so that the motor fields are not energized when the machine is not in use. I agree that it is preferable to disconnect the motor fields when the machine is shut down for a considerable length of time as at noon or during the night. I do not think the fields on machine tool controllers should as a rule be disconnected from the line every time the machine is stopped. The reason is that it requires an appreciable length of time for the shunt field to build up to its full strength and if this building up process occurs during the time that the motor is being started, the result is that there is a weaker average field and an increased starting interval. On work of frequent starting and stopping, more current is wasted this way than would be saved by disconnecting the shunt fields. More important, the productive time of the machine is diminished.

In Par. 21 it is stated that "with adjustable speed motors arrangement should be such as to prevent a *pump back* when stopping the motor." I take it this means that we should not permit regenerative currents due to suddenly increasing the current in the shunt field winding while the motor is still running at high speed. I agree with this most heartily, and I would point out that the most effective way to prevent this is to separate the operations of starting, reversing, or dynamic braking from the operations of speed control. This is one of the vital objections to the drum controller which has, in addition to the limitations of manual acceleration, the great disadvantage that heavy and damaging regenerative currents are caused every time the drum controller is moved quickly from a position of relatively high speed to the off position.

In Par. 22 it is stated that graduated dynamic braking should be employed on planer controllers. I do not think that dynamic braking should be used on the planer controller, and I will refer to this in greater detail later on.

Under the heading of control apparatus appliances, it is stated that a 300-ampere contactor should be good for 1½ million operations without renewal of parts, while ½ million operations would be satisfactory for a 25-ampere contactor. Now, in a general way, a 300-ampere contactor would be used on a 75-h.p. controller and a 25-ampere contactor on about a 5-h.p. controller. On ordinary machine tool work, a 5-h.p. motor is apt to be started and stopped a good deal more frequently than a 75-h.p. motor. It, therefore, seems to me that these specifications conform more to the way in which the contactor happened to be designed than they do to the requirements of the work.

It is stated that operation should be satisfactory with a voltage variation 5 per cent above and 20 per cent below normal. If 230 volts is considered normal, this gives operating limits of voltage of 184 and 242. This may be sufficient in many cases, but I have found there are numerous installations requiring voltage limits of 175 and 250.

Concerning the number of steps of starting resistance, this is too extended a topic to discuss in much detail. My personal opinion is that the table does not specify enough accelerating steps for small motors. In the case of a reversing controller for either a constant speed or an adjustable speed motor, or in the case of a dynamic braking controller for an adjustable speed motor, I do not think that less than three steps should be employed for any size of motor.

Mr. Brooks intimates that the automatic controller is not satisfactory for lathes and he states that for many uses the drum controller has been found to fill the requirements quite satisfactorily. I think there are more automatic controllers on lathes than on all other machine tools combined. Several of the fastest working machine shops in this country are using several hundred such controllers. As long ago as 1913, the R. K. LeBlond Machine Tool Co. wrote a letter containing this statement: "From recent tests made in our shop on one of our heavy duty lathes, equipped with your controlling devices, we find that we are able to produce 20 per cent more work than can be done on an ordinary motor driven lathe. These results are directly due to the automatic stopping, starting, and drift positions, and dynamic braking obtained by the use of your controlling devices."

I think the old style drum controller for lathes is unsatisfactory, and I would first refer to Par. 3 in which is described the discomfort of hand acceleration on a drum type starter, and next to Par. 21 in which it is stated that arrangements should be such as to prevent a *pump back* when stopping the motor. In a recent paper by D. M. Petty, electrical engineer of the Bethlehem Steel Company, are described "comparative tests made on duplicate engine lathes doing the same work and driven by duplicate controllers, the difference being an automatic controller in one case and a drum type controller in the other." Among other features is mentioned that "the stopping time with the automatic controller using dynamic braking in the off point, was 8 sec. as against 40 sec. with the drum type controller." Mr. Petty draws the following conclusions:

First. The automatic controller protects the motor from not only excessive currents in starting, but excessive voltages in stopping.

Second. It decreases the starting and stopping time, which would amount to a considerable item when the operation requires frequent stopping.

Note. The possible exceptions to this conclusion would be that a drum type controller might equal the automatic on applications using speed adjustments of 1 to 1½ or under.

To Mr. Petty's conclusions I would add the following points: *First,* the drum type controller has the damaging *pump back* characteristic which Mr. Brooks condemns, and *second,* with the drum type controller it is necessary to hunt for the desired speed each time the motor is stopped and started, whereas with an automatic controller this best cutting speed is maintained regardless of stopping and starting, until it is purposely changed to suit the changing requirements of new work. The only advantage which the drum controller enjoys over the automatic controller as applied to lathes, is that it requires but one spline shaft for apron control, whereas two are required for automatic control on very long bed lathes.

Under the subject of car wheel lathes it is stated that the controller panel should include in addition to accelerating contactors, two dynamic braking contactors. Why not use the same contactors for acceleration and dynamic braking, as this is easily accomplished?

Mr. Brooks says a reversing planer controller should incorporate dynamic braking. It is my belief that the planer motor should be reversed by the reverse power method instead of being first stopped by dynamic braking and then reversed. The point is to reverse the motor in the quickest safe time. The quickest reversal is accomplished by having field strength at its maximum, armature current of its highest safe value, and a minimum number of movements of magnetically operated switches. To first stop the motor by dynamic braking, means that dynamic braking switches must close and open, and in addition the reversing switches must function.

Mr. Brooks has made statements concerning engineering matters and specifies the operation which he says should characterize automatic controllers for certain different machine tools. I think these statements are, in general, the expression of the engineering policy of one of the electrical manufacturing companies.

I am sorry that Mr. Brooks felt compelled in closing his paper to intimate that it is necessary to buy the motor and the controller from the same company. He bases his argument on the statement that if the controller engineer knows the motor well enough the controller can be made to include less equipment. By inference this controller would not work as well with some other motor. In these days of standardized motors, the controller should be good enough to work with any standard motor. All competent control engineers understand the characteristics of the motors to which their controllers are applied. What is more needed is for the control engineer to get out in the field and study the work from the standpoint of the ultimate user. In general, an appreciation of operating requirements is much more needed than an understanding of motor characteristics.

C. D. Knight. The trend of the discussion seems to be regarding the relative merits of the drum controller vs. the automatic, and also the use of automatic control on lathes.

The drum controller has been in successful use for a great many years, and by many is considered an exceedingly efficient piece of apparatus. In specifying the advantage of automatic control, the author states that a considerable increase in the output of the machine is obtained, and there is no doubt but what in a great number of cases this is so. It stands to reason that a man cannot operate the drum controller throughout the whole day as rapidly as he can the automatic.

Furthermore, with automatic control the motor comes up to the same predetermined speed each time. With the drum controller, especially with adjustable speed motors, the operator is liable to obtain various speeds at different times on the same class of work, due to his not bringing the drum controller handle to the same point on each cycle of operation. With automatic control, including field relays, once the field rheostat is set for a certain speed it is absolutely certain that the motor will accelerate to that speed each time.

Another great advantage of automatic control is the question of insuring safety to the operator. By placing push buttons or small enclosed master switches on the machine, the main part of the controller can be mounted away from the machine, and whether open or enclosed, it will be out of the reach of the operator. I certainly believe that in following the lines of Safety First, we will in time have totally enclosed motor controlling apparatus.

Regarding the question of lathe control, there is no doubt but what a great many machines have been successfully equipped with automatic devices. Where the lathe is small and the speed regulating device can be placed on the headstock of the lathe within the reach of the operator, it is safe to say the problem can and has been fairly well solved. The problems I have in mind are on big lathes where the operator may be 10 or 15 feet away from the headstock of the lathe. In this case it is necessary to have the speed regulating device controlled from the apron of the lathe. There have been many proposed methods of doing this, but I have yet to see anything which looks like a successful solution of the problem.

H. K. Hathaway. The discussion of the possibilities of electrical control on milling machines, drill presses and machines of that character, which are not in the class of automatic machines in the same sense that automatic screw machines are, should open up a very fertile field. There is much to be done in the way of developing mechanism for the starting and reversing of milling machines after the cutting has been done, for the releasing of clamping devices, etc. This is a subject which our Sub-Committee on Machine Shop Practice might very well discuss further.

H. J. Eberhardt. A few years ago, at the Newark Gear Cutting Machine Company, we adapted a standard motor and starting device to one of our standard machines. The problem came up of how to stop the motor when the function of the machine had reached its limit. One way was to put a switch breaking mechanism, operated by a heavy spring, against the main switch and throw it out bodily, with the danger of an arc forming; but what we did was to utilize the overload switch, by putting a small bell crank up against the overload armature, and the pressure exerted by a fibre button, on the end of the bell crank, threw the overload switch and stopped the motor.

Ralph E. Flanders. Regarding the statement of Professor Goss, included in the paper, that in the machine tool of the future we shall not see belts, I am aware that up-to-date electrical controlling apparatus is provided with current overload, coils and switches; but there is nothing like a slipping belt for telling a shop man that there is something wrong with the machine. The very best safety device that I know of between the lineshaft or motor and the machine is a properly proportioned belt, and I wish to put in a word for the belt as a prominent feature of modern design.

Elmer H. Neff. My attention has been attracted to the quotation from Professor Goss' address. I fear the interpretation placed on the quotation is hardly correct because a machine in which there should be no pulleys, belts or gears would surely be something of a problem for a machine designer to produce. Mr. Flanders has spoken a word for the pulleys and belts, and I wish to put in a word for the gears. I am in favor of all machine tool transmission having absolutely positive connections for the main drive and for the feed, and of tying up the feed drive with the main drive so that it cannot work unless the main drive is in satisfactory operation.

At the Milwaukee meeting of this Society in 1901, one session was devoted to the subject of electrically driven machine tools. At that time an enthusiastic electrical advocate stated that in a short time practically all machine tools would be driven by electric motors. Nearly fifteen years have passed and yet today the proportion of machine tools sold fitted up with motors is relatively small. I am inclined to think that such extravagant statements as the above, and other similar statements which have been made by the advocates of electric drive, have done as much as anything else to hinder progress in that direction.

When electrically driven machine tools were first sold, the application of motors was made on machines that had not been designed with their use in view, consequently the arrangements were clumsy and cumbersome. The variations in speed had to be obtained by varying the speed of the motor or by adding to the machines themselves, between the work spindle and the motor, additional apparatus for obtaining the speed variations. This made the motor drive a very expensive one to obtain for two reasons. If the variable speed was obtained by varying the speed of the motor, the motor had to be much too large for ordinary purposes because at the slowest speed the maximum power was required. If mechanical means for changing the speed were supplied, a relatively large expense was involved.

A very fine thing has come out of this agitation for motor driven machines in that machine tools have been redesigned so that it is relatively easy to substitute a motor with sprockets and chains for the belt connection, all the speed changes being regularly incorporated in the machines themselves. These machines which are called single belt drive machines, or constant speed drive machines, furnish the best solution of the motor drive problem. A constant speed motor should be used, and by varying the speed mechanically within the machines you have the maximum rotative effect at the slow spindle speed which is the time at which you want it most. A further advantage has been that, by driving the feed works of the machine from the constant speed main drive shaft, it is possible to vary the spindle speed without changing the rate of feed, consequently any feed can be obtained with any spindle speed, and the most advantageous spindle speeds and the most advantageous feeds can be used on the work in hand. The machine thus designed, which as I have stated especially lends itself to the application of a motor, has equally great advantage when it is used as regularly intended, with a belt drive from the overhead works. With a friction pulley on the machine for the application of power, and when the machines are designed, as they are in many cases, with the possibility of reversing the direction of rotation of the spindle by the mechanism in the machine itself, it is possible to drive these machines without the use of a countershaft, by belting directly from the line shaft to the machine.

For light machine tools I am an advocate of the group drive system rather than the individual drive. By light machine tools, I refer to those requiring a 3-h.p. motor or smaller. A great multiplicity of motors such as involved by the individual motor drive involves too much expense for installation; too much expense for up-keep and attention; too much lost power. The group drive system in which a motor can be applied to a line shaft or to the machines in a particular room furnishes, I believe, the ideal electrical arrangement for light machine tools. In such cases the motors can be of sufficient size to run economically and all of the objections which I have stated with regard to the individual drive are eliminated. For large machines, or for isolated machines whether large or small, the individual motor drive furnishes a neat and desirable arrangement.

THE AUTHOR. It will be noted that the paper contemplates the permanent short circuit of the field rheostat during acceleration, and, as Mr. James states, the operation of the field accelerating relay, with proper rheostat connections, is ideal for adjustable speed motors.

Referring to the use of overload relay vs. fuses, there are, of course, personal opinions as to where the dividing line should be. This point, of course, comes up in connection with the operation of a single step starting resistance which was advocated up to 15 h.p., as the peak current during acceleration under these conditions would be considerably over 200 per cent normal current, and unless the overload relay were fitted with a time limit device, there would not be sufficient protection to the motor during normal working conditions. It will also be noted that the oscillograph tests which our company have made indicate that there is no difference in the peak current, whether the motor starts with a no-load or full-load, the main question being whether or not the motor will stand the service.

In connection with the question of dynamic braking, the opinion seems to be universal as to the necessity for this, the difference being in the method in which it is applied, and it would appear that planers and boring mills especially, where graduated dynamic braking is recommended, would come under Mr. James' classification of heavy inertia loads. The question also arises that if graduated braking is required on heavy inertia loads, why is it not also desirable in other cases? It is very easily proven mathematically that increased torque obtained by dynamic braking of necessity gives decreased stopping time, assuming that the dynamic brake resistance is properly adjusted to the operating conditions of load.

The 65 deg. temperature rise recommended on contactor coils is in accordance with A.I.E.E. rules and U. S. Government standards for this type of appliance. At the present time the use of Class B insulation for contactor coils is not general practice, although it is a probability for the near future.

In the paper and by various members in the discussion, the necessity for coöperation between the machine tool builders and the electrical manufacturers is very forcibly emphasized.

No. 1523

SAFETY CODE FOR THE USE AND CARE OF ABRASIVE WHEELS

During the year 1914 a committee of the National Machine Tool Builders' Association, of which R. G. Williams of the Norton Company was Chairman, gave careful study to the question of providing a Safety Code for the Use and Care of Abrasive Wheels and Grinding Machines. Their report was presented before the Worcester Meeting of the above Association, on April 24th, 1914.

Following the presentation of the report, representatives of the Abrasive Wheel Manufacturers conferred on the matter, and, using the report together with a tentative report of the special committee appointed by the State of Pennsylvania as a basis, brought in their own report, recommending a Safety Code which has already received wide publicity.

This Code was presented to The American Society of Mechanical Engineers, who referred it to its Sub-Committee on Machine Shop Practice for consideration. This Sub-Committee has carefully reviewed the Code, and its members have made various suggestions, involving some further modification. These suggestions have been submitted to all members of the Sub-Committee on Machine Shop Practice, and the Code as now published is endorsed by them.

It is in the main as recommended by the Abrasive Wheel Manufacturers. In nearly all the vital points, the Code recommended by the Abrasive Wheel Manufacturers is approved, and forms the basis of the Code as here printed.

LUTHER D. BURLINGAME
Chairman, Sub-Committee on Machine Shop Practice

SAFETY DEVICES

Three general types of safety devices to be used for grinding wheels, namely: protection flanges, protection hoods and protection chucks, are recommended.

ARTICLE A: PROTECTION FLANGES

A1 Protection flanges of the double or single concave type, used in conjunction with wheels having double or single convex tapered sides or side, are recommended.

Presented at the Annual Meeting, December 1915, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

A2 Flanges of the sizes shown opposite wheel diameters in column C, article A9, shall be used. As wheels wear, size of flanges, as indicated in column C, article A9, shall be maintained.

A3. New installations of protection flanges for double tapered wheels shall have a taper of not less than three-quarters (3/4) of an inch to the foot for each flange, and the center of flange shall conform with the dimensions shown in column B, article A9. Such flanges shall be of a thickness not less than is shown in column D, article A9.

TABLE 1 (ARTICLE A9) DIMENSIONS IN INCHES OF TAPERED FLANGES AND TAPERED WHEELS WHERE HOODS ARE NOT USED IN CONJUNCTION THEREWITH

A Maximum flat spot at center of flange.
B Flat spot at center of wheel.
C Minimum diameter of flange.
D Minimum thickness of flange at bore.
E Minimum diameter of recess in taper flanges.
F Minimum thickness of each flange for single taper at bore.

Diam. of Wheel in In.	A	B	C	D	E	F
6	0	1	3	3/8	2	3/8
8	0	1	5	3/8	3 1/2	3/8
10	0	2	6	1/2	4	1/2
12	4	4 1/2	6	5/8	4	5/8
14	4	4 1/2	8	5/8	5 1/2	3/4
16	4	6	10	5/8	7	7/8
18	4	6	12	3/4	8	1
20	4	6	14	3/4	9	1
22	4	6	16	3/4	10 1/2	1 1/8
24	4	6	18	3/4	12	1 1/8
26	4	6	20	3/4	13 1/2	1 1/8
28	4	6	22	7/8	14 1/2	1 1/4
30	4	6	24	7/8	16	1 1/4

NOTE: Where hoods are used in conjunction with tapered wheels and tapered flanges the specifications given in article D12 may be followed.

A4 New installations of protection flanges for single tapered wheels shall have a taper of not less than three-quarters (3/4) of an inch to the foot, and the center of flange shall conform with dimensions shown in column B, article A9. Thickness of such flanges shall be as shown in column F, article A9.

A5 Each flange, whether straight or tapered, shall be relieved or recessed at the center at least one-sixteenth (1/16) of an inch on the side next to the wheel for a distance as specified in column E, article A9.

A6 All tapered flanges over six (6) inches in diameter shall be of steel, or other material of equal strength. Tapered flanges six (6) inches and smaller in diameter may be made of cast iron.

A7 All flanges shall be accurately turned, correct to dimensions and in balance, except flanges which are purposely made out of balance. Two such flanges are known as balancing flanges and are sometimes used to counteract out-of-balance condition in an abrasive wheel.

A8 Both flanges in contact with the wheels shall be of the same diameter.

A9 Dimensions in inches of tapered flanges and tapered wheels where hoods are not used in conjunction therewith, are given in Table 1.

ARTICLE B: PROTECTION HOODS

B1 Protection hoods shall always be used where practical with wheels not provided with protection flanges. Hoods shall be designed and constructed of a material sufficiently strong to retain all pieces of a broken grinding wheel.

B2 Hoods shall conform as nearly as possible to the periphery of the wheel, and shall be so designed as to leave exposed the least portion of the wheel compatible with the work, and shall be of the adjustable type or provided with a sliding tongue or similar device, or a method of contracting the rim, for the purpose of closing the opening in the hood as the wheel is reduced in diameter, to afford maximum protection at all times.

B3 Protection hoods shall be securely fastened to the grinding machine. If advisable, hoods may also be fastened to the floor.

ARTICLE C: CUPS, CYLINDERS AND SECTIONAL RING WHEELS

C1 Cups, cylinders and sectional ring wheels shall be either protected with hoods, or enclosed in protection chucks, or surrounded with protection bands. Not more than one-quarter (¼) of the height of such grinding wheels shall protrude beyond the provided protection.

ARTICLE D: GENERAL SAFETY REQUIREMENTS

D1 Competent men shall be assigned to the mounting, care and inspection of grinding wheels and machines.

D2 Before mounting, all wheels shall be closely inspected to make sure that they have not been injured in transit, storage or otherwise. For added precaution, wheels other than of the elastic and vulcanite type should be tapped lightly with a hammer; if they do not ring with a clear tone they should not be used. Damp wheels when tapped with a hammer may not give a clear tone. Wheels must be dry and free from sawdust when applying this test.

D3 Grinding wheels shall fit freely on the spindles; they shall not be forced on, nor shall they be too loose.

D4 Wheel arbor holes shall be made .005 inches larger than the machine arbor.

TABLE 2 (ARTICLE D6) MINIMUM SIZES IN INCHES OF MACHINE SPINDLES

Diam. in In.	Thickness of Wheel in In.																		
	¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	4	4½	5
6	½	½	½	½	½	½	⅝	⅝	¾	¾	¾	¾	¾	¾	¾	¾	1	1	1
7	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	¾	¾	1	1	1	1	1	1
8	⅝	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	1	1	1	1	1	1	1	1¼	1¼	1¼
9	⅝	⅝	⅝	⅝	¾	¾	¾	¾	1	1	1	1	1	1¼	1¼	1¼	1¼	1¼	1¼
10	¾	¾	¾	¾	¾	¾	¾	¾	1	1	1	1¼	1¼	1¼	1¼	1¼	1¼	1½	1½
12	¾	¾	¾	¾	¾	1	1	1	1	1	1	1¼	1¼	1¼	1¼	1¼	1½	1½	1½
14	⅞	⅞	⅞	⅞	1	1	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1½	1½	1½	1½	1½	1½
16	...	...	...	...	1¼	1¼	1¼	1¼	1¼	1¼	1½	1½	1½	1½	1½	1¾	1¾	1¾	1¾
18	...	...	...	...	1¼	1¼	1¼	1½	1½	1½	1½	1½	1½	1½	1¾	1¾	1¾	1⅞	1⅞
20	...	...	...	...	...	1½	1½	1½	1½	1½	1½	1½	1¾	1¾	1¾	1⅞	1⅞	1⅞	1⅞
24	...	...	...	...	...	1½	1½	1½	1¾	1¾	1¾	1¾	1¾	1¾	2	2	2	2	2
26	...	...	...	...	...	...	1½	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¼	2¼	2¼
30	...	...	...	...	...	...	...	1¾	1¾	2	2	2	2	2	2¼	2¼	2½	2½	2½
36	...	...	...	...	...	...	...	...	2	2¼	2¼	2¼	2½	2½	2½	2¾	2¾	3	3

TABLE 3 (ARTICLE D12) DIMENSIONS IN INCHES OF STRAIGHT FLANGES AND STRAIGHT WHEELS USED WITH PROTECTION HOODS

A	B	C	D
Diam. of Wheel in In.	Minimum Outside Diam. of Flange	Minimum Diam. of Recess	Minimum Thickness of Flange at Bore
6	2	1	⅜
8	3	2	⅜
10	3½	2¼	⅜
12	4	2¾	½
14	4½	3	½
16	5½	3½	½
18	6	4	⅝
20	7	4½	⅝
22	7½	5	⅝
24	8	5½	⅝
26	8½	6	⅝
28	10	7	¾
30	10	7	¾

D5 The soft metal bushing shall not extend beyond the sides of the wheel at the center.

D6 Minimum sizes of machine spindles in inches for various diameters and thicknesses of grinding wheels are given in Table 2.

D7 Ends of spindles shall be threaded left and right, so that the nuts on both ends will tend to tighten as the spindles revolve. Care should be taken in setting up machines that the spindles are arranged to revolve in the proper direction, else the nuts on the ends will loosen.

D8 Wheel spindles shall be of sufficient length to permit of the nut being drawn up at least flush with the end of the spindle, thus providing a bearing for the entire length of nut.

D8a Protruding ends of the wheel arbors and their nuts shall be guarded.

D9 The surfaces of wheels in contact with straight or tapered flanges, the surfaces of the flanges in contact with the wheels and the wheel washers between the flanges and wheels shall be clean, smooth and free from foreign material.

D10 Size of straight flanges for straight wheels shall not be less than shown by column B, section D12.

D11 All straight flanges shall be relieved or recessed at the center at least one-sixteenth (1/16) of an inch on the inside surface of flange for a diameter as specified in column C, article D12.

D12 Dimensions of straight flanges and straight wheels used with protection hoods are given in Table 3.

D13 Wheels shall never be run without flanges.

D14 Both flanges in contact with the wheels shall be of the same diameter whether straight or tapered.

D15 Wheel washers of compressible material, such as blotting paper, rubber or leather, not thicker than approximately 0.025 inches, shall be fitted between the wheel and its flanges. It is recommended that the wheel washers be slightly larger than the diameter of the flanges used.

D16 When tightening clamping nuts, care shall be taken to tighten same only enough to hold the wheel firmly, otherwise the clamping strain is apt to crack the wheel.

D17 Flanges, whether straight or tapered, shall be frequently inspected to guard against the use of flanges which have become bent or sprung out of true, or out of balance. If a tapered wheel has broken, the tapered flanges shall be carefully inspected for truth before using with a new wheel. Clamping nuts shall also be inspected.

D18 The work rest must be kept adjusted close to the wheel to prevent the work from being caught. Work rests must be rigid and always securely clamped after each adjustment.

D19 (1) A speed of 5000 peripheral feet per minute is recommended as the standard operating speed for vitrified and silicate straight wheels, tapered wheels and shapes other than those known as cup and cylinder wheels, which are used on bench, floor, swing frame and other machines for rough grinding. Speeds exceeding 5000 feet may be used upon recommendation of the wheel manufacturer but in no case shall a speed of 6500 peripheral feet per minute be exceeded.

TABLE 4 R.P.M. FOR VARIOUS SIZES OF GRINDING WHEELS TO GIVE PERIPHERAL SPEED IN FT. PER MIN.

Diam. of Wheel in In.	4,000	4,500	5,000	5,500	6,000	6,500
1	15,279	17,200	19,099	21,000	22,918	24,850
2	7,639	8,590	9,549	10,500	11,459	12,420
3	5,093	5,725	6,366	7,000	7,639	8,270
4	3,820	4,295	4,775	5,250	5,730	6,205
5	3,056	3,440	3,820	4,200	4,584	4,970
6	2,546	2,865	3,183	3,500	3,820	4,140
7	2,183	2,455	2,728	3,000	3,274	3,550
8	1,910	2,150	2,387	2,635	2,865	3,100
10	1,528	1,720	1,910	2,100	2,292	2,485
12	1,273	1,453	1,592	1,750	1,910	2,070
14	1,091	1,228	1,364	1,500	1,637	1,773
16	955	1,075	1,194	1,314	1,432	1,552
18	849	957	1,061	1,167	1,273	1,380
20	764	860	955	1,050	1,146	1,241
22	694	782	868	952	1,042	1,128
24	637	716	796	876	955	1,035
26	586	661	733	809	879	955
28	546	614	683	749	819	887
30	509	573	637	700	764	827
32	477	537	596	657	716	776
34	449	506	561	618	674	730
36	424	477	531	584 ~~55[illegible]~~	637	689
38	402	453	503	553	603	653
40	382	430	478	525	573	621
42	364	409	455	500	546	591
44	347	391	434	477	521	564
46	332	374	415	456	498	539
48	318	358	397	438	477	517
50	306	344	383	420	459	497
52	294	331	369	404	441	487
54	283	318	354	389	425	459
56	273	307	341	366	410	443
58	264	296	330	354	396	428
60	255	277	319	350	383	414

(2) A speed of 4500 peripheral feet per minute is recommended as standard operating speed for vitrified and silicate wheels of the cup and cylinder shape, used on bench, floor, swing frame and other machines for rough grinding. Speeds exceeding 4500 peripheral feet per minute may be used upon recommendation of the wheel manufacturer

facturer but in no case shall 5500 peripheral feet per minute be exceeded.

D20 For elastic, vulcanite and wheels of other organic bonds, the recommendation of individual wheel manufacturers shall be followed.

D21 For precision grinding an operating speed of 6500 peripheral feet per minute may be recommended. Speeds higher than 6500 peripheral feet per minute can be used only upon recommendation of the wheel manufacturer.

D22 Table 4 gives revolutions per minute for various sizes of wheels for the peripheral velocities feet per minute at the head of each column.

D23 Machine spindle speeds shall be tested and determined correct for size of wheel to be operated before wheel is mounted, and shall never be changed as a wheel is reduced in diameter, except by men assigned for such duties.

D24 If a wheel spindle is driven by a variable speed motor, speed control of the motor shall be enclosed in a locked case, or some device shall be used which prevents motor from being run at too high speeds.

D25 Grinding machines shall be sufficiently heavy and rigid to prevent vibration, and they should be securely mounted on substantial foundations.

D26 No user of wheels shall use on any given machine a wheel of larger diameter or greater thickness than specified by the machine builder.

D27 Wheels which wear out of round shall be trued by a man assigned to that duty. If wheels, not provided with balancing flanges, become out of balance through wear and cannot be balanced by truing or dressing, they should be removed from the machine.

D28 A wheel used in wet grinding shall not be allowed to stand partly immersed in the water. Water-soaked portion may throw the wheel dangerously out of balance.

D29 Wheel dressers should be equipped with rigid sheet metal or other guards over the tops of the cutters to protect operator from flying pieces of broken cutters.

D30 Goggles shall be provided for use of grinding wheel operators where there is danger of eye injury. They should be readily accessible, or better, should be the individual property of the operator.

D31 The space about the machine shall be kept dry, clean and as free as possible from castings or other obstructions.

D32 Grinding rooms shall not only be well ventilated and well lighted, but kept warm and dry. Machines used continuously for dry grinding shall be attached to a dust-exhausting system. Besides protection to the workmen, the dust-exhausting system prevents wear and tear on machinery and belts.

D33 Care shall be exercised in the storage of wheels. They shall be stored in dry places and should be well supported on edge in racks. Work shall not be forced against a cold wheel, but the work applied gradually, giving the wheel an opportunity to warm and thereby eliminate possible breakage. This applies to starting work in the morning in grinding rooms which are not heated in winter, and new wheels which have been stored in a cold place.

ARTICLE E: PRECAUTIONARY SUGGESTIONS

E1 Cone pulleys determining the speed of a wheel should never be used unless belt locking devices are provided.

E2 The maximum size of wheel which should be used with given operating speeds should be indicated on each machine.

E3 Grinding machines should be provided with a stop or some method of fixing the maximum size of wheel which may be used, at the speed at which the wheel spindle is running.

E4 Boxes should be of proper length to provide an ample bearing surface, and prevent heating or rapid wear. It is important that the bearings be kept well lubricated and properly adjusted. Ring oiling devices are recommended, amply protected from dust and grit, and box caps should be adjustable for take-up.

E5 For protection against flying chips, etc., plate glass in metal frames can be placed just above the grinding spaces of the wheels.

E6 Where it is impracticable or undesirable to use a glass shield, a leather flap may be attached to the hood and adjusted so as to interrupt sparks and dust.

DISCUSSION

L. D. Burlingame. An important consideration in the automatic control of machine tools is the control of grinding wheel speeds so that wheels cannot be run beyond the safe limit. As the wheel, when it wears down, requires a greater number of revolutions to maintain the same surface speed, the problem is to provide such safety means as to prevent this higher speed being used when a full-sized wheel is in position.

Ralph E. Flanders. I remember some machines in which the hood was so arranged with reference to the belt that it could not be shifted on to the smaller step, except as the wheel grew smaller and the hood was adjusted for the smaller diameter, thus preventing over-speeding of the grinding wheel.

Adolph L. De Leeuw. While it seems to me that the language of the Code is in some particulars rather indefinite, the Code forms a basis and a starting point for something later which may be more definite.

The Cincinnati Milling Machine Company provided for driving their grinding machines by electric motors, the controller for the motor being in a locked box, of which the foreman of the grinding department held the key. The grinding wheels, too, were kept in a locked compartment to which he also had the key. If the machine was set for the proper speed for, say, a small wheel, and it was desired to change to a large wheel, it was thus "up to" only one man with specific instructions, who had the necessary knowledge to open up the box of the rheostat, change the speed, and supply the workman with the required wheel.

As grinding machines are furnished nowadays there is, practically speaking, no means of adjusting the speed of the grinding wheel; the two or three steps vary so widely that the shifting of the belt one single step is entirely too great to obtain the best efficiency of the wheel at all times.

Extended experiments, for instance, on the grinding of cast iron have shown that there is a very marked difference in the amount of metal which can be removed with a certain given amount of weight of wheel, depending on the speed of the wheel. If, for instance, in grinding a certain cast iron bushing it was possible to remove 2.7 cu. in. of metal for 0.002 wear of a wheel running at 5700 ft. per min., it was possible to remove 0.5 and 0.7 in. of cast iron on the same kind of bushing with that same grade of wheel when running at 6000 ft.

There seems to be a critical speed at which the wheel has a maximum metal removing capacity, and that critical speed is so close to some other speed, which is very much less efficient, that the present arrangement of grinding machines does not meet at all the demands of efficiency.

CHARLES FAIR. Mr. Flanders spoke of a grinder on which there was a hood arrangement to regulate the speed of the wheel. That would be a very easy thing to do if an adjustable speed motor were driving the wheel, provided there was an attachment corresponding to a hood. Of course, that would only keep the peripheral speed of the wheel per grade of wheel; it would have to be changed as the grades of the wheels themselves were changed, calling for a difference in peripheral speed.

No. 1524

GAS PRODUCERS WITH BY-PRODUCT RECOVERY

By Arthur H. Lymn,[1] London, England
Non-Member

The art of generating producer gas from coal is a very old one, but the development of the simultaneous recovery of the valuable by-products is comparatively recent, having been confined to the last twenty-five years or so. The object of this paper is to present a historical resumé of this development in Europe.

2 The early attempts in Europe to recover the by-products of the producer gas process are generally recognized to have been made in Great Britain. In that country the knowledge that the treatment of fuel by a mixture of steam and air (the former in excess) would convert a large percentage of the nitrogen contained in the coal into ammonia was first applied in practice on a large scale. The details of a plant to operate on this principle had already been worked out by Messrs. Young and Beilby in England and Grouven in Germany among other investigators.

3 The gas producer designed by Young and Beilby differed in operation from the ordinary by-product gas producer in that it was heated from the outside. The coal was distilled in the upper part of the producer or retort and the tar vapors passed down through red hot coke and were (it was claimed) decomposed into permanent gas and ammonia. The coke in the lower half of the producer was burned in a mixture of steam and air and the resulting gases, together with the gaseous products of the coal distillation, passed out of the producer by way of exits at the middle. The arrangement is shown in Fig. 1.

4 It is particularly interesting to note that, as far back as 1883, Young and Beilby claimed to recover in the form of ammonia from

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Presented at the Annual Meeting, December 1915, of The American Society of Mechanical Engineers. Contributed by the New York Section.

60 to 70 per cent of the total nitrogen in the fuel and that the percentage composition of their gas was:

Carbonic acid	16.6 per cent
Carbonic oxide	8.1 per cent
Methane	2.3 per cent
Hydrogen	28.6 per cent
Nitrogen	44.4 per cent

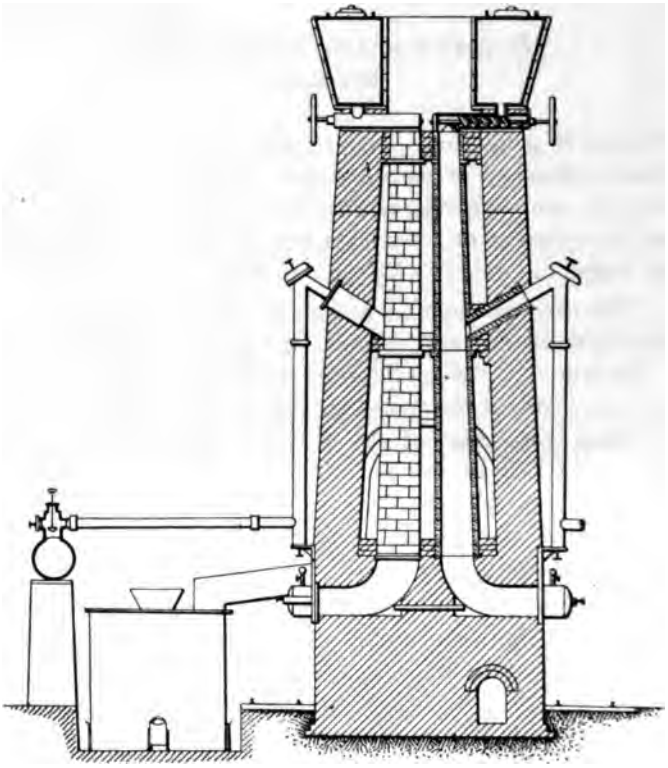

FIG. 1a YOUNG AND BEILBY'S BY-PRODUCT GAS PRODUCER

Although their retort was heated from the outside instead of the air and steam blast being superheated, it will be seen that the results claimed by them as to ammonia were not far short of what we realize today. The gas composition, too, was practically the same as that of the gas which has since become so widely known as Mond gas and which has the following percentage composition:

Carbonic acid	14 to 16 per cent
Carbonic oxide	10 to 12 per cent

Methane	2 to 3 per cent
Hydrogen	25 to 29 per cent
Nitrogen	Difference

The above makes it obvious that Dr. Mond was not the first investigator to produce gas of this composition.

5 It is now rather more than twenty-five years since the late Dr. Ludwig Mond first put into commercial practice the process described in his British Patents No. 3821 of 1883 and 8973 of 1885, of gasifying fuel by means of steam and air and simultaneously recovering the ammonia. His first plant was installed at the works of Brunner, Mond and Co., Northwich, England, and its capacity was developed to some 200 tons a day. Unfortunately this plant was not in a good position to be photographed, but a diagram of it is given in Fig. 2.

6 The Mond producer was rectangular in section and was formed with a kind of double chamber. Its operation was similar to that of Young and Beilby, in that the coal was distilled in a downward direction in the upper part and the coke residue was gasified in the lower part, all the gases mixing and leaving the producer together.

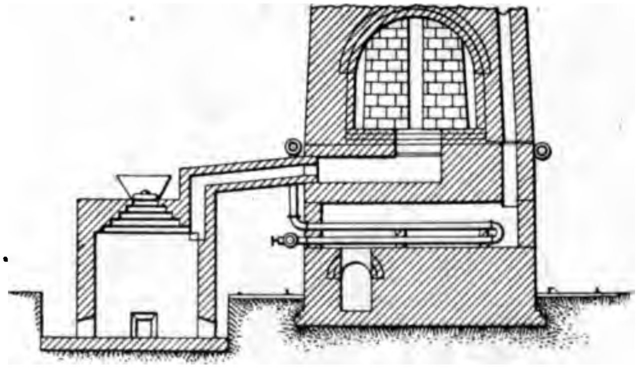

FIG. 1b ADDITIONAL SECTION OF FIG. 1a

The gas was passed into a long horizontal rectangular washer and a fine spray of water was thrown into it by a series of revolving dashers. By this means a large proportion of the dust was removed, which was afterwards taken out of the water lute manually with long scoops, an irksome operation. From the washer, the gas was conducted into a high lead-lined acid tower (filled with earthenware ring tiles) where in passing upwards it came into contact with sulphate of ammonia solution trickling down. This solution contained a slight

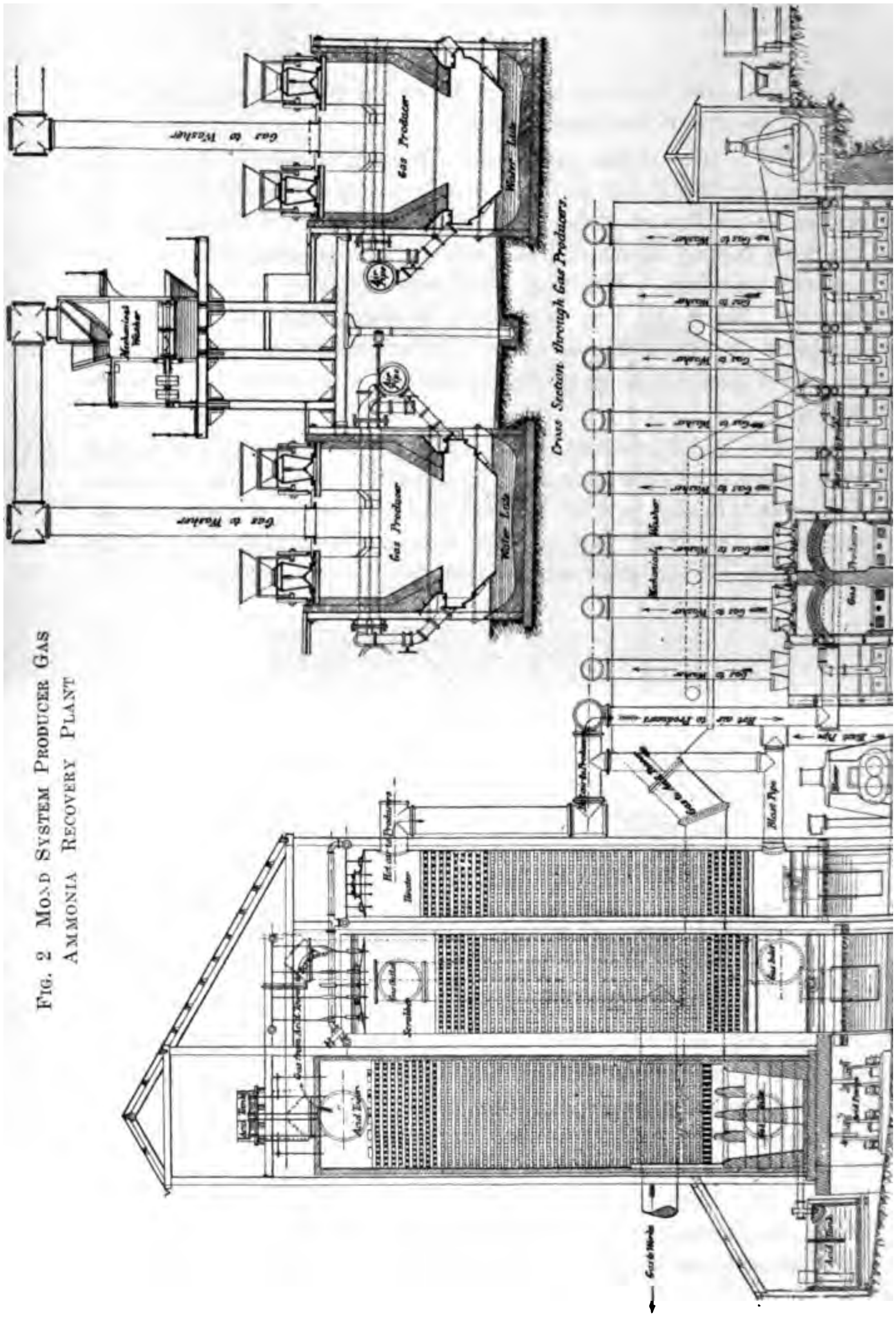

Fig. 2 Mond System Producer Gas Ammonia Recovery Plant

excess of sulphuric acid and deprived the gas, by absorption, of nearly all the ammonia contained in it. From this tower, the gas was passed into a similar tower, called the scrubber, where it was brought into contact with cool water for scrubbing and cooling it and it was then delivered for use.

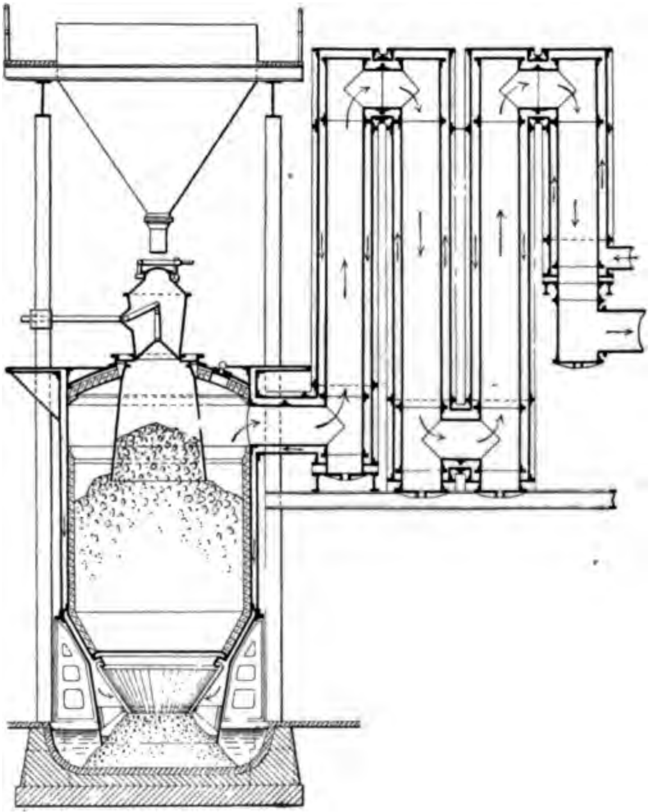

FIG. 3 MOND PRODUCER WITH SUPERHEATER

7 The water, having taken up the heat of the gas, was collected in tanks from which it was introduced into the heater, also of tower construction. It was here brought into contact with the cold air to be used in the producers, saturating this air with water vapor at from 70 to 80 deg. cent. and becoming again cooled. Then it was returned to the scrubber, where it again took up the heat of the gas, and so on in a continuous cycle.

8 After this plant had been in operation for some time, it was found that, owing to the large proportion of steam in the air blast, the heat value of the gas was below the desired standard and also the yield of ammonia was less than that which Dr. Mond had set out to obtain. It was therefore decided to change the design of the gas generating part of the plant so that the air and steam blast would enter the producer with a considerable degree of superheat, thus enabling a still greater excess of steam to be used. This modified design was disclosed in Mond's British Patent No. 12,440 of 1893 and is shown in Fig. 3. The producer was made circular in section instead of rectangular and its whole shell was surrounded by a jacket through which the air was passed on its way to the grate, reducing the losses from radiation and at the same time further superheating the steam and air blast. Directly contiguous to the producer was arranged a superheater, consisting of a series of parallel tubes with alternate ends connected, surrounded by a series of larger tubes forming an annular space. The gas from the producer passed through the inner tubes and superheated the steam and air blast which was passed through the annular space in a counter-current direction on its way to the producer. With this provision, the gas was found to possess a much higher heat value and a considerably increased yield of ammonia was obtained without extra fuel.

9 It is interesting to note that Dr. Mond, with that great courage in engineering undertakings for which he was famed in the days of his greatest activities, built a very large unit plant at the outset, and smaller plants in later years. This was contrary to the usual order of things, and discloses the reason for the wide-spread idea that by-product producer plants were only profitable when built in very large and costly units.

10 Among the plants built by Dr. Mond or his successors in England—The Power-Gas Corporation—the central station at Dudley Port, Staffordshire, for the distribution of Mond gas over an area of about 120 square miles, through about 30 to 40 miles of pipes, is of particular interest. The author had charge of this plant for some time. The gas from this station is supplied to iron and steel works, machine shops, foundries, galvanizing works, pumping stations, enameling works and municipal electric stations. The installation had originally a capacity of 16,000 h.p., which has just recently been largely increased. A general view of this plant which is familiar to many engineers in this country is shown in Fig. 4.

11 The Dudley Port plant is quite unique in that it is the only central station in the world designed and built for the distribution of producer gas. The gas is sold to consumers in competition with coal, ordinary lighting gas and electricity. The advantages to be derived from taking supplies of the gas were not apparent to the public for several years, but once they had been fully demonstrated the number of consumers rapidly increased.

12 Up to about the year 1897, Dr. Mond was constantly endeavoring to improve his process, but after that time he appeared to be satisfied with his design and in fact in advancing years he be-

FIG. 4 MOND GAS CENTRAL STATION AT DUDLEY PORT, ENGLAND

came adverse to any material modifications. The result was that the design of the Mond plants did not advance with the times and the writer is of the opinion that in those cases in America in which Mond system plants have not been so successful as was expected, it has been chiefly attributable to this factor and to the failure to realize that a certain state of reliability and efficiency in other countries, with quite different fuels and under totally unlike conditions, is not necessarily a criterion for exactly the same results in this country.

13 After Dr. Mond had demonstrated the success of his process, it was not long before another worker, E. J. Duff, claimed attention. As a whole, Duff's plant embodied very little to distinguish it from the Mond plant, the same process being of course carried out in the two plants. Duff's producer was of rectangular section inside, with

rounded corners, and his superheater, as shown in Fig. 5, consisted of a series of parallel gas tubes with alternate ends connected, surrounded by one chamber. Baffle plates in this chamber compelled the steam and air blast to take a zig-zag path on its way to the producer. Ex-

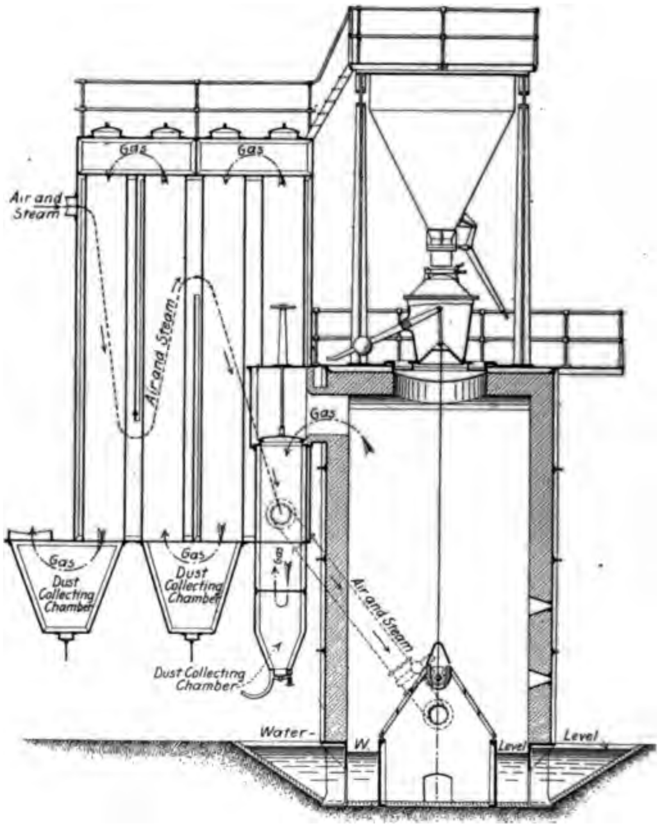

FIG. 5 DUFF PRODUCER WITH SUPERHEATER

cept for this slight difference in the construction of the superheater, which, however, occasioned no difference in the action of the producer, there was practically nothing to distinguish the Duff plant from the Mond plant.

14 Several large plants were constructed according to Mr. Duff's designs. One of these is shown in Fig. 6. This was erected at Fleetwood, England. The photograph does not show the producers very

well but conveys a good idea of the great height and size of the towers for ammonia absorption, gas cooling and air saturation. These large towers are characteristic of the plants of this construction. It is well known in Great Britain that Mond and Duff were at one time in conflict in the matter of their patents, but that their interests were afterwards amalgamated into one company, The Power-Gas Corporation, formed in 1901. The patents in question have since expired in

FIG. 6 DUFF PLANT AT FLEETWOOD, ENGLAND

England, but it may be that some of Duff's American patents are still in force.

15 Messrs. Crossley Bros., of Manchester, were the next to claim material improvements, which, however, do not appear to have been realized in practice. In fact, it is the author's belief that some time ago this firm ceased altogether to build plants for the gasification of coal and the simultaneous recovery of the by-products. It is perhaps of interest nevertheless to include here a diagram of the first form of plant Messrs. Crossley Bros. adopted (Fig. 7).

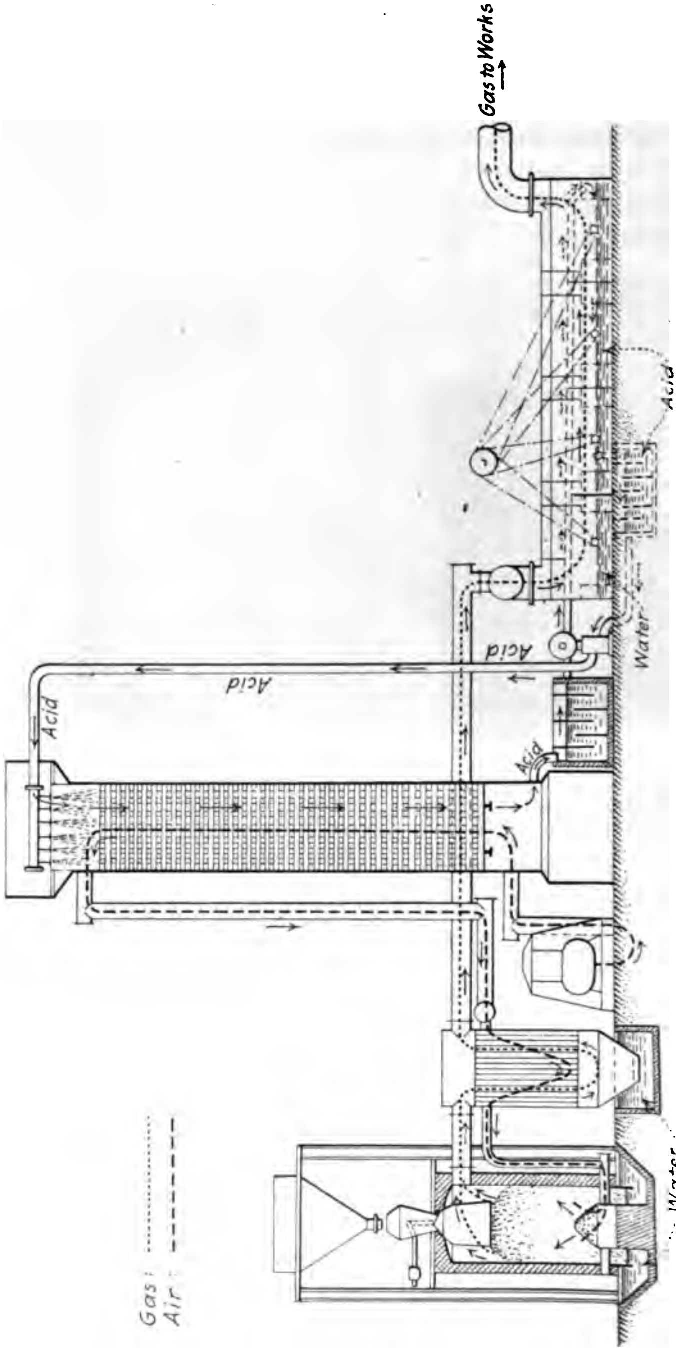

Fig. 7 Crossley and Rigby Patent Ammonia Recovery Plant

16 Briefly the claims made for this plant were that the washing and cooling of the gases, as well as the condensing of the water vapor and the absorption of the ammonia, took place in one and the same apparatus, the ammonium sulphate liquor being utilized for the purpose of saturating the air with water vapor and the liquid being thereby cooled at the same time. As a matter of fact, in the first series of operations a washer made up of two compartments was used; the gas would leave this apparatus in a more or less uncooled state and also practically saturated with water vapor at a comparatively high temperature. Fig. 7 does not show therefore all the necessary apparatus.

17 The second claim made, that is, that the air was saturated by

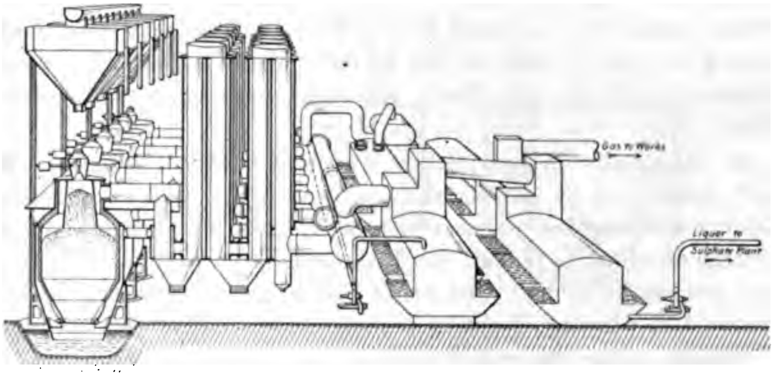

FIG. 8 MOND TYPE AMMONIA RECOVERY PLANT WITH DOUBLE CHAMBER WASHERS

means of the sulphate liquor, represented a very dangerous practice. A fine sulphate of ammonia liquor spray, with its excess of highly-corrosive acid, would naturally be carried forward to the superheaters, etc., and with very obvious results. Messrs. Crossley Bros., as a matter of fact, took out later a patent for the utilization of the above system in combination with a somewhat costly bed of lime, which the saturated air had to pass through, clearly to absorb its contained acid spray.

18 The next step in the development of this process was that taken by the author on the basis of his British Patent No. 8014 of 1908. This was an attempt to dispense entirely with the costly and irksome towers of the Mond and other plants, which became blocked up from time to time causing serious trouble and delay. The attempt

was made by replacing the mechanical washer and towers by washers of special construction, which obviously could not become blocked up. Four double washers were proposed, one for washing the gas, the second for absorbing the ammonia, the third for cooling the gas and the fourth for saturating the air. This design was later modified in favor of a double-luted washer and was changed a third time by The Power-Gas Corporation for a combination of double and single chamber washers upon the same principle. An idea of the general appearance of a plant with these washers may be obtained from Fig. 8.

19 Various modifications of ammonia recovery plants are also disclosed in patents taken out by A. B. Duff, of Pittsburgh. Notable among these is a producer with a circular grate and also a circular section superheater with four enclosed gas tubes. The former is illustrated in British Patent No. 16,164 of 1903 and the latter in British Patent No. 16,243 of 1903. These modifications have been adopted in Great Britain in two or three plants and are, I believe, working successfully with Scotch and other more or less non-caking coals.

20 In another design (British Patent No. 4372 of 1910) A. B. Duff claims that by passing the gas around the evaporator before entering the washers, its heat can be utilized for evaporating the sulphate of ammonia liquor. On first sight, this idea appears to be a good one, but it seems to the writer that great difficulty may be experienced in carrying it out in practice on account of the dust and tar present in the gas at this stage of the operations. Moreover, it must be borne in mind that in this design the gas is washed before it is allowed to enter the ammonia absorption tower, and that the washing water is used for saturating the air going into the producers. It therefore appears that the air will carry to the producers a not inconsiderable proportion of the ammonia which will thus be lost. The author is not aware of a plant on these lines having been built and put into operation, and is but meagrely informed concerning the earlier Duff plants built in the United States.

21 It may be taken that all the designs above referred to were, broadly considered, based upon what is generally known throughout the world as the Mond process. Two different propositions for the gasification of coal and the simultaneous recovery of the by-products will now be referred to.

22 The first of these is that by F. J. Rowan, of Glasgow, who proposed to combine the gas producer process with the usual process

utilized in lighting gas works, viz., to replace the absorption of the ammonia by acid (as is done in the Mond process) by a condensing plant. He did not propose actually to produce the ammonia in any different way, and it is obvious that since, for every ton of coal gasified, approximately 150,000 cu. ft. of gas at normal temperature and pressure, together with about 1.5 tons of excess water vapor, are produced, an apparatus for condensing the ammonia cannot be otherwise than

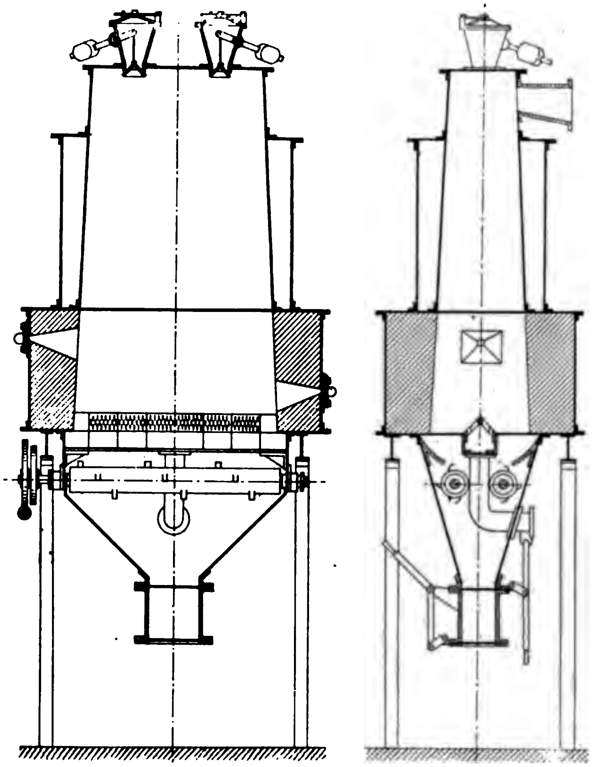

FIG. 9 MOORE'S THREE-PART PRODUCER

very excessive in size. Moreover, the gases enter the condensing plant at a temperature of 400 deg. cent. or more, at which temperature their volume is multiplied several times. The condensed ammonia liquor is extremely dilute and must be distilled with lime in the usual manner adopted in lighting gas works, a not very pleasant operation. These facts make the difficulty of carrying out this proposition obvious, and the writer is not aware that any plant was built on these lines.

The author himself has tried surface cooling for by-product producer gas but has found firstly, that meteörological conditions have too much influence on the result and secondly, that the apparatus required would be much more expensive owing to the large volume of gas to be dealt with and the lower rate of heat transference possible.

23 The second proposition is that by Quintin Moore who designed a producer divided into three parts, a lower brick-lined part, a middle water-jacketed part and an upper air-cooled part. The arrangement is shown in Fig. 9.

24 By this cooling of the upper part of the producer, Moore

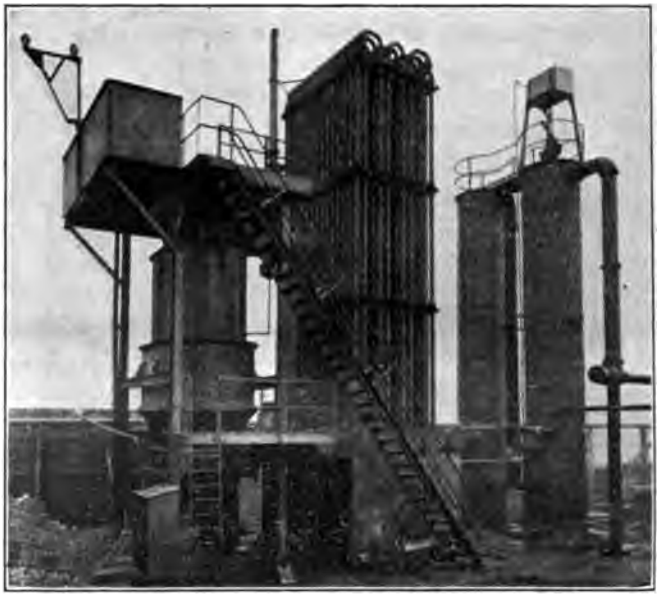

Fig. 10 Moore's Patent System By-Product Recovery Plant

claimed to obtain a good yield of ammonia with about half the amount of steam in the air blast. It does not seem likely, however, that such cooling can penetrate far into the fuel bed, and it should not be overlooked that other workers, in particular the late Dr. Mond, previously considered the possibility of recovering ammonia by means of merely cooling the producer, but came to the conclusion (based on sound scientific knowledge) that cooling was not the only desideratum in ammonia recovery.

25 One can easily overrate, too, the importance of saving steam.

In the Mond process, practically all the steam is generated from waste heat and, moreover, most of it is continuously recovered. Any further saving can only be secured at the expense of some of the sulphate of ammonia yield. At all events, Mr. Moore states in his published matter that he obtains more ammonia when he introduces more steam, hence his usual amount of steam does not produce the maximum ammonia recovery.

26 Fig. 10 illustrates a plant built on the Moore system. The small air-cooled tubes are so constructed and in such a position as to bring about great likelihood of frequent stoppage of the plant on account of tar and dust deposits. Whatever may be the results with this plant with non-caking coals, the plant does not seem to be applicable to caking fuels with any reasonable degree of efficiency. The only publication regarding this system which has come before the writer shows the results of tests of but five hours duration, which tests have obviously but little value for practical purposes.

27 Some five years ago the writer set out to design a new type of plant which should retain the advantages of previous types without their disadvantages. He was led to do this by the realization that, in spite of the extreme cheapness of producer gas as made by plants operating under the Mond system and although a considerable number of plants had been built and operated in a somewhat restricted number of countries, the adoption of the producer gas process had not become general throughout the industrial world. Careful investigations of the situation led to the conclusion that although the process as such was, and is, really good, the means adopted for carrying it out left much to be desired, especially from the point of view of capital outlay, labor requirement, repair costs and simplicity of operation. These drawbacks were inherent primarily to the ammonia absorption, gas washing, gas cooling and air saturating elements of the plant. All these operations were heretofore carried out in high and cumbersome towers packed with earthenware ring tiles, or in equally cumbersome horizontal luted dasher washers.

28 A system of vertical washers, in which an intensive washing of the gases was brought about chiefly by means of the momentum of the gases, was first designed and put into practice. Vertical mechanical washers had not been used previously in gas producer plants and the writer first designed one on the lines already known in other branches of the gas industry, in which the washing liquid was sprayed by means of a series of co-axial revolving discs upon collecting cones,

each of which delivered the liquid directly upon the next revolving disc below, and so on. It was found, however, in setting this washer to work, that with such an arrangement, the momentum of the gas was performing much more work than was the mechanical movement of the discs, a rather surprising fact. Accordingly, the mechanical feature of the washer was eliminated, the collecting cones were cut away to give the gas more play and the washer now had the appearance as shown in Fig. 11. It will be seen that if plumb lines are taken down the inside edges of the collecting cones and down the outside

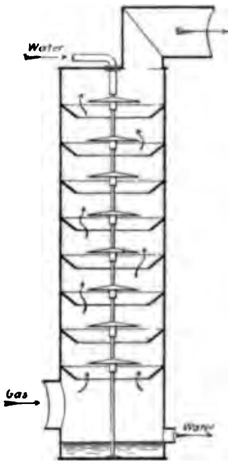

FIG. 11 LYMN WASHER

edges of the discs, a considerable space exists between them, which is such in practice that, if no gas is passing, the water entering at the top falls straight down to the bottom.

29 With this modified washer, a plant originally designed to deal with the gas from 45 to 50 tons of coal per day was able to deal with that from 90 to 100 tons per day, so that the capital outlay of the gas washing part of the plant was straightway reduced one-half. Such washers have now been in operation for approximately two years with entire success, and it is now possible to design washer units of this particular type to deal with quantities of gas from 10,000 up to about 1,300,000 cu. ft. per hr.

30 Fig. 12 illustrates the general appearance of a plant with these washers. The dimensioning of the washers is not a very simple matter, being of necessity purely empirical, and the author has arrived at the dimensions entirely by stepwise trial.

31 If Fig. 12 is compared with the previously shown diagrams of other plants, it will be seen that the whole apparatus looks much less cumbersome and obviously simpler in operation, involves less auxiliary machinery and consequently less first cost. It will be further noted that exactly the same kind of apparatus is employed for each of the operations of absorbing the ammonia, cleaning the gas and recovering the steam. For the final stage of removing the last traces of tar, etc., however, centrifugal cleaners combined with dry scrubbers,

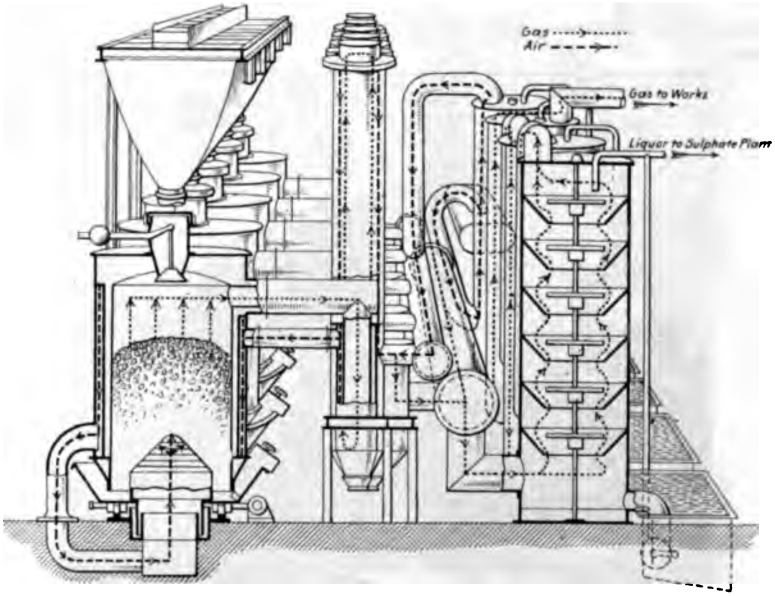

Fig. 12 Lymn Type Producer Gas Ammonia Recovery Plant

or such methods as have been proposed by H. F. Smith, have to be utilized.

32 Steel, instead of lead, is used in these plants for the parts of ammonia absorbing apparatus. A Lymn plant constructed with no lead whatever has been in operation in Germany for approximately four years and no corrosion has yet been discovered. This plant is shown in Fig. 13. Its capacity is 8000 h.p.

33 Improvements in the removal of dust have been accomplished by the adoption of a cyclonic dust separator of somewhat special design. With this separator, the bulk of the dust is removed in a dry state and not by means of water. The removal of all the wet and sloppy

dust from the earlier horizontal rectangular washers was a very troublesome proceeding, involving considerable costly manual labor, as will easily be realized.

34 In the gas producer itself of an ammonia recovery plant, mechanical action which has been so widely applied to ordinary gas producers, both in the United States and in Germany but not so much in England, involves agitation in the fuel and ash zones of the producer and mechanical ash removal. Many attempts in this direction of mechanical action have been made and some of them have met with more or less success. The writer believes the first of such proposals

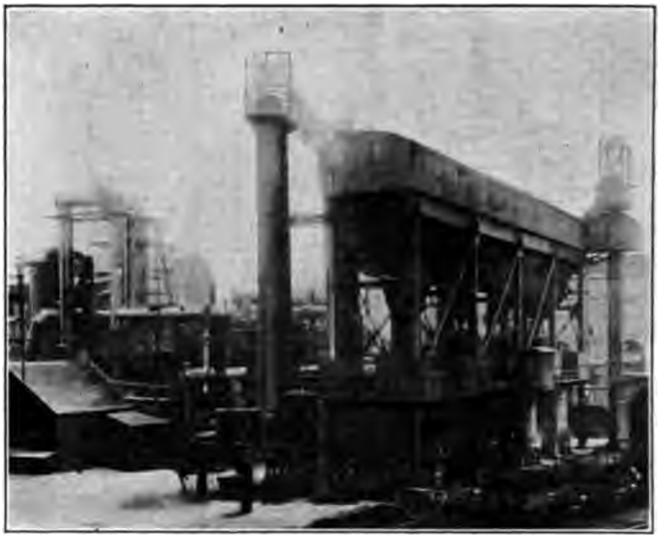

Fig. 13 8000 H.P. Lymn System Producer Gas and By-Product Recovery Plant

was made by E. J. Duff, who designed an octagonal-section revolvable producer with a stationary grate, ash trough and top, as shown in Fig. 14. This design was disclosed in British Patent No. 15,646 of 1901, but the specification does not state that it was intended for ammonia recovery purposes, although, as far as the author's recollection goes, the inventor considered it primarily in this connection. The writer is unaware of any practical trial having been made of this producer.

35 The next of these proposals was a combination of the well-known Talbot stirrer with the Mond producer. This was made by

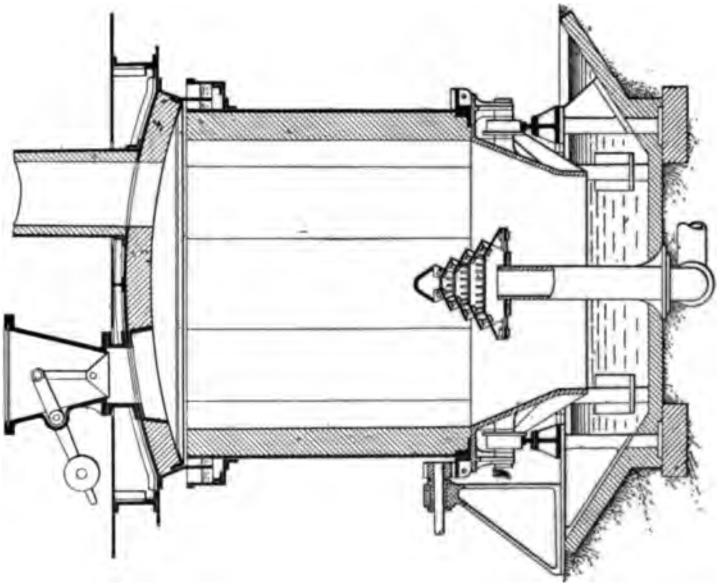

FIG. 14 DUFF PATENT REVOLVABLE PRODUCER

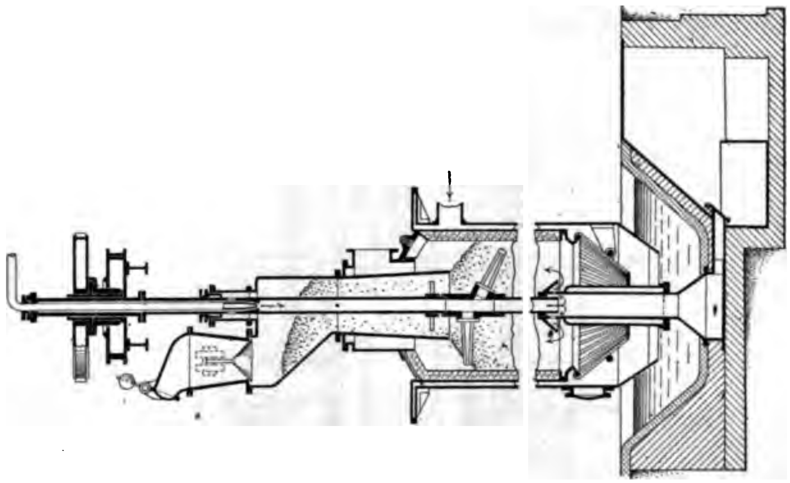

FIG. 15 MOND PRODUCER COMBINED WITH TALBOT STIRRER

Dr. Mond himself and is shown in Fig. 15. This apparatus was tried out thoroughly with various coals of a more or less coking nature. It

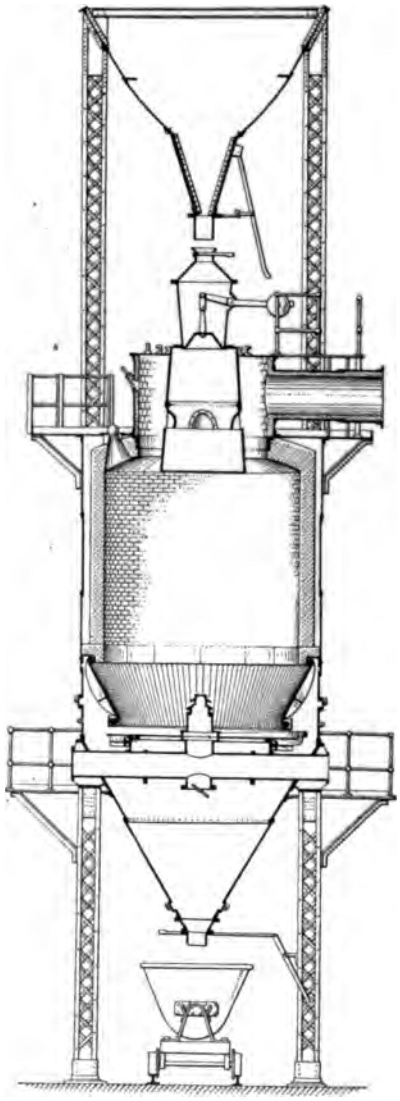

FIG. 16 MOND-TRUMP PRODUCER WITH DRY ASH DISCHARGE

was very costly to install and its operation was not without difficulties. The writer does not know of any more producers of this type being built beyond the first.

36 Another attempt to adapt revolving producers to ammonia recovery plants was one made by the author, but it was not sufficiently successful to warrant general adoption. The great depth of fuel requisite for ammonia recovery renders the operation of mechanical

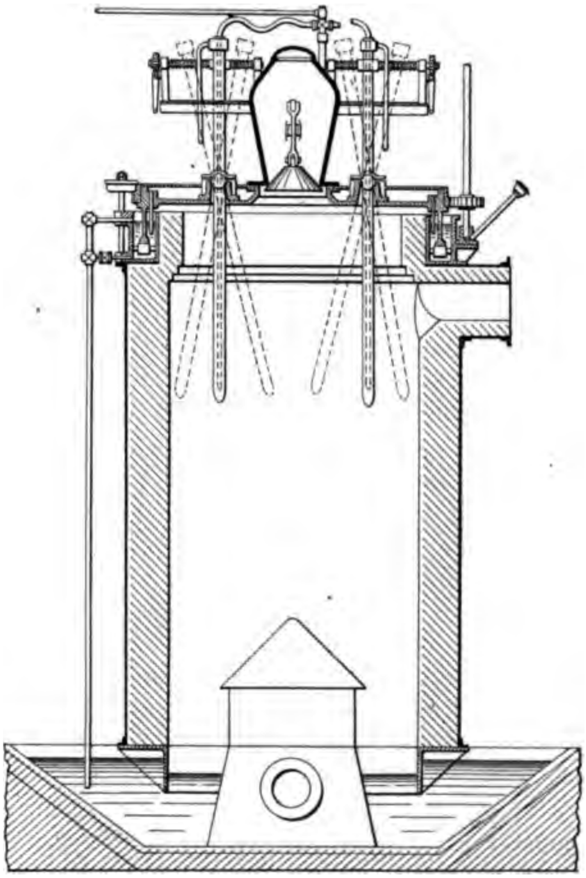

FIG. 17 A. B. DUFF'S PATENT MECHANICAL STIRRER

producers exceedingly difficult. One of the difficulties found by the writer in practice was that, when using a caking coal, the producer revolved while the coal remained more or less stationary, resulting only in the grinding of the coal at the periphery of the producer.

37 Another step in this direction was the combination with the producer of the mechanical ash removing apparatus designed by Mr.

Trump. This combination is illustrated in Fig. 16 and has been adopted on a somewhat large scale. A battery of these so-called Mond-Trump producers was built in England and at the time of its installation the writer was hopeful that the combination, though very costly, would prove to be a valuable development; however, the information to hand regarding it is not very encouraging.

38 Still another proposition is that made by A. B. Duff, in which a mechanically operated stirring poker is utilized in the producer (Fig. 17). This worked very well indeed when tried with Scotch washed nut and the writer believes it has been further adopted for use with this or similar coals. Scotch coal does not cake, however, and therefore with such coal there appears to be insufficient justification for incurring the additional cost of this stirring gear, except it be for the purpose of increasing the rate of gasification. The writer has had a great deal of experience with Scotch nut in stationary producers and has never had the slightest trouble with this coal. He would therefore be interested to know the results obtained with this device when applied to English caking coals, and still more to those in this country.

39 After the previous mentioned failure of the revolving shell producer, the author carried on further trials of mechanical agitation and ash removal. These trials met with success and resulted in the design of producer based on the principle which has been so largely utilized for ordinary hot gas producers in Europe, where the rotary grate and the mechanical ash removal have been further constructionally improved and very widely introduced both by Kerpely of Vienna and by the writer's German licensees. On the basis of the last-named firm's designs as adopted for hot gas producers, the system has been applied to ammonia recovery, material modifications being of course necessary. These modifications are to provide a vastly increased volume of air and steam, a deeper fuel bed, superheating of the blast of air and steam, increased pressure of the air blast and consequently deeper water lute, etc. Plants on this system have been built by the Badische Anilin und Soda-Fabrik, of Ludwigshafen (Fig. 18), and by the German Government at Heinitz (Fig. 19). Several others are under construction. These represent the latest type of Lymn plant adopted in large scale practice.

40 The plants in operation have worked well and a resumé of the operating results of one plant is given in Table 1. These results are taken from the daily log sheets of a plant now gasifying about 80

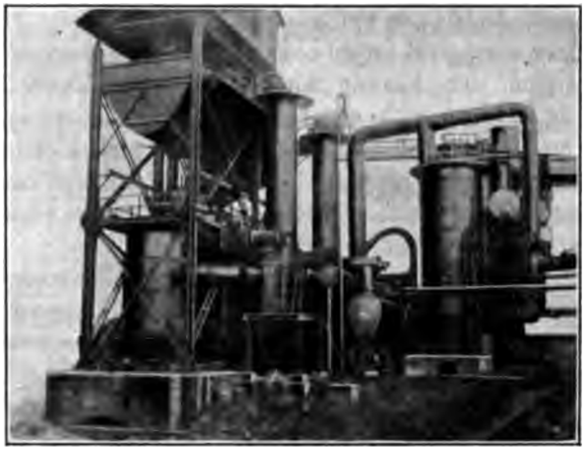

FIG. 18 FIRST SECTION OF A 13,000 H.P. LYMN SYSTEM PRODUCER GAS AND BY-PRODUCT RECOVERY PLANT

FIG. 19 5000 H.P. LYMN SYSTEM PRODUCER GAS AND BY-PRODUCT RECOVERY PLANT DURING ERECTION tons of coal per 24 hours, the gas being used for driving four 1300 h.p. gas engine electric sets and also for firing furnaces. The coal in use is common slack and brown-coal at an average price of 12 to 13 shillings per ton. The heating value of the coal is 10,400 B.t.u. per lb. (6000 kg. cal. per kg.) and the nitrogen contained in the coal averages 0.80 per cent. These records cover two periods of four weeks, each being selected quite haphazard. In spite of the high cost and low nitrogen content of the coal, the cost of gas per kw-hr. worked out at only 0.55 pfg. or 0.13 cents, an interesting result.

41 There is one point in connection with this industry which I think deserves considerable attention. It is well known that the amount of steam generally used in these plants with normal coal is approximately 2½ tons for every ton of coal gasified. Of this amount, up to two-fifths is recovered from the heat of the gases (i.e., during the gas cooling and air saturating cycle of operations) in a modern and properly designed plant. The remainder, 1½ tons, has, however, to be made by direct coal-fired boilers or other means. Needless to say the provision of separate boilers involves a considerable charge on the operating costs of the plant, and it should therefore always be one's endeavor to obtain as large a quantity of steam as possible in the form of waste steam at practically atmospheric pressure (which is quite sufficient) or to raise such steam by utilizing waste heat.

42 In connection with gas power plants, the steam can be made by utilizing the heat of the exhaust gases from the gas engines. This is a problem to which the author has devoted considerable attention. and in the plant referred to above, all the steam is produced in special boilers of his own design which are heated by the exhaust from the gas engines. In this particular installation there are four boilers, each attached to a 1300-h.p. gas engine; and each raising 2 to 3 lb. of steam per h.p-hr. This amount of steam is 25 per cent more than that required for the gasification of the coal.

43 It may be suggested that when boilers are utilized for this purpose there is danger of corrosion, but four years' full-time operation is sufficient criterion that this is not so. The writer's experience is that satisfactory operation is merely a question of properly dimensioning the boilers.

44 The above mentioned plant is the first in the world to be absolutely self-contained as far as steam is concerned and great strides are now being made in the Lymn plants in the matter of utilizing the heat of waste gases from all kinds of operations for the production of steam.

45 At all events, it is important to realize that where coal is expensive gas engines are obviously the more economical to adopt, and where coal is cheap the by-products more than pay for the coal

TABLE 1 ACTUAL OPERATING RESULTS OF POWER GAS PLANT (LYMN SYSTEM) DRIVING LARGE GAS ENGINES AND FIRING FURNACES

First Period of 4 Weeks	Total	Average per Day of 24 Hours	General Average
Coal consumption of the gas plant	1,806 tons	64.6 tons	Per kw-hr. 1.58 lb. (0.72 kg.)
Power produced (kw-hr.)	1,889,740	....	Per hr. 2812 kw.
Yield of sulphate of ammonia	49.11 tons	1.76 tons	Per ton coal 60 lb. (27.1 kg.)
Yield of tar (containing water)	189.7 tons	6.78 tons	Per ton coal 230 lb. (105 kg.)
Average heating value of the gases	155 B.t.u. per cu. ft. (1380 cal. per cu. m.)		
Sulphur contained in the gas (average)	0.63 grams per cu. m.		
Tar contained in the gas (average)	0.04 grams per cu. m.		
The auxiliary machines consumed regularly 71 kw.			
Including 10 per cent depreciation the gas costs per kw-hr. work out at 0.069 penny			
Second period of 4 weeks			
Coal consumption of the gas plant	1,967 tons	70.2 tons	Per kw-hr. 1.72 lb. (0.78 kg.)
Power produced (kw-hr.)	1,899,600	....	Per hr. 2830 kw.
Yield of sulphate of ammonia	54.3 tons	1.94 tons	Per ton coal 61 lb. (27.6 kg.)
Yield of tar (containing water)	231.7 tons	8.27 tons	Per ton coal 257 lb. (117 kg.)
Average heating value of the gases	154 B.t.u. per cu. ft.		
Sulphur contained in the gas (average)	0.38 grams per cu. m.		
Tar contained in the gas (average)	0.057 grams per cu. m.		
The auxiliary machines consumed regularly 78 kw.			
Including 10 per cent depreciation the gas costs per kw-hr. work out at 0.07 penny			

NOTE:—The nitrogen efficiency during these two periods was 70 per cent. It is frequently 75 per cent.

and the gas can be made for nothing or even at a profit. In the latter case, it does not matter very much what quantities of gas are used per h.p., and under such circumstances gas-fired steam plants become as profitable as gas engine plants or more so.

TABLE 2 ESTIMATES OF WORKING COSTS FOR (I) A 2000 H.P. POWER GAS INSTALLATION, (II) A 4500 KW. PRODUCER GAS PLANT, AND (III) A PRODUCER GAS PLANT FOR CONTINUOUS GASIFICATION OF 500 TONS OF COAL DAILY

CONDITIONS LOAD CONDITIONS OF PLANT	I Power	II Power	III Heating
Hours of full load per annum	4000	8500	8760
Size of plant in b.h.p. or kw. or long tons of coal per day	2000 b.h.p. (1350 kw.)	6600 b.h.p. (4500 kw.)	500 tons
Cost of coal in dollars per short ton	2	1	2
Heating value of coal in B.t.u. per lb	12,600	12,600	12,600
Nitrogen content of coal in per cent	1.3	1.3	1.3
Cost of sulphuric acid (140 deg. Twaddell) in dollars per short ton	9	9	9
Value of sulphate of ammonia in dollars per short ton	55	55	55
Value of tar in dollars per short ton	5	5	5
Heat consumption of gas engines in B.t.u. per kw-hr.	14,900	14,300	....
COST OF PLANT			
Producer power gas and ammonia recovery plant (Lymn System) in dollars	40,600	126,500	605,000
Buildings and foundations for same	4,400	12,000	55,000
Complete gas engine installation consisting of gas engines, dynamos, all auxiliary machines, exhaust boilers, overhead crane, etc., in dollars	88,000	335,500 (spare set of 2250 kw.)	....
Buildings and foundations in dollars	13,000	48,000	....
Total cost of installation	138,000	522,000	660,000
WORKING DATA			
Amount of kw-hr. per annum	5,400.000	38,250.000	....
Tons of coal used (including stand-by losses) per annum	4830	29,840	204,400
Tons of sulphate of ammonia recovered per annum	206	1346	9210
Tons of tar recovered per annum	230	1500	10,500
Tons of sulphuric acid consumed per annum	190	1280	8800
Rate of amortization on machines and plant in per cent per annum	12	12	12
Rate of amortization on buildings and foundations in per cent per annum	6	6	6
ANNUAL WORKING COSTS IN DOLLARS OF PRODUCER GAS AND AMMONIA RECOVERY PLANT (LYMN SYSTEM)			
Cost of coal	9660	29,840	408,800
Labor	5600	16,630	49,500
Repairs and maintenance	1230	3780	18,000
Oil, waste, lighting, etc	680	2990	15,330
Sulphuric acid	1710	11,520	79,200
Depreciation and interest	5132	15,900	75,900
Total debit	24,012	80,560	646,730
Credit by sulphate of ammonia	11,330	74,030	506,550
Credit by tar	1150	7500	52,500
Total credit	12,480	81,530	559,050
Total annual cost of gas	11,012	970 Profit	87,680
Cost of gas in cents per 1000 cu. ft. (heating value 150 B.t.u. per cu. ft. net)	2.10	0.03	0.32

(Continued on next page)

TABLE 2 ESTIMATES OF WORKING COSTS FOR (I) A 2000 H. P. POWER GAS INSTALLATION, (II) A 4500 KW. PRODUCER GAS PLANT, AND (III) A PRODUCER GAS PLANT, FOR CONTINUOUS GASIFICATION OF 500 TONS OF COAL DAILY

(Continued from previous page)

	I Power	II Power	III Heating
ANNUAL WORKING COSTS OF GAS ENGINE PLANT (Based upon first class German Gas Engine practice)	Dollars per Annum	Dollars per Annum	
Cost of gas as above	11,012	970 Profit	
Repairs	1250	5170	
Oil, waste, water	840	4420	
Labor at American rates	3590	10,370	
Depreciation and interest	10,380	43,180	
Total costs	27,072	62,170	
Total cost of power in cents per kw-hr	0.50	0.16	
Total cost of power in dollars per kw-year	....	13.80	
Total cost of power in dollars per h.p.-year	....	10.30	

46 To give a general idea of the adaptability of ammonia recovery plants for power as well as for heating purposes, three estimates of working costs have been made for (I) a 2000 h.p. power gas installation, coupled together with gas engines and working 4000 hr. per annum, (II) a 4500 kw. producer gas plant, coupled together with gas engines and dynamos furnishing current for say electrochemical purposes, working 8500 hr. per annum, and erected near a colliery where the coal will be cheap, and (III) a producer gas plant for a daily and continuous gasification of 500 tons of coal, the gas being used say for firing steel furnaces. The estimates are given in Table 2. The working costs of these three plants are based upon the actual results in practice referred to above. It has been assumed that the cost of labor is 50 per cent and the cost of apparatus is 25 per cent more than in England and Germany.

47 It will be realized that for industries such as electrochemical plants requiring a large amount of power, it is quite unnecessary to have recourse to water powers which are almost invariably situated in localities quite unsuitable as manufacturing sites and which therefore require long, costly and unreliable transmission systems subject to the dangers of sleet, wind and electrical failures. Every power user who depends upon an uninterrupted supply of current for the success of his operations would gladly dispense with this transmission, even were its high cost of no importance.

48 In considering the development of ammonia recovery plants, the statements made so far have referred to the treatment of coal, which is obviously the most used combustible. They may also be taken, however, as applying to waste coal containing a high percentage of ash, as well as to other poor grade coals, lignite, coke breeze, etc.

49 Coke breeze, as obtained in the manufacture of lighting gas, has now a particularly advantageous application in these plants. It is well known that as a general rule the retorts in gas works are heated by means of good trade coke which has a high selling value, but the coke breeze which is sieved out is practically a waste product. This

FIG. 20 PEAT POWER GAS PLANT WITH AMMONIA RECOVERY AT PONTEDERA, ITALY

substance can now be dealt with, producing all the gas for firing the retorts together with about 60 lb. (value $1.20) of sulphate of ammonia per ton of breeze. Furthermore, much good coke is thus set free for sale to the public at a high value. A large plant is already operating on these lines in England and is very successful and profitable.

50 There are, however, other combustibles whose use in producer plants is restricted on account of the high percentage of water they contain. Such in particular are peat and wet brown-coal.

51 The writer, as technical manager to The Power-Gas Corporation, was able to apply successfully the Mond Gas process to the treatment of peat between 1904 and 1907. The drying of peat is a most

difficult matter and in view of this fact it is interesting to note that today it is possible to produce regularly power gas and by-products from peat containing up to 60 per cent water. This peat can be obtained by relatively short periods of drying in the atmosphere in practically all countries. Evidences of success in this matter are the facts that a 20-ton plant was erected in Germany some years ago to demonstrate the advantages of this process, and another plant (Fig. 20) dealing with 100 tons of peat per day and producing sulphate of ammonia and power gas has been in operation in northern Italy for about three years. In the latter case a further peat bog has now been purchased and a second and larger plant built on it.

52 The quantity of ammonium sulphate produced per ton of peat depends upon the nitrogen content and varies between 70 and 220 lb. per ton of dry peat gasified. Where peat with about 2 per cent nitrogen is available, one can obtain a large profit simply from the ammonium sulphate, regarding the gases as a by-product. Indeed, with peat which contains little nitrogen, gas can in most cases be produced without cost. Other by-products which can be produced from peat are tar (which contains much paraffin), acetate of lime, etc.

53 The application of the Mond process to peat as worked out in England has at times been erroneously referred to as the Frank-Caro process. As a matter of fact, however, only one plant was built according to Frank and Caro's designs. This was at Osnabruck, Germany, and it was shut down after twelve months' operation.

54 About sixty by-product producer gas plants are already built having a yearly fuel capacity of approximately 2,000,000 tons. These are distributed among Great Britain (which has most of them), Germany, Italy, Spain, China, Japan and this country. The gas from them is being used not only for power production but also for all kinds of industrial heating operations, such as reheating furnaces, forging furnaces, annealing furnaces, steel furnaces, core stoves, crucible heating, galvanizing baths, gas works retort firing, spelter furnaces, glass works operations, evaporating brine, calcining operations, roasting operations, etc.

55 It may be taken for granted that few industrial processes lend themselves to introduction into a foreign country without alteration to meet local conditions and fuels. Some mistakes have been made in the past owing to too rigid adherence to European designs. The recognition of these facts has led to the installation of a demonstration plant of the author's design in the Pittsburgh district, in

which trials of American fuels are being carried out on a large scale.

56 It may be contended by many that the adoption of a large number of ammonia recovery plants would run down the sulphate of ammonia market, but it must be borne in mind that in England, where by-product producer gas plants have made more progress than in all other countries together, the proportion of sulphate of ammonia made by this means amounts to only 13 per cent. The remainder comes from lighting gas plants and coke ovens which in Germany, and also

TABLE 3 THE NITROGEN CONTENT OF AMERICAN COALS

State	Amount of Coal Samples Analysed for Nitrogen Content	Average Content of Nitrogen in Per Cent on Theoretically Dry Fuel	State	Amount of Coal Samples Analysed for Nitrogen Content	Average Content of Nitrogen in Per Cent on Theoretically Dry Fuel
Alabama......	37	1.42	Ohio.........	15	1.30
Alaska........	45	1.14	Oklahoma.....	20	1.63
Arizona.......	1	1.25	Oregon.......	1	1.42
Arkansas......	18	1.41	Pennsylvania..	106	1.28
California.....	4	0.97	Rhode Island..	10	0.19
Colorado......	176	1.36	Tennessee.....	15	1.46
Georgia.......	1	1.13	Texas........	5	1.16
Illinois........	67	1.28	Utah.........	32	1.11
Indiana.......	23	1.27	Virginia.......	27	1.29
Iowa.........	15	1.16	Washington....	169	1.58
Kansas.......	30	1.24	West Virginia..	265	1.37
Kentucky.....	22	1.42	Wyoming.....	192	1.30
Maryland.....	15	1.71			
Michigan......	2	1.38			
Missouri......	40	1.11	Total.....	1467	....
Montana......	81	1.03			
New Mexico...	27	1.29			
North Dakota.	6	1.15	Average	....	1.325

to a great extent in England, are producing nearly as much sulphate of ammonia as is possible.

57 The consumption of sulphate of ammonia is steadily on the increase, although the market has fluctuated considerably during the past year.[1] This substance must therefore be supplied from other sources than those which have so largely supplied it up to now. With reference to the production of nitrogenous fertilizers, there is room for a very large increase in the production of sulphate of ammonia from gas producer plants, in spite of the increasing production of synthetic nitrogenous fertilizers.

[1]1914.

58 Regarding the nitrogen content of American coals, Table 3 represents the average of some 1500 analyses made by that thorough body of workers, the Department of Mines of the U. S. Geological Survey. More than 560 million tons of coal per annum, containing on an average about 1.3 per cent nitrogen, are produced in this country. Imagine this quantity of coal being converted into producer gas and the ammonia recovered from the whole of it, and deduct the amount of sulphate of ammonia which is already produced. The remarkable result is arrived at that about 25 million tons of sulphate of ammonia, having a value of 600 million dollars, are wasted per annum. Surely it is worth while to consider recovering at all events a small portion of this, especially when it is realized that every dollar spent by agriculturists in sulphate of ammonia means improved crops.

No. 1525

REPORT OF THE COMMITTEE ON STANDARDIZATION OF SPECIAL THREADS FOR FIXTURES AND FITTINGS

ON STRAIGHT PIPE THREADS[1]

DEFINITIONS OF TERMS USED

The term "outside diameter" used in this report refers in the case of male threads to the top of the threads or largest diameter, and in the case of female threads to the bottom of the threads or largest diameter. The term "root diameter" similarly refers to the smallest diameter. The "pitch diameter" is of course determined by subtracting from the outside diameter of a theoretical full V-thread the single depth of the thread. Fig. 1 illustrates these various diameters.

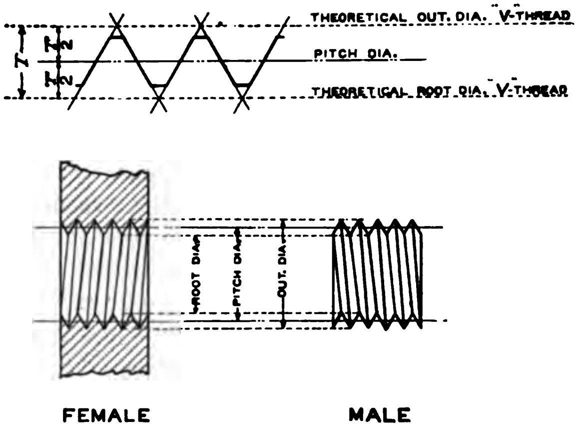

FIG. 1 OUTSIDE, PITCH AND ROOT DIAMETERS

[1]See Paper No. 1474 for report of this committee on Rolled Threads for Screw Shells of Electric Sockets and Lamp Bases.

Received by the Council, November 13, 1915 and ordered printed. Presented at the Annual Meeting, December, 1915, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

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CONDITIONS LOAD CONDITIONS OF PLANT	I Power	II Power	III Heating
Hours of full load per annum	4000	8500	8760
Size of plant in b.h.p. or kw. or long tons of coal per day	2000 b.h.p. (1350 kw.)	6600 b.h.p. (4500 kw.)	500 tons
Cost of coal in dollars per short ton	2	1	2
Heating value of coal in B.t.u. per lb	12,600	12,600	12,600
Nitrogen content of coal in per cent	1.3	1.3	1.3
Cost of sulphuric acid (140 deg. Twaddell) in dollars per short ton	9	9	9
Value of sulphate of ammonia in dollars per short ton	55	55	55
Value of tar in dollars per short ton	5	5	5
Heat consumption of gas engines in B.t.u. per kw-hr	14,900	14,300	....
COST OF PLANT			
Producer power gas and ammonia recovery plant (Lymn System) in dollars	40,600	126,500	605,000
Buildings and foundations for same	4,400	12,000	55,000
Complete gas engine installation consisting of gas engines, dynamos, all auxiliary machines, exhaust boilers, overhead crane, etc., in dollars	88,000	335,500 (spare set of 2250 kw.)	....
Buildings and foundations in dollars	13,000	48,000	....
Total cost of installation	138,000	522,000	660,000
WORKING DATA			
Amount of kw-hr. per annum	5,400.000	38,250.000	....
Tons of coal used (including stand-by losses) per annum	4830	29,840	204,400
Tons of sulphate of ammonia recovered per annum	206	1346	9210
Tons of tar recovered per annum	230	1500	10,500
Tons of sulphuric acid consumed per annum	190	1280	8800
Rate of amortisation on machines and plant in per cent per annum	12	12	12
Rate of amortisation on buildings and foundations in per cent per annum	6	6	6
ANNUAL WORKING COSTS IN DOLLARS OF PRODUCER GAS AND AMMONIA RECOVERY PLANT (LYMN SYSTEM)			
Cost of coal	9660	29,840	408,800
Labor	5600	16,630	49,500
Repairs and maintenance	1230	3780	18,000
Oil, waste, lighting, etc	680	2990	15,330
Sulphuric acid	1710	11,520	79,200
Depreciation and interest	5132	15,900	75,900
Total debit	24,012	80,560	646,730
Credit by sulphate of ammonia	11,330	74,030	506,550
Credit by tar	1150	7500	52,500
Total credit	12,480	81,530	559,050
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Cost of gas in cents per 1000 cu. ft. (heating value 150 B.t.u. per cu. ft. net)	2.10	0.03	0.32

(Continued on next page)

dimensions of pipe and certain formulae shown in Fig. 2. In this figure

$$L=\frac{0.8D+4.8}{n} \qquad E=L+2\left(\frac{1}{n}\right)$$

$$R=D-\frac{0.05D+1.9}{n} \qquad A=R+\frac{0.8}{n}$$

$$B=A+\frac{F}{16} \qquad G=A+\frac{E}{16}$$

$$H=\frac{1}{16\,n}$$

5 The sizes shown in Fig. 2 and above described were adopted

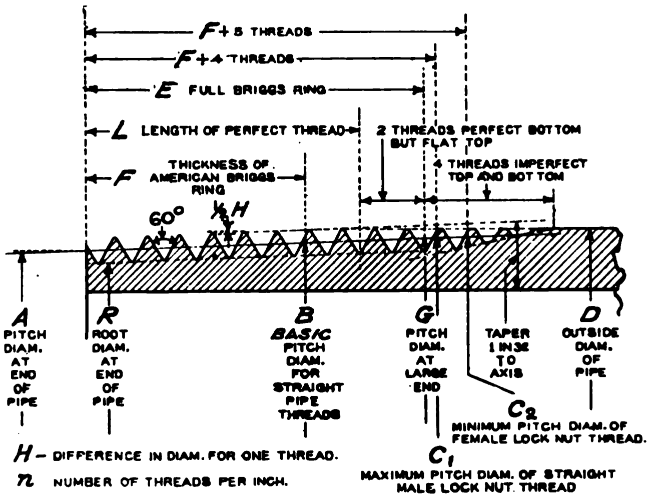

FIG. 2 BASIC STRAIGHT PIPE SIZES

by the Committee of Manufacturers on Standardization of Fittings and Valves on September 17, 1913, except the straight pipe thread and the lock-nut sizes which were later adopted by the same Committee on March 16, 1915. The figures for the various pipe sizes as published by that Committee are given in Table 1, with the addition of columns headed D, G and H. Your Committee heartily endorses the work done by the Committee of Manufacturers on Standardization of Fittings and Valves.

TOLERANCES

6 The maximum tolerances taken on the pitch diameters by

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Size of plant in b.h.p. or kw. or long tons of coal per day	2000 b.h.p. (1350 kw.)	6600 b.h.p. (4500 kw.)	500 tons
Cost of coal in dollars per short ton	2	1	2
Heating value of coal in B.t.u. per lb	12,600	12,600	12,600
Nitrogen content of coal in per cent	1.3	1.3	1.3
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(Based upon first class German Gas Engine practice)	per Annum	per Annum	
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Labor at American rates	3590	10,370	
Depreciation and interest	10,380	43,180	
Total costs	27,072	62,170	
Total cost of power in cents per kw-hr	0.50	0.16	
Total cost of power in dollars per kw-year	....	13.80	
Total cost of power in dollars per h.p.-year	....	10.30	

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Fig. 20 Peat Power Gas Plant with Ammonia Recovery at Pontedera, Italy

substance can now be dealt with, producing all the gas for firing the retorts together with about 60 lb. (value $1.20) of sulphate of ammonia per ton of breeze. Furthermore, much good coke is thus set free for sale to the public at a high value. A large plant is already operating on these lines in England and is very successful and profitable.

50 There are, however, other combustibles whose use in producer plants is restricted on account of the high percentage of water they contain. Such in particular are peat and wet brown-coal.

51 The writer, as technical manager to The Power-Gas Corporation, was able to apply successfully the Mond Gas process to the treatment of peat between 1904 and 1907. The drying of peat is a most

difficult matter and in view of this fact it is interesting to note that today it is possible to produce regularly power gas and by-products from peat containing up to 60 per cent water. This peat can be obtained by relatively short periods of drying in the atmosphere in practically all countries. Evidences of success in this matter are the facts that a 20-ton plant was erected in Germany some years ago to demonstrate the advantages of this process, and another plant (Fig. 20) dealing with 100 tons of peat per day and producing sulphate of ammonia and power gas has been in operation in northern Italy for about three years. In the latter case a further peat bog has now been purchased and a second and larger plant built on it.

52 The quantity of ammonium sulphate produced per ton of peat depends upon the nitrogen content and varies between 70 and 220 lb. per ton of dry peat gasified. Where peat with about 2 per cent nitrogen is available, one can obtain a large profit simply from the ammonium sulphate, regarding the gases as a by-product. Indeed, with peat which contains little nitrogen, gas can in most cases be produced without cost. Other by-products which can be produced from peat are tar (which contains much paraffin), acetate of lime, etc.

53 The application of the Mond process to peat as worked out in England has at times been erroneously referred to as the Frank-Caro process. As a matter of fact, however, only one plant was built according to Frank and Caro's designs. This was at Osnabruck, Germany, and it was shut down after twelve months' operation.

54 About sixty by-product producer gas plants are already built having a yearly fuel capacity of approximately 2,000,000 tons. These are distributed among Great Britain (which has most of them), Germany, Italy, Spain, China, Japan and this country. The gas from them is being used not only for power production but also for all kinds of industrial heating operations, such as reheating furnaces, forging furnaces, annealing furnaces, steel furnaces, core stoves, crucible heating, galvanizing baths, gas works retort firing, spelter furnaces, glass works operations, evaporating brine, calcining operations, roasting operations, etc.

55 It may be taken for granted that few industrial processes lend themselves to introduction into a foreign country without alteration to meet local conditions and fuels. Some mistakes have been made in the past owing to too rigid adherence to European designs. The recognition of these facts has led to the installation of a demonstration plant of the author's design in the Pittsburgh district, in

Minimum pitch diameter $=B-T$

Minimum outside diameter $=B-T+\frac{1}{2}\left(\frac{0.86603}{n}\right)$

FEMALE FITTINGS

Maximum pitch diameter $=B+T$

Maximum root diameter $=B+T-\frac{1}{2}\left(\frac{0.86603}{n}\right)$

Minimum pitch diameter $=B$

Minimum root diameter $=B-\frac{0.64952}{n}$

Minimum outside diameter $=B+\frac{0.64952}{n}$

Tables 2 and 3 are given for convenience, being derived from the above.

TABLE 2 SPECIAL STRAIGHT FIXTURE PIPE THREADS. MALE THREADS

DIMENSIONS IN INCHES

Pipe Size	Threads per In.	Maximum Diameter			Minimum Diameter		Tolerance	Minimum Length of Thread
		Outside	Pitch *Basic*	Root	Outside	Pitch	Pitch Diameter	
1/8	27	0.3989	0.3748	0.3507	0.3868	0.3708	0.004	0.2638
1/4	18	0.5260	0.4899	0.4538	0.5089	0.4849	0.005	0.4018
3/8	18	0.6631	0.6270	0.5909	0.6460	0.6220	0.005	0.4078
1/2	14	0.8248	0.7784	0.7320	0.8033	0.7724	0.006	0.5337
3/4	14	1.0353	0.9889	0.9425	1.0138	0.9829	0.006	0.5457
1	11½	1.2951	1.2386	1.1821	1.2692	1.2316	0.007	0.6828

TABLE 3 SPECIAL STRAIGHT FIXTURE PIPE THREADS. FEMALE THREADS

DIMENSIONS IN INCHES

Pipe Size	Threads per In.	Maximum Diameter		Minimum Diameter			Tolerance	Minimum Length of Thread
		Pitch	Root	Outside	Pitch *Basic*	Root	Pitch Diameter	
1/8	27	0.3788	0.3628	0.3989	0.3748	0.3507	0.004	0.2638
1/4	18	0.4949	0.4709	0.5260	0.4899	0.4538	0.005	0.4018
3/8	18	0.6320	0.6080	0.6631	0.6270	0.5909	0.005	0.4078
1/2	14	0.7844	0.7535	0.8248	0.7784	0.7320	0.006	0.5337
3/4	14	0.9949	0.9640	1.0353	1.9889	0.9425	0.006	0.5457
1	11½	1.2456	1.2080	1.2951	1.2386	1.1821	0.007	0.6828

ELECTRIC CAPS

14 While the sizes given in Tables 2 and 3 should be satisfactory on gas burners, fixture nipples, insulating joints, hickeys and the general run of similar fittings for all female threads and for male

threads when straight threads are necessary, they are not safely usable on electric socket caps as the fit with the nipple or coupling should be sufficiently loose to permit the cap to be easily screwed down to the shoulder without strain.

15 *Male Electric Socket Caps.* Pipe couplings into which these caps are screwed are commercially made with a tolerance of one thread each side of the notch on the American Briggs Standard as adopted by the Committee of Manufacturers on Standardization of Fittings and Valves. Therefore, it is desirable that the maximum be less than *basic* size by 0.001 in. more than one thread. Referring to Fig. 2,

$$\text{Maximum pitch diameter} = B-(H+0.001)$$

As the tolerances as well as the relative root and outside diameters will be as in Par. 6 to 9 and 13, Table 4 is derived for convenience; in this the minimum lengths of thread recommended are also shown.

TABLE 4 SPECIAL STRAIGHT ELECTRIC FIXTURE THREADS. MALE ELECTRIC CAPS, ETC.

DIMENSIONS IN INCHES

Pipe Size	Threads per In.	Maximum Diameter			Minimum Diameter		Tolerance	Minimum Length of Thread
		Outside	Pitch	Root	Outside	Pitch	Pitch Diameter	
⅛	27	0.3956	0.3715	0.3474	0.3836	0.3675	0.004	0.250
¼	18	0.5215	0.4854	0.4493	0.5044	0.4804	0.005	0.281
⅜	18	0.6586	0.6225	0.5864	0.6415	0.6175	0.005	0.281
½	14	0.8193	0.7729	0.7265	0.7978	0.7669	0.006	0.375
¾	14	1.0298	0.9834	0.9370	1.0083	0.9774	0.006	0.375
1	11½	1.2887	1.2322	1.1757	1.2628	1.2252	0.007	0.500

16 *Female Electric Socket Caps.* The ⅛-in. size is most largely used and owing to the oversize fixture nipples now in use an arbitrary minimum pitch diameter of 0.3790 in. is recommended. This size will not be too loose on standard nipples. For sizes above ⅛ in. a minimum pitch diameter one thread larger than *basic* is taken:

$$\text{Minimum pitch diameter} = B+H$$

17 Tolerances on all sizes should be same as established in Par. 6 to 9 and 13, and the root and outside diameters should be proportional. Table 5 is derived for convenience in which the minimum length of thread recommended is shown.

18 Your Committee, therefore, recommends the use of *straight pipe threads* and *locknut threads* as adopted by the Committee of Manufacturers on Standardization of Fittings and Valves and as described in Par. 4 to 12 of this report, and further recommends the

use of the special straight threads described in Par. 8 to 13 and thereafter in this report. It further recommends that there be deposited with the Bureau of Standards, Washington, D. C., master gages, the expense of such gages to be borne by the manufacturers.

TABLE 5 SPECIAL STRAIGHT ELECTRIC FIXTURE THREADS. FEMALE ELECTRIC CAPS, ETC.

DIMENSIONS IN INCHES

Pipe Size	Threads per In.	Maximum Diameter		Minimum Diameter			Tolerance	Minimum Length of Thread
		Pitch	Root	Outside	Pitch	Root	Pitch Diameter	
1/8	27	0.3830	0.3670	0.4031	0.3790	0.3549	0.004	0.2638
1/4	18	0.4984	0.4744	0.5295	0.4934	0.4573	0.005	0.4018
3/8	18	0.6355	0.6115	0.6666	0.6305	0.5944	0.005	0.4078
1/2	14	0.7889	0.7580	0.8293	0.7829	0.7365	0.006	0.437
3/4	14	0.9994	0.9685	1.0398	0.9934	0.9470	0.006	0.437
1	11½	1.2510	1.2134	1.3005	1.2440	1.1875	0.007	0.500

19 The Straight Pipe Threads recommended by the Committee of Manufacturers on Standardization of Fittings and Valves are known by that name. Your Committee, therefore, recommends the use of this name for these standards only, and that the special standards above described be known as Special Straight Fixture Pipe Threads and Special Straight Electric Fixture Threads respectively.

Respectfully submitted,

EDWARD S. SANDERSON, *Chairman*
WM. J. BALDWIN
STANLEY G. FLAGG, JR.
CHAS. R. HARE
HARRY E. HARRIS
A. H. MOORE
W. R. WEBSTER
GEORGE B. THOMAS, *Secretary*

No. 1526

REPORT OF THE POWER TEST COMMITTEE

ON RULES FOR CONDUCTING PERFORMANCE TESTS OF POWER PLANT APPARATUS

To The Council of The American Society of Mechanical Engineers:

The Power Test Committee which was appointed under the Council's resolution of April 13, 1909 to

> "revise the present testing codes of the Society relating to boilers, pumping engines, locomotives, steam engines in general, internal combustion engines, and apparatus and fuels therefor, and to extend these codes so as to apply to such power generating apparatus as the present codes do not cover, including water power, bringing them into harmony with each other and with the best practice of the day,"

begs to submit the results of its work in the accompanying set of revised codes and appendices.

The general plan of the revision was devised by Mr. Barrus, who at the Committee's request soon after its appointment, submitted the first draft of a form of report which was afterwards substantially agreed upon and carried out. The first draft was discussed by the Committee, and later the whole matter was referred to a subcommittee consisting of Messrs. Kent, Wood, and Barrus, who were appointed at a meeting of the General Committee in December 1911. The results of the Sub-Committee's work were approved by a majority of the full committee and submitted to the Society in a preliminary report which was published in The Journal of November 1912.

The preliminary report was presented at the annual meeting in December 1912, and was widely discussed and criticised both verbally and in writing. Since that time it has been further discussed

Presented at the Annual Meeting, December, 1915, of The American Society of Mechanical Engineers. Received by the Council, December 7, 1915, and ordered printed.

by correspondence. In the light of the criticisms attending these discussions, the sub-committee has again made an extensive revision, which has been approved by a majority of the full committee, and it is now submitted as a final report for publication in the Transactions.

The members of the Sub-Committee have applied themselves diligently to the work, having held no less than thirty-eight meetings, each lasting from 6 to 12 hours, all three members being usually present. The work done by the other members of the committee has been carried on mainly by correspondence. It is due to those who have furnished discussions or otherwise assisted the Sub-Committee to say that the members of the Society whose names appear on the list attached herewith (and a few engineers who are not members) have commented upon the work either verbally or in writing; and everyone of their comments has received consideration. In a large number of instances the codes have been modified in accordance with the suggestions thus made.

A report of this kind cannot be expected to deal with all the refinements of laboratory tests, or to conform to the methods practiced by every individual who engages in testing work; but it is intended to set forth the correct governing principles and serve the practical purposes of the engineering public.

The Committee originally organized by electing Dr. Jacobus as chairman. Dr. Chas. E. Lucke, one of the original members of the Committee, resigned early in 1912, and the vacancy thus made was not filled. Owing to pressure of his business duties, Dr. Jacobus resigned the chairmanship in December 1911, but the Committee laid his resignation on the table and appointed Mr. Barrus vice-chairman to serve as active chairman in his place. Later, the Committee accepted the resignation and promoted Mr. Barrus to the Chairmanship, this action being in due time approved by the Council.

Respectfully submitted,

GEO. H. BARRUS, *Chairman*.	EDWARD F. MILLER	COMMITTEE ON POWER TESTS
D. S. JACOBUS	L. P. BRECKENRIDGE	
WILLIAM KENT	ARTHUR WEST	
ALBERT C. WOOD	EDWARD T. ADAMS	

New York, Nov. 15, 1913.

The Preliminary Report was discussed by the engineers named below:

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Arnold, B. H.
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Baumgarten, L. Erwin
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Ennis, W. D.
Ferguson, Hardy S.
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Fisher, E. C.
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Marquis, F. W.
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Thompson, H. L.
Thorkelson, H. J.
Weymouth, C. R.
Whitham, J. M.
Wood, Arthur J.
Yarnall, D. Robert
Young, C. D.
Young, G. A.

Postscript relating to final revision completed in Aug. 1915.

The report of the Committee with the foregoing introduction was printed in December 1914, marked "Subject to revision for typographical errors, errors of diction, and obvious mistakes," and copies were submitted for further comment to those who had taken part in the previous discussion. Copies were also sent to the three Am. Soc. M. E. committees pertaining to Railroads, Gas Power, and Bureau of Engineering Standards (now termed Standardization Committee), and to the Standards Committee, A. I. E. E., the Bureau of Mines, and others interested in the subject, all of whom were likewise asked for comments. Suggestions received from these sources have been availed of in preparing the final revision now submitted, and all the modifications introduced have received the unanimous approval of the Committee.

Geo. H. Barrus, *Chairman*

Boston, December 4, 1915.

(At the meeting of the Council, December 7, 1915, it was voted that a permanent committee be appointed to interpret the rules when called upon to do so, to make such revisions as may be found desirable, and to modify the rules to meet new conditions as they may arise in the future; this new committee to hold meetings from time to time at which all interested parties may have an opportunity to present suggestions.)

CALVIN W. RICE,
Secretary.

New York, Jan. 3, 1916.

TABLE OF CONTENTS

FIRST SECTION: GENERAL MATTERS

SECOND SECTION: INDIVIDUAL CODES

THIRD SECTION: APPENDICES

FIRST SECTION: GENERAL MATTERS

PART I

INSTRUCTIONS REGARDING TESTS IN GENERAL

OBJECT

1 Ascertain the specific object of the test, and keep this in view not only in the work of preparation, but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

2 If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, the methods of testing to be followed, corrections to be made in case the conditions actually existing during the test differ from those specified, and as to all other matters about which dispute may arise, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted:

Determination of capacity and efficiency, and how these compare with standard or guaranteed results

Comparison of different conditions or methods of operation

Determination of the cause of either inferior or superior results

Comparison of different kinds of fuel

Determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

PREPARATIONS

A Dimensions

3 Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the apparatus, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of the heating surfaces of boilers and superheaters to be found are those of surfaces in contact with the fire or hot gases. The submerged surfaces in boilers at the mean water level should be considered as water-heating surfaces, and other surfaces which are exposed to the gases as superheating surfaces.

In the case of condensers, and feedwater heaters, the outside surfaces are to be taken. In reheaters and steam jackets, the surfaces to be considered are those exposed to the steam of lower pressure.

The dimensions of engine cylinders should be taken when they are cold, and, if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The clearance of the cylinders may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the water measurement method. (See Appendix No. 1 for water determination of clearance.)

B Examination of Plant

4 Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

In boilers, for example, examine for leakage of tubes and riveted or other metal joints. Note the condition of brick furnaces, grates and baffles. Examine brick walls and cleaning doors for air leaks, either by shutting the damper and observing the escaping smoke or by candle-flame test. Determine the condition of heating surfaces with reference to exterior deposits of soot and interior deposits of mud or scale.

See that the steam main is so arranged that condensed and entrained water cannot flow back into the boiler.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve-seats belonging thereto. Examine for air leaks in exhaust piping, condenser, packings, etc., using vacuum gage or candle-flame test, or by filling the piping, etc., with warm water under a slight head. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak. Determine the condition of the blading, nozzles, and valves in steam turbines, and of buckets, guides and draft-tubes in water turbines.

See Appendices Nos. 2 and 3 for general methods of leakage examination and test.

5 If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects, or defects of operation.

tending to make the result unfavorable should first be remedied; all fouled parts being cleaned, and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

C General Precautions against Leakage

6 In steam tests make sure that there is no leakage through blow-offs, drips, etc., or through any steam or water connections of the plant or apparatus undergoing test, which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that no leakage is going on either out or in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

See Appendix No. 3 for further details.

D Apparatus and Instruments

7 Select the apparatus and instruments specified in the Code of Rules applying to the test in hand, locate and install the same, and complete the preparations for the work in view.

8 The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz., *see that the apparatus and instruments are substantially reliable, and arranged in such a way as to obtain correct data.*

9 A summary is given below, embracing the entire list of apparatus and instruments referred to in the various Codes, with descriptions of their leading features, methods of application and use, and, where needed, methods of calibration; these particulars being supplemented in some instances by further descriptions in the Appendix.

9a Weighing Scales

For determining the weight of coal, oil, water, etc., ordinary platform scales serve every purpose. Too much dependence, however, should not be placed upon their reliability without first calibrating them by the use of standard weights, and carefully examining the knife-edges, bearing plates, and ring suspensions, to see that they are all in good order.

Other scales required in connection with test work are small scales for weighing coal-samples used for moisture tests, and laboratory scales for analyses and calorific determinations pertaining to fuels. Such scales should be sensitive to 1/1000 of the quantity weighed.

For testing locomotives and some classes of marine boilers, where room is lacking, sacks or bags are sometimes required to facilitate the handling of coal, the weighing being done before loading on the tender or delivery to the fire room.

9b Water Weighing and Measuring Apparatus

(1) *Feedwater.* Wherever practicable the feedwater should be weighed, especially for guarantee tests. The most satisfactory and reliable apparatus for this purpose consists of one or more tanks each placed on platform scales, these being elevated a sufficient distance above the floor to empty into a receiving tank placed below, the latter being connected to the feed pump. Where only one weighing tank is used the receiving tank should be of larger size than the weighing tank, to afford sufficient reserve supply to the pump while the upper tank is filling. If a single weighing tank is used it should preferably be of such capacity as to require emptying not oftener than every five minutes. If two or more tanks are used, the intervals between successive emptyings should not be less than three minutes. Measuring tanks calibrated by weighing may also be used.

In tests of complete steam power plants, where it is required to measure the feedwater without unnecessary change in the working conditions, a water meter may be employed. Meter measurement may also be required in many other cases, such as locomotive and marine service. The accuracy of meters should be determined by calibration in place under the conditions of use.

If a large quantity of water is to be measured, an automatic waterweigher, a rotary, disk, or Venturi meter, a weir, or some form of orifice measurement may be employed. In any case the measuring apparatus should be calibrated under the conditions of use, unless its design is such that standard formulæ and constants may be applied for determining the discharge. If recording mechanism is employed in connection with orifice or weir measuring apparatus, make sure that its record is reliable.

See Appendix No. 4 for methods of calibrating meters and Appendix No. 5 for orifice formula.

(2) *Water of Condensation Under Pressure.* In measuring jacket water or any supply under pressure which has a temperature exceeding 212 deg. fahr., the water should first be cooled, as may be done by discharging it into a tank of cold water previously weighed, or by passing it through a coil of pipe submerged in running and colder water, preventing thereby the loss of evaporation which occurs when such hot water is discharged into the open air. If such water is untrapped, the drain pipe should be provided with a gage glass and the outlet choked, so as to keep the water in sight in the glass.

(3) *Water Flowing in Large Pipes Under Pressure.* Venturi meters, Pitot tubes, pitometers and orifices may be used for measuring water

discharged by pumps through pipes under pressure. (See Appendix No. 5.)

(4) *Water Flowing in Streams and Canals.* Weirs, current meters, float rods, etc., may be employed for measuring the water used by waterwheels, and in some cases that discharged by pumping engines. (See Appendix No. 5.)

9c Steam Measuring Apparatus

Various forms of steam meters may be employed for measuring steam, provided such meters are properly calibrated under conditions of use, and the pulsations of pressure, if any, are not serious. For measuring the steam used by the auxiliaries of a steam plant, either individually or collectively, the orifice form of steam meter may be used, consisting of an orifice in a plate inserted between the two halves of a pair of flanges in the pipe through which the steam passes, or placed in a by-pass through which the steam is diverted, with gage pipe on either side for determining the fall of pressure. The quantity of steam represented by the various differences of pressure which occur, may be found by arranging the apparatus so as to draw steam through the orifice, and discharge it into a tank of water resting on platform scales, by which its actual weight in a given time is determined.

A plate ⅛ in. thick containing an orifice 1 in. diameter, with square edges, will discharge the approximate quantities of dry steam per hour given in Table 1, with various pressure drops, the pressure below the orifice being 100 lb. by gage.

TABLE 1 DISCHARGE THROUGH ORIFICE 1 IN. DIA. AT 100 LB. PRESSURE

Pressure Drop. Lb. Per Sq. In.	Lb. of Dry Steam Per Hour.
½	430
1	615
2	930
3	1200
4	1400
5	1560
10	2180
15	2640
20	3050

The water-glass method affords an approximate means for determining the steam consumption of auxiliaries, and for measuring the leakages of steam and water from the boiler and its connections. (See Appendix No. 3 for description of water-glass method.)

9d Gas and Air Measuring Apparatus

Apparatus to be used for measuring gas and air embraces gas meters, gasometers, Venturi meters,[1] orifices and Pitot tubes. The standard of reference for calibrating apparatus for this purpose is the gasometer. If a Pitot tube is used, and this instrument cannot be calibrated by actual measurement, the constants employed should be those obtained from a similar instrument which has been calibrated by actual reference to gasometer measurement under as nearly as possible the same conditions.

Directions for using the Pitot tube for gas and air measurement are given in Appendix No. 6. See also paper by W. C. Rowse, Trans. Am. Soc. M. E., Vol. 35, p. 633.

In this connection attention is directed to the electric method of gas measurement proposed by C. C. Thomas and described in his paper in Volume 31 of the Transactions,[2] also to orifice measurements described by R. J. Durley, in Volume 27.[3]

Air and other gases may also be measured by passing the air or gases over a steam coil and noting the rise in temperature due to the condensation of a certain amount of steam in a given time, making suitable correction for radiation. In the case of gases which are hot, the measurement can be made by cooling the gas, using a coil through which cold water is passed, care being taken that the temperature of the coil is above the dew-point of the air or gas.

9e Anemometers

For determining the approximate quantity of air discharged under light pressures by fans or blowers, or the amount of air used in the combustion of fuels, an anemometer of the fan-wheel type may be used, provided all the air can be passed through one or more openings of suitable size in which the anemometer may be placed. These openings should preferably be of such size (say at least 15 in. sq.) that the resistance interposed by the instrument (the fan being about 2½ in. in diameter) may be neglected. The instrument, when in use, can best be supported on the end of a small rod and moved slowly across the open end, also up and down, so as to obtain an average for the whole area. In the case of very large openings the area should be subdivided into a number of equal parts, the velocity determined for each subdivision, and the results averaged.

The anemometer is usually calibrated by mounting it on the end of a long and light rod which can be revolved around a central point and its reading compared with the velocity of the rod when moved in still air. Where suitable means can be provided for calibrating the anemometer under approximately its working conditions, such calibration is much to be preferred.

[1] See E. P. Coleman's paper on "The Flow of Fluids in a Venturi Tube," Trans. Am. Soc. M. E vol. 28, p. 483.

[2] C. C. Thomas, An Electric Gas Meter. Trans. Am. Soc. M. E., vol. 31, p. 655.

[3] R. J. Durley, On the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and under Small Differences of Pressure. Trans. Am. Soc. M. E., vol. 27, p. 193.

The quantity of air supplied for combustion in boiler and other furnaces may be determined from the analysis of the products of combustion, using the formula given in the Boiler Code, ¶ 58 (g).

9f Screens for Sizing Coal

The dimensions of screen openings to be used for sizing anthracite coals are given in Table 2, the sizes in each case being the opening through which the specified grade will pass, and that over which it will be carried without passing through. The openings referred to are circular.

TABLE 2 ANTHRACITE COAL SIZES

Name	Diameter of opening through or over which coal will pass, in.		Name	Diameter of opening through or over which coal will pass, in.	
	Through	Over		Through	Over
Broken..	$4\frac{1}{2}$	$3\frac{1}{4}$	No. 1 Buckwheat*	$\frac{9}{16}$	$\frac{5}{16}$
Egg.....	$3\frac{1}{4}$	$2\frac{5}{16}$	No. 2 Buckwheat*.	$\frac{5}{16}$	$\frac{3}{16}$
Stove....	$2\frac{5}{16}$	$1\frac{5}{8}$	No. 3 Buckwheat*.	$\frac{3}{16}$	$\frac{3}{32}$
Chestnut.	$1\frac{5}{8}$	$\frac{7}{8}$	Culm...........	$\frac{3}{32}$	..
Pea.....	$\frac{7}{8}$	$\frac{9}{16}$			

*The terms "Buckwheat," "Rice," and "Barley," respectively, are used in some localities instead of No. 1, No. 2, and No. 3 Buckwheat.

The sizes and grades of bituminous and semi-bituminous coals vary so much according to kind and locality that there are no standards of size for these coals which are generally recognized. (For bituminous coal sizes see Appendix No. 7.)

9g Pressure Gages

For determining pressure, the gages belonging to the plant may be used, provided they are compared with a standardized gage of the spring or mercury type and verified, due allowance being made for the head of water, if any, standing in the connecting pipe. Such comparisons should be made with both gages at their respective normal temperatures. In the use of spring gages for steam the gages should be protected by proper syphons or water seals and no leakage should be allowed at the gage-cock. The gages should also be located so that they will not be unduly heated.

For measuring low pressure or vacuum, a U-tube gage may be employed or a spring gage may be used, provided it is referred to a standard and corrected for water in the connecting pipe. In cases where extremely high vacuums are to be measured, as in turbine practice, the absolute-pressure gage is useful, provided the exhaustion is complete and no air is admitted afterwards.

For determining steam pressure on the two sides of an orifice, two gages should be used which are carefully graduated to single pounds, or, better, one gage should be used and this piped up so as to connect at will to either side of the orifice. A differential gage may also be employed, indicating at once the pressure drop. If the pressure drop is small, a glass U-tube containing mercury may be used. If the drop is less than one-half a pound, water columns may be substituted.

For determining the water pressure in the force main of a pumping engine, the gage should be one which is sensitive to changes amounting to ½ per cent. of the pressure indicated. If such a gage is not a part of the equipment of the plant, a special test gage should be attached.

Between the gage and the force main a small reservoir having an air chamber should be interposed, in the manner shown in Appendix No. 8, so as to prevent undue fluctuations of the gage and allow the gage cock to be run wide open. By means of a gage glass on the side of the chamber and an air valve, the average water level may be adjusted to the height of the center of the gage, and correction for this element of variation avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

For calibrating gages indicating pressures above the atmosphere, the dead-weight testing apparatus which is manufactured by many of the prominent gage makers may be employed as a standard of comparison. It consists of a vertical plunger nicely fitted to a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger (including weight of plunger) divided by the average area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever-varying zero point; that the mercury is pure; and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressures below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 in., with both arms properly filled with pure mercury.

The practice of choking the gage cock to reduce fluctuations of the gage pointer, if carried too far, is objectionable, especially when there is leakage around the plug. The pressure indicated under these circumstances is less than the true pressure. Before reading a gage of the spring type, the free working of the mechanism within should be assured by tapping the outside of the gage and thereby slightly moving the pointer.

9h Thermometers

Thermometers should be of the kind having graduations marked on the glass stem. Those used for temperatures above the boiling point of mercury (or say 500 deg. fahr.) should have nitrogen in the top of the bore. They should also have a small safety bulb at the top. Thermometers constructed in this way can be used satisfactorily up to 1000 deg. fahr.

Thermometers which are used for important data should be calibrated before and after a test, by reference to standard thermometers.

Standard thermometers are those which indicate 212 deg. fahr. in steam escaping from boiling water at the normal barometric pressure of 29.92 in. (referred to 32 deg.), the whole stem up to the 212 deg. point being surrounded by the steam; which indicate 32 deg. fahr. in melting ice, the stem being likewise completely immersed to the 32 deg. point; and which are calibrated for points between and beyond these two reference marks. For temperatures between 212 deg. and 400 deg. fahr., the comparison of the thermometer should be made with the temperature given in Marks and Davis Steam Tables, the method required being to place it in a thermometer-well surrounded by saturated steam under sufficient pressure to give the desired temperature. The pressure should be determined by a correct gage, and the thermometer should be immersed to the same extent as it is under its working condition.

A thermometer-well consists of a hollow plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far, if practicable, as the center. The inside diameter should be slightly larger than the outside diameter of the thermometer tube, and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 deg., and with soft solder for higher temperatures. For superheated steam the immersed portion should be fluted so as to increase the area of the absorbing surface.

When the stem is not immersed the correction to be added is 0 000088 n $(T-t)$, in which n is the number of degrees on the scale not immersed, T the indicated temperature, and t the mean temperature of the air surrounding the stem as shown by a thermometer suspended at the mean point.

For accurate work thermometers should be standardized for the immersion at which they are intended to be used, and such immersion should be recorded.

Thermometers are so readily broken that it is desirable in important tests to have a sufficient number on hand so that in case of accident the readings will not be interrupted. These spare thermometers should preferably be calibrated.

Thermometers may be calibrated, if desired, by direct comparison with Standard Thermometers certified by the U. S. Bureau of Standards.

For ordinary work thermometers may be used without correction if they are of the type that are graduated at a given immersion, the degree of immersion being marked on the stem and the temperature of the exposed stem being approximately that at which it was graduated.

9i Barometers

For important or extremely accurate steam tests and for gas engine tests the pressure of the atmosphere should be taken, either by a mercurial or aneroid barometer, and the reading from this instrument, reduced to pounds pressure per square inch, should be employed in determining the absolute steam pressure. In many cases it is sufficient to refer to the daily records of the nearest station of the Government Weather Bureau. These records, which refer to sea-level, should be corrected for altitude. Aneroid barometers may be readily calibrated by comparing them with a mercury barometer, making proper temperature corrections.

9 j Hygrometers

In tests where the hygrometric conditions of the atmosphere play an important part, it is necessary to use a hygrometer. The ordinary instrument consisting of a wet and dry bulb thermometer, preferably of the sling type, is suitable for the purpose. Hygrometric tables prepared for the Committee by Messrs. W. H. Carrier and J. A. Moyer, Members Am. Soc. M.E., are given in Appendix No. 9.

9 k Pyrometers

Metallic pyrometers used for determining high temperatures must be handled cautiously owing to the difficulty of exposing the whole of the stem to the current of gas the temperature of which is to be determined. Electric pyrometers either of the thermo-couple or resistance type are satisfactory for this work within their practical range, which is 1800 deg. fahr. for iron-nickel couples and 3000 deg. fahr. for platinum-iridium couples or platinum resistance pyrometers. Instruments of this kind can readily be calibrated by comparing them at low ranges of temperature with a standardized mercurial thermometer, both being placed for example in a current of hot air the temperature of which is under control. For extremely high temperatures such as that of a boiler furnace, optical, pneumatic, and radiation pyrometers may be used. The calibration of high-temperature instruments can best be undertaken in a laboratory especially fitted for the purpose. See Appendix No. 10.

9 l Draft Gages

The simplest form of draft gage is the ordinary U-tube. When the tube is kept clean and the two legs are close together with the scale extending at least to the center of each leg, it gives satisfactory indications. For measuring small amounts of draft some form of multiplying gage may be employed. One of the simplest multiplying instruments consists of a U-tube in which one leg is inclined from the horizontal and the amount of multiplication varies inversely as the sine of the angle of inclination, the tube being filled with a light mineral oil. Various satisfactory instruments having the multiplying feature are on the market. These can readily be calibrated by comparison with the simple U-tube gage when indicating a high draft, say one inch or more. It is preferable to use kerosene instead of water in the U-tube, and make allowance for the difference of specific gravity. Draft readings should be expressed in inches of water column.

9 m Steam Calorimeters

The most satisfactory instruments for determining the amount of moisture in steam are calorimeters that operate upon the throttling principle, or that combine the throttling and separating principles; the orifice used being of such size as to throttle to atmospheric pressure, and the instrument being provided with two thermometers, one showing the temperature above the orifice and the other that below it. If no commercial make of calorimeter is available on a test, an instrument of the throttling type can be made of pipe fittings as shown in Appendix No. 11. Instruments working on the separating principle alone may also be employed; also certain forms of electric calorimeters. See Trans. Am. Soc. M. E., Vol. 28, p. 616.

Directions for applying sampling nozzles are given in Part III, ¶ 28 and 29.

Further references to Steam Calorimeters are included in Appendix No. 11.

9n Coal Calorimeters

To determine the total heat of combustion of a sample of coal or other fuel, the best form of calorimeter to use is one in which the fuel is burned in an atmosphere of oxygen gas. The Mahler type of calorimeter is recognized as the most complete and accurate apparatus of this kind. Where the engineer conducting a test does not have this instrument, or some other reliable calorimeter at hand, the heat units can be determined by sending samples to a testing laboratory where such instruments are used.

For description of one form of the Mahler Calorimeter see Appendix No. 12.

9o Gas Calorimeters

The total heat of combustion of gas should be found by burning the gas in the Junker calorimeter, described in Appendix No. 13.

9p Coal Analysis Apparatus

The analyses commonly made are what are termed "proximate" analyses. For complete determinations of the quality of coal, it is necessary also to make the "ultimate" analysis. Approved methods of analysis are briefly described in Appendix No. 14.

9q Gas Analysis Apparatus

For determining the composition of flue gases in ordinary boiler work one of the simplest and most convenient instruments is the Orsat apparatus. This instrument can readily be used by the person conducting a test, or by some assistant whom he directs.

For determining the hydrogen and other unburned combustible matter in the flue gases, and for general gas analysis, the Hempel apparatus, or some modification thereof, is required. Work of this kind should be entrusted to a person who is familiar with all phases of the subject. These instruments are briefly described in Appendix No. 15.

Instruments known as CO_2 recorders are useful, if their accuracy is established.

9r Appliances and Methods Pertaining to Smoke Determination

No wholly satisfactory methods for either quantitative or qualitative smoke determinations have yet come into use, nor have any reliable methods been established for definitely fixing even the relative density of the smoke issuing from chimneys at different times. One method commonly employed, which answers the purpose fairly well, is that of making frequent visual observations of the chimney at intervals of one minute or less for a period of one hour and recording the observed characteristics according to the degree of blackness and density, and giving to the various degrees of smoke an arbitrary percentage value rated in some such manner as that expressed in Table 3.

TABLE 3 SMOKE PERCENTAGES

Dense black	100
Medium black	80
Dense gray	60
Medium gray	40
Light gray	20
Very light	5
Trace	1
Clear chimney	0

The color and density of smoke depend somewhat on the character of the sky or other background, and on the air and weather conditions obtaining when the observation is made, and these should be given due consideration in making comparisons. Observations of this kind are also subject to personal errors and errors of judgment. Nevertheless, these methods are useful, especially when the results are plotted, according to the percentage scale determined on, so that a graphic representation of the changes can be shown.

Various forms of charts and clouded glass arrangements for comparing and fixing smoke densities have been proposed, and to some extent used, but these have proved more or less unsatisfactory and they are subject to personal errors, and to sky, wind, and weather conditions, the same as the simpler method above described.

Among the chart methods referred to, the use of the Ringelmann smoke chart is perhaps the most familiar. This is shown in Appendix No. 16.

Another method of smoke determination consists in the use of a narrow flat metal plate suspended in the flue, the character of the smoke being indicated by the amount and quality of the soot and dust deposited upon the plate in a given time. This method, like others, is useful in furnishing a means of comparison in different cases rather than a means of exact determination. See Appendix No. 17 for further description.

Among the latest methods brought out for indicating and recording the density of smoke is one depending on the variations in the electrical conductivity of the metal selenium due to variations in the intensity of light shining upon it. Openings are provided on either side of the flue directly opposite each other. The selenium is located at one opening and a strong light at the other. The intensity of the light rays falling on the selenium varies with the density of the smoke. A milliampere meter in circuit with the selenium cell registers the variations.

9 s Indicators

To determine the amount of power developed in the cylinder of a reciprocating engine, or that expended in a pump or compressor cylinder, the instrument required is the steam engine indicator. One or more of these instruments is attached to the cylinder or cylinders, and operated from the cross-head or main shaft by the use of proper driving rig. As to the selection of the make of instrument, it should be one which is in all respects of first-class construction, and adapted to the purpose for which it is to be used.

Outside spring indicators are preferred for superheated steam, and in other cases where the temperature of the gas or vapor is very high or very low. For fuller particulars see Appendix No. 18.

9 t Planimeters

To determine the area of indicator diagrams from which to ascertain the mean effective pressure, it is convenient to use some form of planimeter. The simplest and probably the most desirable instrument is the Amsler polar planimeter, in which the area is registered in square inches.

It is desirable to calibrate a planimeter from time to time by running it over a figure having a known area, such as a right angle triangle of say 4 in. in length and 2 in. in height, observing whether it checks with the computed area.

9 u Tachometers and Other Speed Measuring Apparatus

For determining the speed of revolution of an engine shaft, especially where the speed exceeds 300 r.p.m., a convenient instrument is a tachometer which continuously indicates on a dial the number of turns per minute. This instrument can be arranged to have a permanent location and to be operated continuously when the engine is running, or it can be a portable instrument which is held in the hand and applied for the time being to the end of the shaft. These instruments are of four general classes, viz: fly-ball, liquid, electro-magnetic, and vibration.

These instruments should be calibrated by comparison with the record obtained by counting with the watch and a speed recorder or indicator, the number of turns per minute.

The determination of variation of speed during a single revolution, or the effect due to sudden changes of the load, is desirable, especially in engines driving electric generators used for lighting purposes. There is no recognized standard method of making such determinations, and if they are desired the method may be devised to suit the requirements.

One method for determining the instantaneous variation of speed which accompanies a change of load, is described as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of an electric current. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the fly-wheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots. When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel. When a change in the velocity of the fly-wheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the angular change of velocity during any given time is revealed.

9 v Friction Brakes or Absorption Dynamometers

The power delivered by an engine may be determined by the application of a Prony brake to the rim of the fly-wheel. The friction device may consist of a simple band or rope, a number of ropes, or a series of blocks, encircling the wheel. Weighing scales either of the platform or spring type are required for measuring the torque. For long runs the wheel is made with interior flanges for holding water to keep the rim cool.

The most satisfactory brake for absorbing and measuring power is some form of water friction brake. The advantage of a water brake is that it can be employed equally well for large or small amounts of power, and it is necessarily kept cool by the water upon which it depends for its operation. With this brake, the determination of the quantity of water used and the number of degrees its temperature is raised (when corrected for radiation) furnishes a means of computing the amount of heat converted into work, and thereby obtaining an additional measurement of the amount of power developed.

Another satisfactory form of brake is the electric dynamometer, in which the work is transformed into electric energy, and the torque is measured in the same manner as in a Prony brake.

Several brakes are described in Appendix No. 19.

9 w Transmission Dynamometers

Transmission dynamometers furnish means for determining the amount of power delivered by an engine under working conditions. In the case of a mill engine it is the power transmitted from the main shaft of the engine to the driving shaft of the mill. If this power is carried through a belt, the dynamometer measures the net amount of force transmitted.

In a marine engine driving a screw propeller through a long shaft, the dynamometer shows the torsional strain on the shaft at a point as near as practicable to the engine. In a locomotive the dynamometer measures the amount of pull on the draw-bar through which the power is transmitted to the first car of the train.

See Appendix No. 20 for details regarding dynamometers.

9x Electrical Instruments

The output of an engine driving an electric generator should be determined by the use of a set of electrical instruments supplementary to the switchboard equipment, and connected to the generator so as to be thrown in and out of circuit by means of switches. These instruments should include the shunts, transformers, or multipliers, which belong to them, and should be such as have been verified by comparison with known standards based on those of the U. S. Bureau of standards at Washington. The instruments selected should be as nearly "dead-beat" as practicable, that is, the pointer should come to rest at once, after changes of load.

The instruments required for a direct-current generator, where the load is substantially constant, embrace a single ammeter and a single voltmeter. Where the load is rapidly fluctuating, a watt-hour meter is also needed.

Those required for a single-phase alternating-current generator carrying a substantial constant load embrace one ammeter, one voltmeter, and one single-phase wattmeter. For a rapidly fluctuating load a watt-hour meter should also be used. For a two, or three-phase alternating-current generator, with substantially constant load, one ammeter and one voltmeter, each arranged for ready connection at will to any phase, and two single-phase wattmeters, are required. Where the load is rapidly fluctuating a polyphase watt-hour meter is also needed.

In the case of an alternating-current generator excited from an independent source of power or by a motor generator, additional instruments are necessary for measuring the power used for excitation.

The instruments should be so located with reference to the generator, switchboard, and neighboring conductors, as to avoid the disturbing effects due to stray fields. If the switchboard equipment is complicated, the wiring should be connected under the supervision of one who is familiar with all the details of the switchboard, its connections, and the surrounding cables.

Watt-hour meters should be verified in place and under working conditions. Directions for calibrating these meters and further details regarding the application and use of electrical instruments, determination of power factor, etc., are given in Appendix No. 21.

9y Water Rheostats

For tests of engines driving electric generators under certain fixed loads it is often necessary to regulate the output of current by the use of a water rheostat. One of the simplest and most effective forms of this apparatus for direct-current work consists of a coil of wire wound spirally on a wooden reel immersed in a stream of running water, or in a tank or some natural body of water where circulation can be provided for, arranged with a system of switches so that more or less of this resistance can be turned on as may be required. A No. 12 B. G. iron wire 2000 ft. long, thus used, will handle a current of 1600 amperes at 250 volts.

Other rheostats are described in Appendix No. 22.

9z Steam Tables

Quantities depending upon the properties of saturated and superheated steam, which are used throughout the Codes, such as B.t.u. per pound of steam, temperatures corresponding to various pressures, etc., are based on Marks and Davis tables (edition of 1909). The report of a test should state the authorship of the tables on which the calculations are based.

MISCELLANEOUS INSTRUCTIONS

10 The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test, and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

11 Before a test is undertaken, it is important that the boiler, engine, or other apparatus concerned shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance.

It would, for example, be manifestly improper to start a test for determining the maximum efficiency of an externally fired boiler with brick setting, until the boiler had been at work a sufficient number of days to dry out thoroughly and heat the brick work to its working temperature; and likewise improper to begin an engine test for determining the performance under certain prearranged conditions until those conditions had become established by a suitable preliminary run.

12 An exception should be noted where the object of the test is to obtain the working performance, including the effect of preliminary heating, in which case all the conditions should conform to those of regular service.

13 In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

OPERATING CONDITIONS

14 In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feedwater, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice, and when this is the object the usual working conditions should prevail throughout the trial.

RECORDS

15 A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hourly periods, or, if necessary, periods of less duration, and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

16 The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, and gas), should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations. When it is essential that a number of instruments be read simultaneously, there should be an observer stationed at each one and the readings should be taken on signal from a time-keeper.

17 Make a memorandum of every event connected with the progress of a test, however unnecessary at the time it may appear. A record should be made of the exact time of every such occurrence and the time of taking every weight and every observation. For the purpose of identification the signature of the observer and the date should be affixed to each log sheet or record.

18 In the simple matter of weighing coal by the barrow-load, or weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each load is weighed or emptied should be recorded. The weighing of coal should not be delegated to unreliable assistants, and whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feedwater.

PLOTTING DATA AND RESULTS

19 If it is desired to show the uniformity of the data at a glance the whole log of the trial should be plotted on a chart, preferably while the test is in progress, using horizontal distances to represent times of observation, and vertical distances on suitable scales to represent various data as recorded. A chart showing the log of a boiler test is illustrated in Appendix No. 23.

REPORT

20 The report of a test should present all the leading facts bearing on the design, dimensions, condition, and operation of the apparatus tested, and should include a description of any other apparatus and auxiliaries concerned, together with such sketches and photographs as may be needed for a clear understanding of all points under consideration. It should state clearly the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, closing with a tabular summary of the principal data and results.

PART II

STANDARDS RELATING TO CAPACITY, EFFICIENCY AND ECONOMY

21 The standard units on which to base the various measures of capacity, and the standard forms of expressing efficiency and economy to which the Codes apply, are assembled in Tables 4, 5 and 6.

TABLE 4 DEFINITION OF UNITS

One British thermal unit, or heat unit as herein used = 1/180 of the heat required to raise 1 lb. of water from 32 deg. to 212 deg. fahr.

One unit of evaporation (U. E.) = heat required to evaporate 1 lb. of water at 212 deg. into steam at the same temperature = 970.4 British thermal units.

One foot-pound = work done by 1 lb. force acting through 1 ft.

One pound (of force) = the force exerted by gravity on 1 lb. of matter where the acceleration due to gravity is 32.1740 ft. per second per second; that is, (very nearly) the force of gravity on 1 lb. of matter at latitude 45 deg. at the sea level.

Mechanical equivalent of heat: 1 B.t.u. = 777.54 ft-lb. or 1 ft-lb. = 0.0012861 B.t.u.*

One horsepower = 33,000 ft-lb. per min. = 550 ft-lb. per sec.
= 1,980,000 ft-lb. per hour.
= 2,546.5 B.t.u. per hour = 42.44 B.t.u. per min.
= 745.7 watts = 0.7457 kilowatts.*

One kilowatt = 1000 watts = 1.3410 h.p. = 3,415 B.t.u. per hour = 737.56 ft-lb. per sec.

One kilowatt-hour = 1.3410 h.p-hr. = 2,655,200 ft-lb.

One U. S. gallon = 231 cu. in.

One atmosphere = 760 m.m. or 29.921 in. of mercury at 32 deg. fahr.
= 29.951 in. of mercury at 62 deg.** fahr.
= 14.6963 lb. per sq. in.

* Based on the following accepted values:—
1 mean calorie = 4.1834 × 107 dyne-centimeters (Marks & Davis)
1 dyne = $\frac{1 \text{ gram}}{980.665}$ (International Standard). 1 c.m. = $\frac{0.3937}{12}$ ft.
1 B. t. u. = 1.8 lb-deg. cent. 1 gram = .002204622 lb.
1 watt = 10.7 dyne-centimeters per sec. = 0.73756046 ft. lb. per sec.
1 h.p. = 550 ft. lb. per sec. = $\frac{550}{0.73756046}$ = 745.702 watts.

** At latitude 45 deg. at sea level the density of mercury at 32 deg. is 13.59545 grams per cu. cm. Linear expansion of mercury 0.0000333 per deg. fahr.

TABLE 5 STANDARD UNITS OF CAPACITY

a Boilers*	One pound of water evaporated into dry steam from and at 212 deg. per hour
b Reciprocating Steam Engines	One indicated horsepower developed in the main cylinders One brake horsepower delivered by the main shaft
c Steam Turbines	One brake horsepower delivered by the main shaft
d Turbo-generators, (including engine-driven generators)	One kilowatt delivered at the generator terminals,† not including kilowatts used by exciter‡
e Pumping Machinery	One gallon of water discharged to the force main in 24 hr. One gallon of water discharged per min.§ One water horsepower delivered to the force main, based on the total head including suction
f Compressors, Blowers, and Fans	One cu. ft. of air at 62 deg. and 30 in.¶ One air horsepower
g Locomotives	One indicated horsepower developed in the main cylinders One dynamometer horsepower delivered to the draw-bar
h Gas Producers	One pound of dry fuel of given quality consumed per hour. One cu. ft. per hour of dry gas having a stated quality at 60 deg. and 30 in.
i Gas and Oil Engines	One brake horsepower delivered by the main shaft One indicated horsepower developed in the engine cylinder
j Waterwheels	One brake horsepower delivered by the main shaft One kilowatt delivered at the generator terminals,† not including kilowatts used by exciter‡

* A subsidiary unit which may be used for stationary boilers is a "Boiler Horsepower," or 34½ lb. of water evaporated from and at 212 deg. per hour, i.e., from water at 212 deg. into steam at the same temperature. The unit called "Myriawatt" has been suggested by some engineers as a unit of boiler capacity. It is 2 per cent greater than the "boiler horsepower" and is equivalent to 34,150 heat units per hour, the "boiler horsepower" being 33,479 heat units per hour.

† If switchboard instruments are used for the electrical measurements, correction should be made for the drop in voltage between generator and switchboard, unless the drop is so small as to be negligible.

‡ If the exciter current is taken from an outside source the kw. thus supplied including field rheostat losses are to be deducted from the total output. Likewise the kw. used by separately driven ventilating fan.

§ This unit applies to small pumps and some classes of large sized pumps.

¶ 30 in. mercury barometer refers in round numbers to a standard atmosphere at 62 deg. In exact figures the standard atmosphere is 29.951 in. of mercury at 62 deg.

22 Contracts for power plant apparatus should specify the leading dimensions of the apparatus and its rated capacity, expressed in the units given in Table 5. If a specific guarantee of capacity is made, either working capacity or maximum capacity, the operating conditions under which the guarantee is to be met should be clearly set forth; such, for example, as steam pressure, speed, vacuum, quality of fuel, force of draft, etc. Likewise if a contract contains a guarantee of economy all the conditions should be fully specified.

23 The commercial rating of capacity determined on for power plant apparatus, whether for the purpose of contracts for sale, or otherwise, should be such that a sufficient reserve capacity beyond the rating is available to meet the contingencies of practical operation; such contingencies, for example, as the loss of steam pressure and capacity due to cleaning fires, inferior coal, oversight of the attendants, sudden demand for an unusual output of steam or power, etc.

TABLE 6 STANDARDS OF EFFICIENCY AND ECONOMY

a Boilers	Relation between B.t.u. absorbed by boiler per lb. of dry coal and calorific value of 1 lb. dry coal. (Efficiency of boiler furnace and grate.)
	Relation between B.t.u. absorbed by boiler, per lb. of combustible burned and calorific value of 1 lb. combustible. (Efficiency based on combustible.)
b Reciprocating Steam Engines	(1) B.t.u. per i.h.p-hr.
	(2) B.t.u. per brake h.p-hr.
	(3) Ft-lb. of net work per B.t.u.
	(4) Thermal efficiency referred to i.h.p.
	(5) Thermal efficiency referred to br. h.p.
	(6) Rankine cycle ratio referred to i.h.p.
	(7) Rankine cycle ratio referred to br. h.p.
	(8) Lb. of steam per i.h.p-hr.
	(9) Lb. of steam per br. h.p-hr.
c Steam Turbines	(1) B.t.u. per br. h.p-hr.
	(2) Ft-lb. of net work per B.t.u.
	(3) Thermal efficiency.
	(4) Rankine cycle ratio.
	(5) Lb. of steam per br. h.p-hr.

NOTE:— The term "steam" in Table 6 means dry steam, either saturated or superheated as the case may be. If it contains moisture, the moisture is to be deducted. If superheated, no correction is to be made.

d Turbo-generators, (including engine-driven generators)	(1) B.t.u. per kw-hr. (2) Ft-lb. of net work per B.t.u. (3) Thermal efficiency. (4) Rankine cycle ratio. (5) Lb. of steam per kw-hr.
e Pumping Engines.	(1) Ft-lb. of work per million B.t.u. (2) Ft-lb. of net work per B.t.u.
f Compressors, Blowers, and Fans ...	(1) B.t.u. per net air h.p-hr. (2) Ft-lb. of net work per B.t.u. (3) Lb. of steam per net air h.p-hr. (4) Lb. of steam per 1000 cu. ft. of free air compressed to 100 lb. gage pressure reduced to atmospheric temperature.

g Complete Steam Power Plants:

Plants in General.	(1) Lb. of coal as fired per i.h.p-hr. (2) Lb. of steam per i.h.p-hr. (3) Heat units in fuel consumed per i.h.p-hr.
Electric Plants ..	(1) Lb. of coal as fired per kw-hr. (2) Lb. of steam per kw-hr. (3) Heat units in fuel consumed per kw-hr.
Pumping Plants .	(1) Ft-lb. of work per million B.t.u. (2) Lb. of coal as fired per water h.p-hr.
Air Machinery Plants	(1) Lb. of coal as fired per air h.p-hr. (2) Lb. of steam per air h.p-hr.
h Locomotives	(1) Lb. of coal as fired per i.h.p-hr. (2) Lb. of coal as fired per dyn. h.p-hr. (3) Lb. of steam per i.h.p-hr. (4) Lb. of steam per dyn. h.p-hr. (5) Lb. of coal as fired per ton-mile.
i Gas Producers	Relation between B.t.u. of the gas output per lb. of dry fuel and calorific value of 1 lb. of dry fuel.

NOTE:—The i.h.p. and brake h.p. in this table refer to that of the main engine, turbine, or waterwheel, and the kw. to the power measured at the generator terminals, not including exciter output.

j Gas and Oil Engines
- (1) B.t.u. per i.h.p-hr.
- (2) B.t.u. per br. h.p-hr.
- (3) Ft-lb. of net work per B.t.u.
- (4) Thermal efficiency referred to i.h.p.
- (5) Thermal efficiency referrèd to br. h.p.
- (6) Lb. of oil per i.h.p-hr.
- (7) Lb. of oil per br. h.p-hr.
- (8) Cu. ft. of dry gas at 60 deg. and 30 in. per i.h.p-hr.
- (9) Cu. ft. of dry gas at 60 deg. and 30 in. per br. h.p-hr.

k Waterwheels .....
- (1) Relation between brake h.p. and potential h.p. of total water used.
- (2) Relation between kw-hr. delivered and potential kw-hr.

The English Standards used in Tables 4 to 6 may be converted into metric units (and vice versa) by the use of the following factors:

Factors for Converting English Units to Metric Units.

One inch	= 2.54	centimeters
One foot	= 0.3048	meter
One sq. in.	= 6.4516	sq. c.m.
One cu. ft.	= 0.028317	cubic meter
One U. S. gallon	= 3.7854	liters
One pound	= 0.453592	kilogram
One lb. per sq. in	= 0.070307	kg. per sq. c.m.
One foot-pound	= 0.13826	meter-kilogram
One horsepower	= 1.0139	cheval-vapeur
One B.t.u.	= 0.252	kilogram-calorie
One deg. Fahr.	= 0.55556	deg. centigrade

Factors for Converting Metric Units to English Units.

One centimeter	= 0.3937	inch
One meter	= 3.28083	foot
One sq. c.m.	= 0.155	sq. in.
One cubic meter	=35.3145	cu. ft.
One liter	= 0.26417	U. S. gallon
One kilogram	= 2.20462	pounds
One kg. per sq. c.m.	=14.223	lb. per sq. in.
One meter-kilogram	= 7.233	foot-pound
One cheval-vapeur	= 0.98629	horsepower
One kilogram-calorie	= 3.9683	B.t.u.
One deg. centigrade	= 1.8	deg. Fahr.

The "calorific value" of fuel is the number of heat units developed in completely burning one pound of the fuel including the heat contained in any water vapor formed through burning the hydrogen component. This is the higher heat value and not the so-called net or available heat value, in obtaining which the latent heat in the vapor thus formed is deducted.

PART III

RULES FOR SAMPLING AND DRYING COAL AND ASH AND SAMPLING STEAM AND GASES

A Sampling and Drying Coal

24 Select a representative shovelful from each load[1] as it is drawn from the coal pile or other source of supply, and store the samples in a cool place in a covered metal receptacle. When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it by the process of repeated quartering and crushing to a sample weighing about 5 lb., the largest pieces being about the size of a pea. From this sample two 1-qt. air-tight glass fruit jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value, and chemical composition. These operations should be conducted where the air is cool and free from drafts.

25 When the sample lot of coal has been reduced by quartering to say 100 lb., a portion weighing say 15 to 20 lb. should be withdrawn for the purpose of immediate moisture determination. This is placed in a shallow iron pan and dried on the hot iron boiler flue for at least 12 hours, being weighed before and after drying on scales reading to quarter ounces.

26 The moisture thus determined is approximately reliable for anthracite and semi-bituminous coals, but not for coals containing much inherent moisture. For such coals, and for all reliable determinations the following method should be pursued:

Take one of the samples contained in the glass jars, and subject it to a thorough air drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill or other suitable crusher adjusted so as to produce somewhat coarse grains (less than 1-16 in.), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams,[2] weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it for one hour in an air or sand bath at a temperature between 240 and 280 deg. fahr., (this temperature being necessary with coal which is not

[1] Or say one small shovelful for each 500 lb.

[2] Say ½ oz. to 2 oz.

powdered). Weigh it and record the loss, then heat and weigh again until the minimum weight has been reached. The difference between the original and the minimum weight is the moisture in the air-dried coal. The sum of the moisture thus found and that of the surface moisture is the total moisture.

If a large drying oven is available, the moisture may be determined by heating one of the glass jars full of coal (the cover being removed) at a temperature between 240 and 280 deg. fahr. until it reaches a minimum weight.

With certain lignites lower temperatures for drying may be advisable. (See Appendix No. 38.)

B Sampling Ashes and Refuse

27 The general method above described may also be followed for obtaining a sample of the ashes and refuse, and for determining the amount of moisture, if any, in the sample.

C Sampling Steam

28 Construct a sampling pipe or nozzle made of ½-in. iron pipe and insert it in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed. The inner end of the pipe, which should extend nearly across to the opposite side of the main, should be closed and the interior portion perforated with not less than twenty ⅛-in. holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than half an inch from the wall of the pipe.

The sampling pipe should not be placed near a point where water may pocket or where such water may affect the amount of moisture contained in the sample. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam owing to a lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With goose-neck connections a small amount of water may collect on the bottom of the pipe near the upper end where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the goose-neck and increases the moisture at that point, making it an undesirable location for sampling. In any case it should be borne in mind that with low velocities the tendency is for drops of entrained water to settle to the

bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met.*

29 If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting, and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calorimeter; or better, a steam separator should be installed at the point noted.

30 In testing a stationary boiler the sampling pipe should be located as near as practicable to the boiler, and the same is true as regards the thermometer-well when the steam is superheated. In an engine or turbine test these locations should be as near as practicable to the throttle valve. In the test of a plant where it is desired to get complete information, especially where the steam main is unusually long, sampling nozzles or thermometer-wells should be provided at both points, so as to obtain data at either point as may be required.

31 In a locomotive, the calorimeter should be attached either to the steam dome where it may be connected to the throttle opening, or to the steam passage in the saddle casting on one side.

D Sampling Gases

32 Producer Gas. A satisfactory sample of the gas flowing through the main may be obtained by tapping into the main a ¼-in. pipe, extending it to the center, the end of the pipe being left open. The point of attachment should be near the producer or near the scrubber according as the gas is used for heating purposes or for power, or if desired samples may be obtained at both points. The thermometer showing the temperature of the gas should be located near the sampling pipe.

33 Flue Gases. The sample for flue gas analysis should be drawn from the region near the center of the main body of escaping gases, using a sampling pipe not larger than ¼-in. gas pipe. The point selected should be one where there is no chance for air-leakage into the flue which could affect the average quality. In a round or square flue having an area of not more than one-eighth of the grate surface, the sampling pipe may be introduced horizontally at a centre line, or preferably a little higher than this line, and the pipe should con-

* With reference to sampling exhaust steam see paper by H. G. Stott and R. J. S. Pigott, Trans. Am. Soc. M. E., Vol. 32, pp. 75-77.

tain perforations extending the whole length of the part immersed, pointing toward the current of gas, the collective area of the perforations being less than the area of the pipe. The pipe should be frequently removed and cleaned.

It is advisable to take samples both from the flue and from the furnace, so as to determine the amount of air leakage through the setting and the changes in the composition of the gas between the furnace and the flue. It is best to draw a continuous sample, using a suitable aspirator, and provide a branch pipe from which to obtain the test-sample. The test-sample can then be taken either momentarily or continuously, according to the requirements.

SECOND SECTION: INDIVIDUAL CODES

PART IV

RULES FOR CONDUCTING EVAPORATIVE TESTS OF BOILERS

OBJECT AND PREPARATIONS

34 Determine the object of the test, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given in ¶ 1 to 33, and make preparations for the test accordingly.

FUEL

35 Determine the character of fuel to be used.[1] For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made.

In the Eastern States the standards thus regarded for semi-bituminous coals are Pocahontas (Va. and W. Va.) and New River (W. Va.); for anthracite coals those of the No. 1 buckwheat size, fresh-mined, containing not over 13 per cent ash by analysis; and for bituminous coals, Youghiogheny and Pittsburgh coals. In some sections east of the Allegheny Mountains the semi-bituminous Clearfield (Pa.) and Cumberland (Md.) are also considered as standards. These coals when of good quality possess the essentials of excellence, adaptability to various kinds of furnaces, grates, boilers, and methods of firing required, besides being widely distributed and generally accessible in the Eastern market.

There are no special grades of coal mined in the Western States which are widely and generally considered as standards for testing purposes, the best coal obtainable in any particular locality being regarded as the standard of comparison.

36 A coal selected for maximum efficiency and capacity tests should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

37 For guarantee and other tests with a specified coal containing not more than a certain amount of ash and moisture, the coal selected

[1] This code relates primarily to tests made with coal. For reference to oil and gas fuel tests see ¶61.

should not be higher in ash and in moisture than the stated amounts, because any increase is liable to reduce the efficiency and capacity more than the equivalent proportion of such increase.

38 In cases of guarantee tests with fuel of a specified calorific value, there should be a clear understanding as to the permissible variation from the given value.

39 The size of the coal, especially where it is of the anthracite class, should be determined by screening a suitable sample. (See ¶ 9f and Appendix No. 7 for list of sizes.)

APPARATUS AND INSTRUMENTS

40 The apparatus and instruments required for boiler tests are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Graduated scales attached to the water glasses.
(*c*) Tanks and platform scales for weighing water (or water meters calibrated in place).
(*d*) Pressure gages, thermometers, and draft gages.
(*e*) Calorimeters for determining the calorific value of fuel and the quality of steam.
(*f*) Pyrometers.
(*g*) Gas analyzing apparatus.

41 Full directions regarding the use and calibration of the abovementioned appliances are given under the heading Apparatus and Instruments, ¶ 7 to 9, and in ¶ 24 to 33.

42 For particulars regarding the best location of the various instruments and apparatus, see Appendix No. 24.

OPERATING CONDITIONS

43 Determine what the operating conditions and method of firing should be to conform to the object in view, as pointed out in ¶ 14, and see that they prevail throughout the trial, as nearly as possible.

Where uniformity in the rate of evaporation is required, arrangement can usually be made to dispose of the steam so that this result can be attained. In a single boiler it may be accomplished by discharging steam through a waste pipe and regulating the amount by means of a valve. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers to meet the varying demands

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Where uniformity in the rate of evaporation is required, arrangement can usually be made to dispose of the steam so that this result can be attained. In a single boiler it may be accomplished by discharging steam through a waste pipe and regulating the amount by means of a valve. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers to meet the varying demands for steam, leaving the test boiler to work under a steady rate of evaporation.

DURATION

44 The duration of tests to determine the efficiency of a hand-fired boiler should be at least 10 consecutive hours. In case the rate of combustion is less than 25 lb. per square foot of grate per hour, the tests should be continued for such a time as may be required to burn a total of at least 250 lb. of coal per square foot of grate. Tests of longer duration than 10 hours are advisable in order to obtain greater accuracy.

45 In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least 24 hours. If the stoker is of a type that permits the quantity and condition of the fuel bed at beginning and end of the test to be accurately estimated, the duration may be reduced to 10 hours, or such time as may be required to burn the above noted total of 250 lb. per sq. ft.

In commercial tests where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of the test, whether the boilers are hand-fired or stoker-fired, should be at least 24 hours. Likewise in commercial tests, either of a single boiler or of a plant of several boilers, which operate regularly a certain number of hours and during the balance of the day the fires are banked, the duration should not be less than 24 hours.

The duration of tests to determine the maximum evaporative capacity of a boiler, without determining the efficiency, should not be less than three hours.

STARTING AND STOPPING

46 The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, should be as nearly as possible the same at the end as at the beginning of the test.

47 To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run, burn the fire low, and thoroughly clean it, leaving enough live coal spread evenly over the grate (say from 2 to 4 in.)[1], to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured; also the water level[2], the steam pressure, and the time, and record the latter as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level[2], the steam pressure, and the time, and record the latter as the stopping time. If the water level is lower than at the beginning, a correction should be made by computation, rather than by feeding additional water. Finally remove the ashes and refuse from the ashpit.

In a plant containing several boilers where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition. After the last fire is cleaned and in working condition, burn all the fires low (say 4 to 6 in.), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various final observations made as noted.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the above directions pertaining to starting and stopping in a plant of several boilers may be followed.

48 To obtain the desired equality of conditions of the fire when a mechanical stoker other than a chain grate is used, the procedure should be modified where practicable as follows:

Regulate the coal feed so as to burn the fire to the low condition required for cleaning. Shut off the coal-feeding mechanism and fill the hoppers level full. Clean the ash or dump plate, note quickly the depth and condition of the coal on the grate, the water level,[2] the steam pressure, and the time, and record the latter as the starting time. Then start the coal-feeding mechanism, clean the ashpit, and proceed with the regular work of the test.

When the time arrives for the close of the test, shut off the coal-feeding mechanism, fill the hoppers and burn the fire to the same low

[1] 1 to 2 in. for small anthracite coals.

[2] Do not blow down the water-glass column for at least one hour before these readings are taken. An erroneous indication may otherwise be caused by a change of temperature and density of the water within the column and connecting pipe.

point as at the beginning. When this condition is reached, note the water level, the steam pressure, and the time, and record the latter as the stopping time. Finally clean the ash plate and haul the ashes.

In the case of chain grate stokers, the desired operating conditions should be maintained for half an hour before starting a test and for a like period before its close, the height of stoker gate or throat plate and the speed of the grate being the same during both of these periods.

RECORDS

49 The records of data should be obtained as pointed out in Part I, ¶ 15 to 18. Half-hourly readings of the instruments are usually sufficient. If there are sudden and wide fluctuations, the readings in such cases should be taken every fifteen minutes, and in some instances oftener.

In hand-fired tests the coal should be weighed and delivered to the firemen in portions sufficient for one hour's run, thereby ascertaining the degree of uniformity of firing. An ample supply of coal should be maintained at all times, but the quantity on the floor at the end of each hour should be as small as practicable, so that the same may be readily estimated and deducted from the total weight. Likewise in stoker tests the weight of coal fed each hour to the furnace should be obtained.

The records should be such as to ascertain also the consumption of feedwater each hour, and thereby determine the degree of uniformity of evaporation.

QUALITY OF STEAM

50 If the boiler does not produce superheated steam the percentage of moisture in the steam should be determined by the use of a throttling or separating calorimeter, in the manner pointed out in Part III, ¶ 28-29. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer-well, as pointed out in ¶ 9h, Part I.

SAMPLING AND DRYING COAL

51 During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, ¶ 24 to 26.

ASHES AND REFUSE

52 The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed so far as possible in a dry state. If wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis, for the determination of unburned carbon, and for fusing tests.*

CALORIFIC TESTS AND ANALYSES OF COAL

53 The quality of the fuel should be determined by calorific tests and analyses of the coal sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, ¶ 9n and ¶ 9p and in Appendices Nos. 12 and 14.

ANALYSES OF FLUE GASES

54 For approximate determinations of the composition of the flue gases, the Orsat apparatus, or some modification thereof, should be employed. If momentary samples are obtained the analyses should be made as frequently as possible, say every 15 to 30 minutes, depending on the skill of the operator, noting the furnace and firing conditions at the time the sample is drawn. If the sample drawn is a continuous one, the intervals may be made longer.

55 Fuller directions will be found in Part I, under the heading Gas Analysis Apparatus, ¶ 9q, and in Appendix No. 15.

SMOKE OBSERVATIONS

56 In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes (or if necessary every minute),** noting at the same time the furnace and firing conditions.

57 Full particulars regarding these observations are given in Part I, under the heading Appliances and Methods Pertaining to Smoke Determination, ¶ 9r, and in Appendices Nos. 16 and 17.

* See paper by F. C. Hubley on Bituminous Coals; Predetermination of Their Clinkering Action (Proc. The Engineers' Club of Phila., Jan. 1915) and by Lionel S. Marks on the Clinkering of Coal (Trans. Am. Soc. M. E., vol. 36, p. 801).

** For observations covering a period of one or more single firings, the intervals should be quarter-minutes.

CALCULATION OF RESULTS

58a Corrections for Quality of Steam

When the percentage of moisture is less than 2 per cent it is sufficient merely to deduct the percentage from the weight of water fed, in which case the factor of correction for quality is

$$1 - \frac{\text{per cent moisture}}{100}$$

When the percentage is greater than 2 per cent or if extreme accuracy is required, the factor of correction is

$$1—P\,\frac{H—h_1}{H—h}$$

in which P is the proportion of moisture, H the total heat of 1 lb. of saturated steam, h_1 the heat in water at the temperature of saturated steam, and h the heat in water at the feed temperature.

When the steam is superheated the factor of correction for quality of steam is $\frac{H_s—h}{H—h}$ in which H_s is the total heat of 1 lb. of superheated steam of the observed temperature and pressure.

Unless otherwise provided, a combined boiler and superheater should be treated as one unit, and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler.

58b Correction for Steam or Power Used for Aiding Combustion

The quantity of steam or power, if any, used for producing draft, injecting fuel, or aiding combustion, should be determined and recorded in the Table of Data and Results. There should also be recorded, by foot-note below the table, a statement showing whether or not a deduction has been made from the total evaporation for steam or power so used, and if such deduction has been made, the method of computing it.

58c Equivalent Evaporation

The "equivalent evaporation from and at 212 deg." is obtained by multiplying the weight of water evaporated, corrected for moisture in steam, by the "factor of evaporation." The latter equals.

$$\frac{H—h}{970.4}$$

in which H and h are respectively the total heat of saturated steam and of the feedwater entering the boiler.

The "factor of evaporation" and the "factor of correction for quality of steam" may be combined into one expression in the case of superheated steam as follows:

$$\frac{H_s - h}{970.4}$$

58d Efficiency

The "efficiency of boiler, furnace and grate" is the relation between the heat absorbed per pound of dry coal, and the calorific value of one pound of dry coal, or the relation between the two based on coal as fired.

The "efficiency based on combustible" is the relation between the heat absorbed per pound of combustible burned, and the calorific value of one pound of combustible. This expression of efficiency furnishes an approximate means for comparing the results of different tests, when the losses of unburned coal due to grates, cleaning, etc., are eliminated. (For "furnace efficiency" see Appendix No. 32).

The "combustible burned" is determined by subtracting from the weight of coal supplied to the boiler, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ashpit, and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues, and combustion chambers, including ash carried away in the gases, if any, determined from the analyses of coal and ash.* The "combustible" used for determining the calorific value is the weight of coal less the moisture and ash found by analysis.

The "heat absorbed" per pound of coal, or combustible, is calculated by multiplying the equivalent evaporation from and at 212 deg. per lb. of coal or combustible by 970.4.

58e Heat Balance

A "heat balance," or approximate distribution of the calorific value of one lb. of dry coal among the several items of heat utilized and heat lost, should be obtained in cases where the flue gases have been analyzed and a complete analysis made of the coal.

The loss due to moisture in the coal is found by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of moisture referred to dry coal.

The loss due to moisture formed by the burning of hydrogen is obtained by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of the hydrogen determined from the analysis of the coal referred to dry coal, and multiplying the result by 9.

The loss due to heat carried away in the dry gases is found by multiplying the weight of gas per pound of dry coal by the elevation of temperature of the gases above the temperature of the boiler room, and by the specific heat of the gases (0.24). The weight of gas per pound

* In cases of high rates of combustion the determination of the combustible burned may be subject to considerable error on account of the loss of cinder, soot, and unburned fuel which are blown to waste.

of dry coal is obtained by finding the weight of dry gas per pound of carbon burned, using the formula

$$\frac{11\ CO_2+8\ O+7\ (CO+N)}{3\ (CO_2+CO)}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, and multiplying this result by the proportion of the carbon burned to the whole amount of coal as determined from the results of the analysis of the coal, ash, and refuse.

The loss due to incomplete combustion of carbon is found by first obtaining the proportion of the carbon monoxide in the gases to the sum of the carbon monoxide and carbon dioxide, then multiplying this proportion by the proportion of carbon in the coal minus the carbon lost in the ash and refuse, referred to the total carbon in the coal, and finally multiplying the product by 10,150, which is the number of heat units generated by burning to carbon dioxide one pound of carbon contained in carbon monoxide.

The loss due to combustible matter in the ash and refuse is found by multiplying the proportion that this combustible bears to the whole amount of dry coal, by its calorific value per pound. For most purposes it is sufficient to assume the latter to be 14,600 B.t.u., the same as that of carbon.

The loss due to moisture in the air is determined by multiplying the weight of such moisture per pound of dry coal by the elevation of temperature of the flue gases above the temperature of the boiler room and by 0.47. The weight of moisture is found by multiplying the weight of air per pound of dry coal by the moisture in one pound of air determined from readings of the wet and dry-bulb thermometer.

58f Heat of Combustion of Coal, by Analysis

The heat of combustion may be computed from the results of the ultimate analysis by using the formula

$$14{,}600\ C+62{,}000\ (H-\frac{O}{8})+4000\ S$$

in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively.

58g Air for Combustion

The quantity of air used may be calculated by the formulae:

$$\text{Lb. of air per lb. of carbon} = \frac{3.032\ N}{CO_2+CO}$$

in which N, CO_2 and CO are the percentages of dry gas obtained by analysis, and

Lb. of air per lb. of coal = lb. air per lb. C × (per cent C in the coal less per cent carbon in refuse referred to coal).

The ratio of the air supply to that theoretically required for complete combustion is $\frac{N}{N - 3.782(O - \frac{1}{2}CO)}$

DATA AND RESULTS

59 The data and results should be reported in accordance with the form printed below (Table 7), adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form of report is desired the items in fine print designated by letters of the alphabet may be omitted; or if only the principal data and results are desired the subjoined abreviated table (Table 8) may be used. Unless otherwise indicated, the items should be the averages of the data.

CHART

60 In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be plotted on a chart and represented graphically. See Appendix No. 23.

TESTS WITH OIL AND GAS FUELS

61 Tests of boilers using oil or gas for fuel should accord with the rules here given, excepting as they are varied to conform to the particular characteristics of the fuel. The proper length of tests with gas and oil fuels may be determined by a consideration of the probable errors and the degree of accuracy desired, the minimum duration for economy tests being 5 hours. With these fuels the "flying" method of starting and stopping is employed.

The table of data and results should contain items stating character of furnace and burner, quality and composition of oil or gas, temperature of oil, and data regarding the performance of apparatus supplying the fuel.

TABLE 7 DATA AND RESULTS OF EVAPORATIVE TEST

Code of 1915

(1) Test of ....boiler located at ....
To determine ....
Test conducted by ....

Dimensions

(2) Number and kind of boilers ....
(3) Kind of furnace ....
(4) Grate surface (width ——— length ———)*..................sq. ft.
(a) Approximate width of air openings in grate...............................in.
(b) Percentage of area of air openings to grate surface...................per cent

(5) Water heating surface ....sq. ft.
(6) Superheating surface ....sq. ft.
(7) Total heating surface...sq. ft.
(a) Ratio of water heating surface to grate surface..........................(—) to 1
(b) Ratio of total heating surface to grate surface..........................(—) to 1
(c) Ratio of minimum draft area to grate surface..........................1 to (—)
(d) Volume of combustion space between grate and heating surface..........cu. ft.
(e) Distance from center of grate to nearest heating surface...................ft.

Date, Duration, etc.

(8) Date ....
(9) Duration ....hr.
(10) Kind and size of coal ....

Average Pressures, Temperatures, etc.

(11) Steam pressure by gage..............................lb. per sq. in.
(a) Barometric pressure ....in. of mercury

(12) Temperature of steam, if superheated ....deg.
(a) Normal temperature of saturated steam...........................deg.

(13) Temperature of feed water entering boiler ....deg.
(a) Temperature of feed water entering economizer...................deg.
(b) Increase of temperature of water due to economizer................deg.

(14) Temperature of escaping gases leaving boiler ....deg.
(a) Temperature of gases leaving economizer..........................deg.
(b) Decrease of temperature of gases due to economizer................deg.
(c) Temperature of furnace..deg.

(15) Force of draft between damper and boiler................in. of water
(a) Draft in main flue near boiler............................in. of water
(b) Draft in main flue between economizer and chimney.........in. of water
(c) Draft in furnace..in. of water
(d) Draft or blast in ash pit..................................in. of water

(16) State of weather ....
(a) Temperature of external air.....................................deg.
(b) Temperature of air entering ash pit**..........................deg.
(c) Relative humidity of air entering ash pit.....................per cent

* Unless otherwise designated this is the total area enclosed within the furnace walls projected horizontally.

** Thermometer should be protected from direct radiation of boiler and furnace.

Quality of Steam

(17) Percentage of moisture in steam or number of degrees of superheating ....per cent or deg.
(18) Factor of correction for quality of steam ....

Total Quantities

(19) Total weight of coal as fired† ....lb.
(20) Percentage of moisture in coal as fired....................per cent
(21) Total weight of dry coal (Item 19 $\times \left[1 - \frac{\text{Item } 20}{100}\right]$) ....lb.
(22) Ash, clinkers and refuse (dry)
  (*A*) Withdrawn from furnace and ash pit....................lb.
  (*B*) Withdrawn from tubes, flues and combustion chamber..lb.
  (*C*) Blown away with gases....................lb.
  (*D*) Total ....lb.
  (*a*) Weight of clinkers contained in total ash....................lb.
(23) Total combustible burned (Item 21 — Item 22D)** ....lb.
(24) Percentage of ash and refuse based on dry coal....................per cent
(25) Total weight of water fed to boiler§ ....lb.
(26) Total water evaporated, corrected for quality of steam (Item 25 × Item 18) ....lb.
(27) Factor of evaporation based on temperature of water entering boiler ....
(28) Total equivalent evaporation from and at 212 deg. (Item 26 × Item 27) ....lb.

Hourly Quantities and Rates

(29) Dry coal per hour....................lb.
(30) Dry coal per sq. ft. of grate surface per hour....................lb.
(31) Water evaporated per hour, corrected for quality of steam....................lb.
(32) Equivalent evaporation per hour from and at 212 deg.* ....lb.
(33) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface* ....lb.

Capacity

(34) Evaporation per hour from and at 212 deg. (same as Item 32)......lb.
  (*a*) Boiler horsepower developed (Item 34÷34½).................... bl.-h.p.
(35) Rated capacity per hour, from and at 212 deg. ....lb.
  (*a*) Rated boiler horsepower....................bl.-h.p.
(36) Percentage of rated capacity developed ....per cent

† The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.
** If either of the two items 22*B* and 22*C* is omitted, the fact should be so stated.
§ Corrected for inequality of water level and of steam pressure at beginning and end.
* The symbol "U. E." meaning Units of Evaporation, may be substituted for the expression, Equivalent evaporation from and at 212°.

ECONOMY

(37) Water fed per lb. of coal as fired (Item 25 ÷ Item 19) ....lb.
(38) Water evaporated per lb. of dry coal (Item 26 ÷ Item 21) ....lb.
(39) Equivalent evaporation from and at 212 deg. per lb. of coal as fired (Item 28 ÷ Item 19) ....lb.
(40) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 28 ÷ Item 21) ....lb.
(41) Equivalent evaporation from and at 212 deg. per lb. of combustible (Item 28 ÷ Item 23) ....lb.

EFFICIENCY

(42) Calorific value of 1 lb. of dry coal by calorimeter* ....B.t.u.
(a) Calorific value of 1 lb. dry coal by analysis ....B.t.u.

(43) Calorific value of 1 lb. of combustible by calorimeter ....B.t.u.
(a) Calorific value of 1 lb. combustible by analysis ....B.t.u.

(44) Efficiency of boiler, furnace, and grate

$$\left[100\times\frac{\text{Item } 40\times 970.4}{\text{Item } 42}\right]$$ See Appendix No. 32......... per cent

(45) Efficiency based on combustible

$$\left[100\times\frac{\text{Item } 41\times 970.4}{\text{Item } 43}\right]$$ .... per cent

COST OF EVAPORATION

(46) Cost of coal per ton of ——— lb. delivered in boiler room......dollars
(47) Cost of coal required for evaporating 1,000 lb. of water under observed conditions.................................dollars
(48) Cost of coal required for evaporating 1,000 lb. of water from and at 212 deg..................................dollars

SMOKE DATA

(49) Percentage of smoke as observed ....per cent
(a) Weight of soot per hour obtained from smoke meter............per cent

FIRING DATA

(50) Kind of firing, whether spreading, alternate, or coking ....
(a) Average thickness of fire...in.
(b) Average intervals between firings for each furnace during time when fires are in normal condition...min.
(c) Average interval between times of leveling or breaking up................min.

*If the calorific value is desired per lb. of coal "as fired," multiply Item 42 by $\frac{100-\text{Item } 20}{100}$

(51) Analysis of dry gases by volume

(a) Carbon dioxide (CO_2)........per cent
(b) Oxygen (O)........per cent
(c) Carbon monoxide (CO)........per cent
(d) Hydrogen and hydrocarbons........per cent
(e) Nitrogen, by difference (N)........per cent

(52) Proximate analysis of coal

	As fired	Dry coal	Combustible
(a) Moisture........	....	....	....
(b) Volatile matter........	....	....	....
(c) Fixed carbon........	....	....	....
(d) Ash........	....	....	....
	100 per cent	100 per cent	100 per cent

(e) Sulphur, separately determined referred to dry coal........per cent

(53) Ultimate analysis of dry coal

(a) Carbon (C)........per cent
(b) Hydrogen (H)........per cent
(c) Oxygen (O)........per cent
(d) Nitrogen (N)........per cent
(e) Sulphur (S)........per cent
(f) Ash........per cent

100 per cent

(54) Analysis of ash and refuse, etc.

(a) Volatile matter........per cent
(b) Carbon........per cent
(c) Earthy matter........per cent

100 per cent

(d) Sulphur, separately determined........per cent
(e) Fusing temperature of ash........deg.

(55) Heat balance, based on dry coal

	Dry coal B.t.u.	Dry coal Per cent
(a) Heat absorbed by the boiler (Item 40 × 970.4)........		
(b) Loss due to evaporation of moisture in coal........		
(c) Loss due to heat carried away by steam formed by the burning of hydrogen........		
(d) Loss due to heat carried away in the dry flue gases........		
(e) Loss due to carbon monoxide........		
(f) Loss due to combustible in ash and refuse........		
(g) Loss due to heating moisture in air........		
(h) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for........		
(i) Total calorific value of 1 lb. of dry coal (Item 42.)........		100

If it is desired that the heat balance be based on coal "as fired" or on "combustible burned" the items in the first column are multiplied by the proportion

$\frac{100-\text{Item } 20}{100}$ for coal "as fired" or by the proportion $\frac{100-\text{Item } 20}{100-(\text{Item } 20+\text{Item } 24)}$ for "combustible burned."

TABLE 8 PRINCIPAL DATA AND RESULTS OF BOILER TEST

(1) Grate surface (width ——— length ———)....................sq. ft.
(2) Total heating surface ....sq. ft.
(3) Date ....
(4) Duration ....hr.
(5) Kind and size of coal ....

(6) Steam pressure by gage............................lb. per sq. in.
(7) Temperature of feed water entering boiler ....deg.
(8) Percentage of moisture in steam or number of degrees of superheating ....per cent or deg.
(9) Percentage of moisture in coal ....per cent

(10) Dry coal per hour...lb.
(11) Dry coal per sq. ft. of grate surface per hour....................lb.
(12) Equivalent evaporation per hour from and at 212 deg. ....lb.
(13) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of heating surface ....lb.
(14) Rated capacity per hour, from and at 212 deg. ....lb.
(15) Percentage of rated capacity developed ....per cent

(16) Equivalent evaporation from and at 212 deg. per lb. of dry coal......lb.
(17) Equivalent evaporation from and at 212 deg. per lb. of combustible ....lb.

(18) Calorific value of 1 lb. of dry coal by calorimeter ....B.t.u.
(19) Calorific value of 1 lb. of combustible by calorimeter ....B.t.u.
(20) Efficiency of boiler, furnace, and grate.....................per cent
(21) Efficiency based on combustible............................per cent

PART V

RULES FOR CONDUCTING TESTS OF RECIPROCATING STEAM ENGINES

INTRODUCTION

62 The code for steam engine tests applies to tests for determining the performance of the engine alone (including reheaters and jackets, if any) apart from that of steam-driven auxiliaries which are necessary to its operation. For tests of engine and auxiliaries combined, and tests of multiple expansion engines from which steam is withdrawn for heating feedwater or otherwise, refer to the Code for Complete Steam Power Plants, Part IX.

OBJECT AND PREPARATIONS

63 Determine the object of the test, take the dimensions, and note the physical condition not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given in ¶ 1 to 33, and prepare for the test accordingly.

APPARATUS AND INSTRUMENTS

64 The apparatus and instruments required for a performance test of a reciprocating steam engine are:

(a) Tanks and platform scales for weighing water (or water meters calibrated in place).
(b) Graduated scales attached to the water glasses of the boilers if the feedwater is measured.
(c) Pressure gages, vacuum gages, and thermometers.
(d) Steam calorimeter.
(e) Barometer.
(f) Steam engine indicators.
(g) Planimeter.
(h) Tachometer, revolution-counter, or other speed measuring apparatus.
(i) Friction brake, or dynamometer if available.

65 Directions regarding the use and calibration of these appliances are given in ¶ 7 to 9, and in ¶ 24 to 33.

66 The determination of the heat and steam consumption of an engine by feedwater test requires the measurement of the various supplies of water fed to the boiler; that of the water wasted by separators and drips on the main steam line; that of steam used for other purposes than the main engine cylinders; and that of water and steam which escape by leakage of the boiler and piping; all of these last being deducted from the total feedwater measured.

Where a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had that all the steam passing into the cylinders finds its way into the condenser. If the condenser leaks, the defects causing such leakage should be remedied, or suitable correction should be made. The water of condensation from jackets and reheaters, if not included in the air pump discharge, should be added thereto.

67 When no other method is available the steam consumption may be determined by the use of a steam meter, bearing in mind the caution that it should be calibrated under the exact conditions of use.

68 The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should not be included in the total steam from which the heat consumption is calculated, but the quantity of steam thus used should be determined and reported.

69 For fuller particulars, see description of Steam Measuring apparatus, ¶ 9 (*c*), Part I, and Appendices Nos. 3 and 25.

OPERATING CONDITIONS

70 Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out in ¶ 14, Part I.

DURATION

71 A test for steam or heat consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feedwater for this purpose, five hours duration is sufficient. Where a surface condenser is used, and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hourly records may be compared and the time correspondingly reduced.

72 When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

STARTING AND STOPPING

73 The engine and appurtenances having been set to work and thoroughly heated under the prescribed conditions of test, (except in cases where the object is to obtain the performance under working conditions), note the water levels in the boilers and feed reservoir, take the time, and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

74 Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test.

Care should be taken in cases where the activity of combustion in the boiler furnaces affects the height of water in the gage glasses that the same conditions of fire and drafts obtain at the beginning and end of the test. For this reason it is best to start and stop a test without interfering with the regularity of the operation of the feed pump, provided the latter can be regulated to run so as to supply the feedwater at a uniform rate. In some cases where the supply of feedwater is irregular, as, for example, where an injector of excessive capacity is used, the supply of feedwater should be temporarily shut off.

Suitable care should be observed in noting the average height of the water in the glasses, taking sufficient time to satisfactorily judge of the full extent of the fluctuation of the water line, and thereby its mean position.

RECORDS

75 The general data should be recorded as pointed out in Part I, ¶ 15 to 18, under the heading Records. Half-hourly readings of the instruments are sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 15 or 20 minutes, and oftener if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the steam and vacuum gages. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

76 Further directions may be found in Part I, ¶ 9 (*s*), under the heading Indicators, and in Appendix No. 18.

CALCULATION OF RESULTS

77a Dry Steam

The quantity of dry steam consumed is determined by deducting the moisture, if any, found by calorimeter test from the total amount of feedwater (the latter being corrected for leakages and other losses) or from the amount of air pump discharge, as the case may be. If the steam is superheated, no correction is to be made for the superheat.

77b Heat Consumption

The number of heat units consumed by the engine is found by multiplying the weight of feedwater consumed, corrected for moisture in the steam, if any, and for plant leakages and other exterior losses, by the total heat of 1 lb. of steam (saturated or superheated) at the pressure in the steam pipe near the throttle, less the heat in 1 lb. of water at the temperature corresponding to the pressure in the exhaust pipe near the engine.

77c Indicated Horsepower

In a single double-acting cylinder the indicated horsepower is found by using the formula

$$\frac{PLAN}{33{,}000}$$

in which P represents the average mean effective pressure in pounds per sq. in. measured from the indicator diagrams, L the length of stroke in ft., A the area of the piston less one-half the area of the piston rod, or the mean area of the rod if it passes through both cylinder heads, in sq. in., and N the number of single strokes per minute. See Appendix No. 36 for method of determining mean effective pressure.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the results added together.

77d Brake Horsepower

The brake horsepower is found by multiplying together the net pressure or weight in pounds on the brake arm (the gross weight minus the weight when the brake is entirely free from the pulley), the circumference of the circle whose radius is the horizontal distance between the centre of the shaft and the bearing point at the end of the brake arm in feet, and the number of revolutions of the brake shaft per minute, and dividing the product by 33,000.

77e Electrical Horsepower

The electrical horsepower of a direct-connected generator is found by dividing the output at the terminals, expressed in kilowatts, by the decimal 0.7457. In the case of alternating-current generators the net output is to be used, this being the total output less that consumed for excitation and separately-driven ventilating fan. See Appendix No. 21.

77f Efficiency

The thermal efficiency, that is the proportion of the total heat consumption which is converted into work, is found by dividing 2546.5, (B.t.u. equivalent of one h.p-hr.), by the number of heat units actually consumed per h.p-hr.

The efficiency of the Rankine cycle is found by dividing the heat utilized per pound of steam in an ideal engine working on the Rankine cycle between the pressure and temperature in the steam pipe near the throttle and the pressure and temperature in the exhaust pipe near the engine, by the difference between the total heat of 1 lb. of steam at the throttle pressure and temperature (saturated or superheated as the case may be) and the heat of 1 lb. of water at the temperature of the steam in the exhaust pipe near the engine. See Appendix 28.

The Rankine cycle ratio (or the efficiency ratio referred to the Rankine cycle) is found by dividing the efficiency of the actual engine (referred to the i.h.p. or br-h.p., as the case may be), by the efficiency of the Rankine cycle.

77g Steam Accounted for by Indicator Diagrams at Points Near Cut-off and Release

The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13{,}750}{\text{m.e.p.}}\left[(C+E)\,W_c-(H+E)\,W_h\right]$$

in which

m.e.p. = mean effective pressure.

C = proportion of direct stroke completed at points on expansion line near cut-off or release.

E = proportion of clearance.

H = proportion of return stroke uncompleted at point on compression line just after exhaust closure.

W_c = weight of 1 cu. ft. steam at pressure shown at cut-off or release point.

W_h = weight of 1 cu. ft. steam at pressure shown at compression point.

The points near cut-off release and compression referred to are indicated in Fig. 1.

In multiple expansion engines the mean effective pressure to be used in the above formula is the aggregate m.e.p. referred to the cylinder under consideration. In a compound engine the aggregate

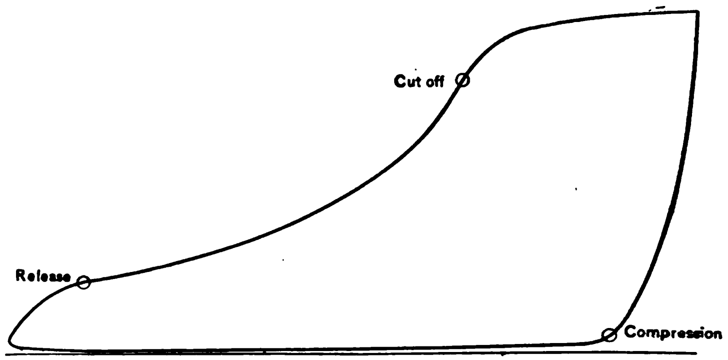


FIG. 1 POINTS WHERE "STEAM ACCOUNTED FOR BY INDICATOR" IS COMPUTED

m.e.p. for the h.p. cylinder is the sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. cylinder multiplied by the cylinder ratio. Likewise the aggregate m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio. (See Appendix No. 34.)

The relation between the weight of steam shown by the indicator at any point in the expansion line and the weight of the mixture of steam and water in the cylinder may be represented graphically by plotting on the diagram a saturated steam curve showing the total consumption per stroke (including steam retained at compression) and comparing the abscissae of this curve with the abscissae of the expansion line, both measured from the line of no clearance.

77h Cut-Off and Ratio of Expansion

To find the percentage of cut-off, or what may best be termed the "commercial cut-off," the following rule should be observed:

> Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the ratio of expansion divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple expansion engine the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

77i Miscellaneous

The method of obtaining a "combined diagram" in compound and other multiple expansion engines is described in Appendix No. 26. The method of finding the "diagram factor" and the manner of determining the "efficiency of the Rankine cycle" are given in Appendices Nos. 27 and 28, respectively.*

DATA AND RESULTS

78 The data and results should be reported in accordance with the form (Table 9) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form is desired the items in fine print designated by letters of the alphabet may be omitted; or if the principal data and results pertaining to steam consumption only are desired, the subjoined abbreviated table (Table 10) may be used. Unless otherwise indicated, the items should be the averages of the data.

(See footnote of Table 9 for reference to engines driving electric generators and other machinery.)

TABLE 9 DATA AND RESULTS OF STEAM-ENGINE TEST

Code of 1915

(1) Test of ....engine located at....................
To determine ....
Test conducted by ....

*For other matters relating to the analysis of engine performance, see treatises on thermodynamics.

Dimensions, etc.

(2) Type of engine (simple or multiple expansion) ....
(3) Class of service (mill, marine, electric, etc.) ....
(4) Auxiliaries (steam or electric driven) ....
- (a) Type and make of condenser equipment ....
- (b) Rated capacity of condenser equipment .... h.p.
- (c) Type of oil pump, jacket pump, and reheater pump (direct or independently driven) ....

(5) Rated power of engine ....
- (a) Name of builders ....
- (b) Kind of valves ....
- (c) Type of governor ....

		1st	2d	3rd
(6)	Diameter of cylinders .... in.	......	......	......
(7)	Stroke of pistons .... ft.	......	......	......
	(a) Diameter of piston-rod, each end .... in.	......	......	......
(8)	Clearance (average) in per cent of piston displacement	......	......	......
(9)	H.P. constant 1 lb. 1 rev. .... h.p.	......	......	......
	(a) Cylinder ratio (based on net piston displacement) .... 1 to —	......	......	......
	(b) Area of interior steam surface .... sq. ft.	......	......	......
	(c) Area of jacketed surfaces .... sq. ft.	......	......	......
(10)	Capacity of generator or other apparatus consuming power of engine .... h.p.	......	......	......

Date and Duration

(11) Date ....
(12) Duration .... hr.

Average Pressures and Temperatures

(13) Pressure in steam pipe near throttle, by gage .... lb. per sq. in.
(14) Barometric pressure .... in. of mercury
- (a) Pressure at boiler, by gage .... lb. per sq. in.

(15) Pressure in 1st receiver, by gage .... lb. per sq. in.
(16) Pressure in 2nd receiver, by gage .... lb. per sq. in.
(17) Pressure in exhaust pipe near engine by gage .... lb. per sq. in.
(18) Vacuum in condenser .... in. of mercury
- (a) Corresponding absolute pressure .... lb. per sq. in.

(19) Pressure in jackets and reheaters .... lb. per sq. in.
(20) Temperature of steam near throttle .... deg.
- (a) Temperature of saturated steam at throttle pressure .... deg.
- (b) Temperature of steam leaving 1st receiver, if superheated .... deg.
- (c) Temperature of steam leaving 2nd receiver, if superheated .... deg.

(21) Temperature of steam in exhaust pipe near engine .... deg.
- (a) Temperature of injection or circulating water entering condenser .... deg.
- (b) Temperature of injection leaving condenser .... deg.
- (c) Temperature of air in engine room .... deg.

Quality of Steam

(22) Percentage of moisture in steam near throttle or number of degrees of superheating ....per cent or deg.

Total Quantities

(23) Total water fed to boilers......................................lb.
(24) Total condensed steam from surface condenser (corrected for condenser leakage) ....lb.
(25) Total dry steam consumed (Item 23 or 24 less moisture in steam)*..lb.

Hourly Quantities

(26) Total water fed to boilers or drawn from surface condenser per hour ....lb.
(27) Total dry steam consumed for all purposes per hour (Item 25 ÷ Item 12) ....lb.
(28) Steam consumed per hour for all purposes foreign to the main engine ....lb.
(29) Dry steam consumed by engine per hour (Item 27 — Item 28)......lb.
 (a) Circulating water supplied to condenser per hour..........................lb.

Hourly Heat Data

(30) Heat units consumed by engine per hour [Item 29 × (total heat of steam per pound at pressure of Item 13 minus heat in 1 lb. of water at temperature of Item 21)]..........................B.t.u.
 (a) Heat converted into work per hour...................................B.t.u.
 (b) Heat rejected to condenser per hour (Item 29a × [Item 21b—21a]) (approximate)..B.t.u.
 (c) Heat rejected in form of uncondensed steam withdrawn from cylinders†..B.t.u.
 (d) Heat lost by radiation...B.t.u.

Indicator Diagrams

		1st Cyl.	2d. Cyl.	3d Cyl.
(31)	Commercial cut-off in per cent of stroke............................per cent	.....	.....	.....
(32)	Initial pressure above atmosphere.....lb. per sq. in.	.....	.....	.....
(33)	Back pressure at lowest point above or below atmosphere................lb. per sq. in.	.....	.....	.....
	(a) Mean back pressure above atmosphere or zero...............lb. per sq. in.	.....	.....	.....

* See footnote Table 6, also first paragraph under "Calculation of Results" ¶77a.
† In multiple expansion engines.

(34) Mean effective pressure.......lb. per sq. in. ....
- (*a*) Equivalent m.e.p. referred to 1st cylinder.......lb. per sq. in. ....
- (*b*) Equivalent m.e.p. referred to 2nd cylinder.......lb. per sq. in. ....
- (*c*) Equivalent m.e.p. referred to 3rd cylinder.......lb. per sq. in. ....

(35) Aggregate m.e.p. referred to each cylinder.......lb. per sq. in. ....

(36) Steam accounted for per i.h.p-hr. at point on expansion line shortly after cut-off.......lb. ....

(37) Steam accounted for per i.h.p-hr. at point on expansion line just before release.......lb. ....
- (*a*) Pressure at selected point near cut-off†.......lb. per sq. in. ....
- (*b*) Pressure at selected point near release.......lb. per sq. in. ....
- (*c*) Pressure at point on compression curve shortly after exhaust closure.......lb. per sq. in. ....
- (*d*) Proportion of direct stroke completed at selected point near cut-off....... ....
- (*e*) Proportion of direct stroke completed at selected point near release....... ....
- (*f*) Proportion of return stroke uncompleted at selected point on compression line....... ....
- (*g*) Ratio of expansion....... ....
- (*h*) M.e.p. of hypothetical diagram (App. 27).......lb. per sq. in. ....
- (*i*) Diagram factor (App. 27)....... ....

Speed

(38) Revolutions per minute.......r.p.m.

(39) Piston speed per minute.......ft.
- (*a*) Variation of speed between no load and full load.......per cent
- (*b*) Momentary fluctuation of speed on suddenly changing from full load to half-load.......per cent

Power

(40) Indicated h.p. developed, whole engine.......i.h.p.
- (*a*) I.h.p. developed by 1st cylinder.......i.h.p.
- (*b*) I.h.p. developed by 2nd cylinder.......i.h.p.
- (*c*) I.h.p. developed by 3rd cylinder.......i.h.p.

(41) Brake h.p.......br. h.p.

(42) Friction of engine (Item 40 — Item 41).......h.p.
- (*a*) Friction expressed in percentage of i.h.p. (Item 42 ÷ Item 40 × 100).......per cent
- (*b*) Indicated h.p. with no load, at normal speed.......i.h.p.

Economy Results

(43) Dry steam consumed by engine per i.h.p-per hr.......lb.

(44) Dry steam consumed by engine per brake h.p-hr.......lb.

(45) Percentage of steam consumed by engine accounted for by indicator at point near cut-off.......per cent

(46) Percentage of steam consumed near release.......per cent

†Pressures all referred to zero.

(47) Heat units consumed by engine per i.h.p-hr.
(Item 30 ÷ Item 40) .... B.t.u.

(48) Heat units consumed by engine per br. h.p-hr.
(Item 30 ÷ Item 41) .... B.t.u.

Efficiency Results

(49) Thermal efficiency of engine referred to i.h.p. [(2546.5 ÷ Item 47) × 100] .... per cent

(50) Thermal efficiency of engine referred to br. h.p. [(2546.5 ÷ Item 48) × 100] .... per cent

(51) Efficiency of Rankine cycle between temperatures of Items 20 and 21....

(52) Rankine cycle ratio referred to i.h.p. (Item 49 ÷ Item 51)..........

(53) Rankine cycle ratio referred to br. h.p. (Item 50 ÷ Item 51)........

Work Done per Heat Unit

(54) Net work per B.t.u. consumed by engine (1,980,000 ÷ Item 48) ... Ft-lb.

Sample Diagrams

(55) Sample diagrams from each cylinder...............................

(a) Steam pipe diagrams.

Note:—For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watt hours, average power factor, etc.; and the economy results based on the electric output embracing the heat units and steam consumed per electric h.p-hr. and per kw-hr., together with the efficiency of the generator. (See table for Steam Turbine Code, Part VI.)

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

TABLE 10 PRINCIPAL DATA AND RESULTS OF RECIPROCATING ENGINE TEST

(1) Dimensions of cylinders ....

(2) Date ....

(3) Duration .... hr.

(4) Pressure in steam pipe near throttle by gage............ lb. per sq. in.

(5) Pressure in receivers .... lb. per sq. in.

(6) Vacuum in condenser.................................. in. of mercury

(7) Percentage of moisture in steam near throttle or number of degrees of superheating .... per cent or deg.

(8) Net steam consumed per hour .... lb.

(9) Mean effective pressure in each cylinder................ lb. per sq. in.

(10) Revolutions per minute .... r.p.m.

(11) Indicated horsepower developed.............................. i.h.p.

(12) Steam consumed per i.h.p-hr.................................. lb.

(13) Steam accounted for at cut-off each cylinder .... lb.

(14) Heat consumed per i.h.p-hr.................................. B.t.u.

PART VI

RULES FOR CONDUCTING TESTS OF STEAM TURBINES AND TURBO-GENERATORS

INTRODUCTION

79 The code for steam turbine tests applies to tests for determining the performance of the turbine alone, apart from that of steam-driven auxiliaries which are necessary to its operation. For tests of turbine and auxiliaries combined, and tests of turbines from which steam is withdrawn for heating feed water or other purposes, refer to the Code for Complete Steam Power Plants, Part IX. For methods of conducting tests of generators, motors, etc., and for general information bearing on the subject, reference may be made to the Standardization Rules of the A. I. E. E.

OBJECT AND PREPARATIONS

80 Determine the object of the test, take the dimensions and note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given in ¶ 1 to 33 and prepare for the test accordingly.

APPARATUS AND INSTRUMENTS

81 The apparatus and instruments required for a performance test of a steam turbine or turbo-generator, are:

(*a*) Tanks and platform scales for weighing water, (or water meters calibrated in place).
(*b*) Graduated scales attached to the water glasses of the boilers.
(*c*) Pressure gages, vacuum gages, and thermometers.
(*d*) Steam calorimeter.
(*e*) Barometer.
(*f*) Tachometer, revolution-counter, or other speed-measuring apparatus.
(*g*) Friction brake or dynamometer.
(*h*) Volt meters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator.

82 Directions regarding the use and calibration of these appliances are given in ¶ 7 to 9, and in ¶ 24 to 33, 83 The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code, Part V.

84 If the steam consumption is determined from the water discharged by the wet vacuum or hot-well pump, correction should be made for water drawn in through the packing glands of the turbine shaft, for condenser leakage, and for any other foreign supply of water.

OPERATING CONDITIONS
DURATION
STARTING AND STOPPING
RECORDS
CALCULATION OF RESULTS

85 The rules pertaining to the subjects Operating Conditions, Duration, Starting and Stopping, Records, and Calculation of Results, are identically the same as those given under the respective headings in the Steam Engine Code, ¶ 71 to 77, with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

DATA AND RESULTS

86 The data and results should be reported in accordance with the form (Table 11) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form of report is desired, the items in fine print designated by letters of the alphabet, may be omitted; or if only the principal data and results are desired, the subjoined abbreviated table (Table 12) may be used. Unless otherwise indicated, the items should be the averages of the data.

TABLE 11 DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TEST

Code of 1915

(1) Test of.....................turbine located at......................
To determine ....
Test conducted by ....

Dimensions, etc.

(2) Type of turbine (impulse, reaction, or combination)................
- (a) Number of stages..
- (b) Condensing or non-condensing......................................
- (c) Diameter of rotors..
- (d) Number and type of nozzles..
- (e) Area of nozzles...
- (f) Type of governor..

(3) Class of service (electric, pumping, compressor, etc) ....

(4) Auxiliaries (steam or electric driven) ....
- (a) Type and make of condensing equipment.............................
- (b) Rated capacity of condensing equipment............................
- (c) Type of oil pumps (direct or independently driven)................
- (d) Type of exciter (direct or independently driven)..................
- (e) Type of ventilating fan, if separately driven.....................

(5) Rated capacity of turbine...
- (a) Name of builders..

(6) Capacity of generator or other apparatus consuming power of turbine..

Date and Duration

(7) Date ....

(8) Duration ....hr.

Average Pressures and Temperatures

(9) Pressure in steam pipe near throttle by gage............lb. per sq. in.

(10) Barometric pressure ....in of mercury
- (a) Pressure at boiler by gage.............................lb. per sq. in.
- (b) Pressure in steam chest by gage.........................lb. per sq. in.
- (c) Pressure in various stages.............................lb. per sq. in.

(11) Pressure in exhaust pipe near turbine, by gage..........lb. per sq. in.

(12) Vacuum in condenser...............................in of mercury
- (a) Corresponding absolute pressure.........................lb. per sq. in.
- (b) Absolute pressure in exhaust chamber of turbine...........lb. per sq. in.

(13) Temperature of steam near throttle.............................deg.
- (a) Temperature of saturated steam at throttle pressure.....................deg.
- (b) Temperature of steam in various stages, if superheated...................deg.

(14) Temperature of steam in exhaust pipe near turbine...............deg.
- (a) Temperature of circulating water entering condenser.....................deg.
- (b) Temperature of circulating water leaving condenser......................deg.
- (c) Temperature of air in turbine room..................................deg

Quality of Steam

(15) Percentage of moisture in steam near throttle, or number of degrees of superheating ....per cent or deg.

Total Quantities

(16) Total water fed to boilers...lb.
(17) Total condensate from surface condenser (corrected for condenser leakage and leakage of shaft and pump glands)................lb.
(18) Total dry steam consumed (Item 16 or 17 less moisture in steam)....lb.

Hourly Quantities

(19) Total water fed to boilers or drawn from surface condenser per hour ....lb.
(20) Total dry steam consumed for all purposes per hour (Item 18 ÷ Item 8) ....lb.
(21) Steam consumed per hour for all purposes foreign to the turbine (including drips and leakage of plant) ....lb.
(22) Dry steam consumed by turbine per hour (Item 20 — Item 21)......lb.
 (a) Circulating water supplied to condenser per hour.......................lb.

Hourly Heat Data

(23) Heat units consumed by turbine per hour [Item 22 × (total heat of steam per pound at pressure of Item 9 less heat in 1 lb. of water at temperature of Item 14)].........................B.t.u.
 (a) Heat converted into work per hour..................................B.t.u.
 (b) Heat rejected to condenser per hour (Item 22a × [Item 14b—Item 14a]) (approximate)...B.t u.
 (c) Heat rejected in the form of steam withdrawn from the turbine..........B.t.u.
 (d) Heat lost by radiation from turbine, and unaccounted for..............B.t.u.

Electrical Data

(24) Average volts, each phase.......................................volts
(25) Average amperes, each phase...........................amperes
(26) Average kilowatts, first meter..................................kw.
(27) Average kilowatts, second meter...............................kw.
(28) Total kilowatts output...kw.
(29) Power factor ....
(30) Kilowatts used for excitation, and for separately driven ventilating fan ....kw.
(31) Net kilowatt output...kw.

Speed

(32) Revolutions per minute..r.p.m.
(33) Variation of speed between no load and full load..............r.p.m.
(34) Momentary fluctuation of speed on suddenly changing from full load to half-load...r.p.m.

Power

(35) Brake horsepower, if determined............................br. h.p.
(36) Electrical horsepower ....e.h.p.

Economy Results

(37) Dry steam consumed by turbine per br. h.p-hr....................lb.
(38) Dry steam consumed per net kw-hr.............................lb.
(39) Heat units consumed by turbine per br. h.p-hr. (Item 23 ÷ Item 35) ....B.t.u.
(40) Heat units consumed per net kw-hr..........................B.t.u.

Efficiency Results

(41) Thermal efficiency of turbine (2546.5 ÷ Item 39) × 100.....per cent
(42) Efficiency of Rankine cycle between temperatures of Items 13 and 14 ....per cent
(43) Rankine cycle ratio (Item 41 ÷ Item 42)........................

Work Done per Heat Unit

(44) Net work per B.t.u. consumed by turbine (1,980,000 ÷ Item 39)...Ft-lb.

TABLE 12 PRINCIPAL DATA AND RESULTS OF TURBINE TEST

(1) Dimensions ....
(2) Date ....
(3) Duration ....hr.

(4) Pressure in steam pipe near throttle by gage............lb. per sq. in.
(5) Vacuum in condenser ....in. of mercury
(6) Percentage of moisture in steam near throttle or number of degrees of superheating ....per cent or deg.
(7) Net steam consumed per hour ....lb.

(8) Revolutions per minute ....r.p.m.
(9) Brake horsepower developed...............................br.h.p.
(10) Kw. output ....kw.

(11) Steam consumed per brake h.p-hr. ....lb.
(12) Heat consumed per brake h.p-hr. ....B.t.u.
(13) Steam consumed per kw-hr. ....lb.
(14) Heat consumed per kw-hr. ....B.t.u.

PART VII

RULES FOR CONDUCTING DUTY TRIALS OF STEAM PUMPING MACHINERY

INTRODUCTION

87 The code for steam pumping machinery tests applies to tests for determining the performance of the engine and pump (including reheaters and jackets, if any) or turbine and pump, apart from that of steam driven auxiliaries, which are concerned in their operation. For tests of the pumping machine and auxiliaries combined, and tests in cases where steam is withdrawn from the engine or turbine for heating the feedwater or other purposes, reference may be made to the Code for Complete Steam Power Plants, Part IX.[1]

OBJECT AND PREPARATIONS

88 Read the general instructions given in ¶ 1 to 33. Determine the object, take the dimensions, note the physical conditions not only of the pumping machinery but of all parts of the plant concerned, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

89 In the case of a reciprocating pump, where it is impracticable to measure the actual quantity of water discharged, determine the quantity of water leakage or slip past the plungers and pump valves, if any, as pointed out in Appendix No. 29.

APPARATUS AND INSTRUMENTS

90 The apparatus and instruments required for a test of pumping machinery are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place).

(*b*) Graduated scales attached to the water glasses of the boilers if the feedwater is measured.

[1] In the case of a pump driven by some other prime mover than a steam engine or steam turbine the code may be modified to suit the particular circumstances. If the pump is driven by an internal combustion engine the modifications should be in accord with the Gas and Oil Engine Code.

(*c*) Pressure gages, vacuum gages, and thermometers.
(*d*) Steam calorimeter.
(*e*) Barometer.
(*f*) Tachometer, revolution-counter, or other speed-measuring apparatus.
(*g*) A weir or other means for measuring the quantity of water pumped.
(*h*) Stroke scales for direct-acting pumps.

91 In trials of a reciprocating steam pumping engine, following additional instruments are required:

(*i*) Steam engine indicators.
(*j*) A planimeter.

In trials of pumping machinery other than steam-driven pumps, means should be provided for determining the shaft (or brake) horsepower delivered to the pump, using for example a belt-transmission dynamometer, or if electric driven, multiplying the kilowatts supplied to the motor by the efficiency taken from a calibration curve of the motor, and dividing the product by 0.7457.

92 Directions regarding the use and calibration of these appliances are given in ¶ 7 to 9 and ¶ 24 to 33.

93 The determination of the heat and steam consumption should conform to the same methods as those described in the Steam Engine Code, Part V.

OPERATING CONDITIONS

94 Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial, as pointed out in ¶ 14.

95 In trials for maximum duty, care should be taken that no air is snifted into the pump cylinders, causing imperfect filling. In such cases, and indeed in all cases where air is thus admitted in sufficient quantity to affect the performance as revealed by indicator diagrams from the water end, the result should be corrected accordingly.

DURATION

STARTING AND STOPPING

RECORDS

96 The rules pertaining to the subjects Duration, Starting and Stopping, and Records, are identically the same as those given under the respective headings in the Steam Engine Code, ¶ 71 to 78, and reference may be made to that code for the necessary directions in these particulars. Where the pump-end is of the reciprocating class, the indicator diagrams should be taken not only from the steam cylinders but also from the water cylinders.

CALCULATION OF RESULTS

97 The rules pertaining to Dry Steam, Heat Consumption, and Indicated Horsepower, are identically the same as those given in ¶ 77a to 77c of the Steam Engine Code; and reference may be made to that code for the necessary directions in these particulars.

97a Water Horsepower

In cases where the water discharged is determined by weir or other measurement, the water horsepower is found by multiplying the weight of water discharged per hour in lb., by the total head in feet,* and dividing the product by 1,980,000.

The water horsepower in a reciprocating pump is found by multiplying together the net area of the plunger in sq. in., the total head* expressed in lb. per sq. in., the length of the stroke in ft., and the number of single strokes per minute, and dividing the final product by 33,000, (after correcting for pump leakage).

97b Duty

The duty per million heat units is found by dividing the number of ft-lb. of work done during the trial by the total number of heat units consumed, and multiplying the quotient by 1,000,000. The amount of work done in the case of reciprocating pumps is found by multiplying together the net area of the plunger in sq. in., by the total head expressed in lb. per sq. in.* by the length of the stroke in ft., and by the total number of single strokes during the trial; finally correcting for the percentage of leakage of the pump. In cases where the water discharged is determined by weir or other measurement, the work done is found by multiplying the weight in pounds of water discharged during the trial by the total head in feet.

* The total head is determined by adding together the pressure shown by the gage on the force main, the vacuum shown by the gage on the suction main, and the vertical distance between the centre of the force-main gage and the point where the suction gage pipe connects to the suction main, all expressed in the same units (lb. per sq. in. or ft. head). A pet-cock should be attached to the gage pipe below each gage cock, and opened occasionally to free the pipe (of air in the case of the force-main gage and of water in the case of the suction gage.) If the suction main is under a pressure instead of a vacuum, the suction gage should be attached at such a level that the connecting pipe may be filled with water when the pet-cock is opened, in which case the correction for difference in elevation of gages is the vertical distance between the centers of the gages and the reading of the suction gage is to be subtracted from that of the force main gage.

If the water is drawn from an open well beneath the pump, the total head is that shown by the force main gage corrected for the elevation of the center of the gage above the level of water in the pump well.

If there is a material difference in velocity of the water at the points where the two gages are attached, a correction should be made for the corresponding difference in "velocity-head."

97c Capacity

In cases where the water discharged is not otherwise measured, the capacity in gal. per 24 hr. for reciprocating pumps is found by multiplying together the net area of the plunger in sq. ft., the length of the stroke in ft., (in direct-connected engines the average length of stroke), the number of single strokes per minute, and the constant 7.48, and correcting for the leakage of the pump.

97d Leakage of Pump

The percentage of leakage in a reciprocating pump is the percentage of the quantity of leakage, found on the leakage trial, to the quantity of water discharged on the duty run determined from plunger displacement.

97e Friction

The percentage of total friction in a reciprocating pump is the percentage of the friction horsepower to the indicated horsepower of the steam cylinders.

97f Miscellaneous

For the calculation of other results pertaining specially to the performance of the steam end of a reciprocating pump, reference may be made to the Steam Engine Code.

DATA AND RESULTS

98 The data and results should be reported in accordance with the form (Table 13) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

99 In the case of a pumping engine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Steam Engine Code may supplement those here given.

TABLE 13 DATA AND RESULTS OF STEAM PUMPING MACHINERY TEST

Code of 1915

(1) Test of..........pump located at..............................
To determine ....
Test conducted by...

Dimensions, etc.

(2) Type of machinery..
(3) Rated capacity in gallons per 24 hr..........................gal.
(4) Size of engine or turbine....................................
(5) Size of pump...
(6) Auxiliaries (steam or electric driven)..........................
(a) Type and make of condenser equipment..................................
(b) Rated capacity of condenser equipment..................................
(c) Type of oil pump, jacket pump, and reheater pump, (direct or independently driven)

Date and Duration

(7) Date . . ..
(8) Duration . . ..hr.

Average Pressures and Temperatures

(9) Pressure in steam pipe near throttle by gage............lb. per sq. in.
(10) Barometric pressure ....in. of mercury
(a) Steam chest pressure...................................lb. per sq. in.
(b) Pressure in receivers and reheaters by gage...............lb. per sq. in.
(c) Pressure in turbine stages by gage........................lb. per sq. in.

(11) Pressure in exhaust pipe near engine or turbine by gage..lb. per sq. in.
(12) Vacuum in condenser ....in. of mercury
(a) Corresponding absolute pressure.........................lb. per sq. in.
(b) Absolute pressure in exhaust chamber....................lb. per sq. in.

(13) Temperature of steam, if superheated, at throttle.................deg.
(a) Normal temperature of saturated steam at throttle pressure........deg.
(b) Temperature of steam leaving receivers, if superheated..............deg.

(14) Temperature of steam in exhaust pipe near engine or turbine......deg.
(a) Temperature of circulating water entering condenser...........deg.
(b) Temperature of circulating water leaving condenser................deg.

(15) Pressure in force main by gage........................lb. per sq. in.
(16) Vacuum or pressure in suction main by gage..in. of mercury or lb. per sq. in.
(a) Correction for difference in elevation of the two gages........lb. per sq. in.

(17) Total head expressed in lb. pressure per sq. in. ....lb. per sq. in.
(a) Total head expressed in ft......................................ft.

QUALITY OF STEAM

(18) Percentage of moisture in steam near throttle, or number of degrees of superheating .... per cent or deg.

TOTAL QUANTITIES

(19) Total water fed to boilers .... lb.

(20) Total condensed steam from surface condenser (corrected for condenser leakage) .... lb.

(21) Total dry steam consumed (Item 19 or 20 less moisture in steam) .... lb.

(22) Total water discharged, by measurement .... gal.

(a) Total water discharged, by plunger displacement, uncorrected .... gal.

(b) Percentage of slip $\left[\frac{\text{Item 22a} - \text{Item 22}}{\text{Item 22a}} \times 100\right]$

(c) Leakage of pump .... gal.

(d) Total water discharged, by calculation from plunger displacement. corrected for leakage .... gal.

(e) Total weight of water discharged, as measured .... lb.

(f) Total weight of water discharged, by calculation from plunger displacement, corrected for leakage .... lb.

HOURLY QUANTITIES

(23) Total water fed to boilers or drawn from surface condenser per hour .. lb.

(24) Total dry steam consumed for all purposes per hour, (Item 21 ÷ Item 8) .... lb.

(25) Steam consumed per hour for all purposes foreign to main engine .... lb.

(26) Dry steam consumed by engine or turbine per hour (Item 24 — Item 25) .... lb.

(a) Circulating water supplied to condenser per hour .... lb.

(27) Weight of water discharged per hour, by measurement .... lb.

(a) Weight of water discharged per hour, calculated from plunger displacement, corrected .... lb.

HOURLY HEAT DATA

(28) Heat units consumed by engine or turbine per hour [Item 26 × (total heat of one lb. of steam at pressure of Item 9, less heat in one lb. of water at temperature of Item 14)] .... B.t.u.

INDICATOR DIAGRAMS

(29) Mean effective pressure, each steam cylinder .... lb. per sq. in.

(a) Mean effective pressure, each water cylinder, if any .... lb. per sq. in.

SPEED AND STROKE

(30) Revolutions per minute .... r.p.m.

(a) Number of single strokes per minute .... strokes

(b) Average length of stroke .... ft.

Power

(31) Indicated horsepower developed ....i.h.p.

(a) Brake horsepower consumed by pump ....

(32) Water horsepower ....h.p.

(33) Friction horsepower (Item 31 — Item 32) ....h.p.

(34) Percentage of i.h.p. lost in friction ....per cent

Capacity

(35) Water discharged in 24 hr., as measured ....gal.

(a) Water discharged in 24 hr., calculated from plunger displacement, corrected ....gal.

(b) Water discharged per minute, as measured ....gal.

(c) Water discharged per minute, calculated from plunger displacement, corrected ....gal.

Economy Results

(36) Heat units consumed per i.h.p-hr ....B.t.u.

(37) Heat units consumed per water h.p-hr. ....B.t.u.

(a) Dry steam consumed per i.h.p-hr ....lb.

(b) Dry steam consumed per water h.p-hr ....lb.

Efficiency Results

(38) Thermal efficiency referred to i.h.p. [(2546.5 ÷ Item 36) × 100].per cent

(a) Thermal efficiency referred to water h.p. [(2546.5 ÷ Item 37) × 100] ....per cent

(b) Mechanical efficiency $\left[\frac{\text{Item } 32}{\text{Item } 31} \times 100\right]$ ....per cent

(c) Pump efficiency $\left[\frac{\text{Item } 32}{\text{Item } 31\text{a}} \times 100\right]$ ....per cent

Duty

(39) Duty per 1,000,000 heat units ....ft-lb.

Work Done per Heat Unit

(40) Work per B.t.u. (1,980,000 ÷ Item 37) ....ft-lb.

Sample Diagrams

(41) Sample indicator diagrams from each steam and pump cylinder.......

NOTE:—The items relating to indicator diagrams and indicated horsepower are to be used only in the case of reciprocating machines.

PART VIII

RULES FOR CONDUCTING TESTS OF STEAM-DRIVEN COMPRESSORS, BLOWERS AND FANS[1]

OBJECT AND PREPARATIONS
APPARATUS AND INSTRUMENTS
OPERATING CONDITIONS
DURATION
STARTING AND STOPPING
RECORDS

100 The directions pertaining to the above divisions of the subject are substantially the same as those given in the Pumping Machinery Code under the same headings, the difference being that the work done is expended in moving air instead of water, and the measurements taken at the air end are modified accordingly.

101 The quantity of air discharged may be measured by a gasometer, or by delivery into tanks of known capacity. Where these means are not available the pitot tube may be used, as described in Appendix No. 6, or some other means may be employed which is subject to calibration. See Par. 9d.

102 If the air end is of the reciprocating type, indicator diagrams should be regularly taken from this end as well as from the steam end.

CALCULATION OF RESULTS

103 The rules pertaining to dry steam, heat consumption, and indicated horsepower of the steam end, are identically the same as those given in Par. 77, of the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars.

103a Air Horsepower

The gross work done at the air end of a reciprocating machine, expressed in horsepower, is found by multiplying together the net area of the air piston in sq. in., the mean effective air pressure in lb. per sq. in. as determined from indicator diagrams, the length of the stroke in ft., and the number of single strokes per minute, and dividing their product by 33,000.

[1] In the case of air machinery driven by some other prime mover than a steam engine or turbine, the code may be modified to meet the particular requirements.

The net work at the air end of either reciprocating or rotary machines, expressed in ft-lb. per minute, is found by multiplying the corrected volume of the compressed air in cu. ft. delivered into the main delivery pipe per minute, by the impact or total pressure in lb. per sq. ft. and by the hyperbolic logarithm of the ratio of the total pressure to the atmospheric pressure (all pressures being absolute pressures). The net air horsepower is found by dividing the product by 33,000. The corrected volume of the compressed air may be found by multiplying the sectional area of the delivery main in sq. ft. by the mean velocity in ft. per minute as determined by pitot tube or other measurement, and reducing the result to atmospheric temperature by multiplying by the proportion.

$$\frac{460+T_1}{460+T_2}$$

in which T_1 is the temperature of the air supplied to the machine and T_2 the temperature of the air in the delivery main.

103b Capacity

The capacity is the number of cu. ft. of air discharged through the delivery main per minute, as determined by gasometer, tank, or other mode of measurement, reduced to the equivalent free air at the atmospheric temperature and pressure. The correction for pressure is made by multiplying by the proportion $\frac{P_2}{P_1}$, in which P_1 is the atmospheric pressure and P_2 the total pressure in the main (absolute pressures), while the correction for temperature is determined as above stated.

The capacity may also be expressed in the number of cu. ft. of compressed air delivered per minute at a given pressure above the atmosphere reduced to the atmospheric temperature.

103c Miscellaneous

For methods of calculating results pertaining especially to the performance of the steam-end of a reciprocating air pumping machine, reference may be made to the Steam Engine Code, ¶ 77.

The "efficiency of compression" in a reciprocating machine is determined by first ascertaining the net work at the air end given above under the heading, "103 a Air Horsepower," and dividing the net work thus found by the gross work given under the same heading.

The "mechanical efficiency" of a reciprocating machine is determined by dividing the gross air horsepower at the air end by the indicated horsepower at the steam end, or by the horsepower delivered by the belt or motor in the case of other means of driving.

DATA AND RESULTS

104 The data and results should be reported in accordance with the form (Table 14) given herewith, adding lines for data not provided for and omitting those not required, as may conform with the object in view. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted. Unless otherwise indicated the items should be the averages of the data.

105 In the case of an air-pumping machine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Steam Engine Code may supplement those here given:

TABLE 14 DATA AND RESULTS OF TEST OF COMPRESSORS, BLOWERS, OR FANS

Code of 1915

(1) Test of..........located at..
To determine ....
Test conducted by...

DIMENSIONS, ETC.

(2) Type of machinery...
(3) Rated capacity in cu. ft. of free air per minute..................cu. ft.
(a) Rated capacity in cu. ft. of air discharged per minute at 100 lb. pressure, reduced to 62 degrees..cu. ft.
(b) Name of builder..

(4) Size of engine or turbine (see Engine and Turbine Codes)............
(5) Size of air cylinders or blowers.................................
(6) Auxiliaries (steam or electric driven)............................
(a) Type and make of condenser equipment..............................
(b) Rated capacity of condenser equipment..............................
(c) Type of oil pump, jacket pump, and reheater pump (direct or independently driven)..

DATE AND DURATION

(7) Date ....
(8) Duration ....hr.

AVERAGE PRESSURES AND TEMPERATURES

(9) Pressure in steam pipe near throttle, by gage............lb. per sq. in.
(10) Barometric pressure ....in. of mercury
(a) Steam chest pressure................................lb. per sq. in.
(b) Pressure in receivers, by gage.......................lb. per sq. in.
(c) Pressure in turbine stages, by gage..................lb. per sq. in.

(11) Pressure in exhaust pipe near engine or turbine, by gage...lb. per sq. in.
(12) Vacuum in condenser...............................in. of mercury

(a) Corresponding absolute pressure..........................lb. per sq. in.
(b) Absolute pressure in exhaust chamber....................lb. per sq. in.

(13) Temperature of steam, if superheated, at throttle................deg.

(a) Normal temperature of saturated steam at throttle pressure...............deg.
(b) Temperature of steam leaving receivers, if superheated...................deg.

(14) Temperature of steam in exhaust pipe near engine or turbine......deg.

(a) Temperature of circulating water entering condenser...................deg.
(b) Temperature of circulating water leaving condenser....................deg.

(15) Temperature of air in delivery main*...........................deg.

(a) Temperature of air supplied to machine...............................deg.
(b) Temperature by wet bulb thermometer.................................deg.
(c) Relative humidity..deg.

(16) Pressure in delivery main by gage (impact pressure)*.....lb. per sq. in.

(a) Pressure in each stage, if more than one.................lb. per sq. in.

Quality of Steam

(17) Percentage of moisture in steam near throttle, or number of degrees of superheating.................................per cent or deg.

Total Quantities

(18) Total water fed to boilers.......................................lb.

(19) Total condensed steam from surface condenser (corrected for condenser leakage) ....lb.

(20) Total dry steam consumed (Item 18 or 19 less moisture in steam)....lb.

(21) Total volume of compressed air delivered, as measured..........cu. ft.

(a) Total volume of compressed air delivered, reduced to atmospheric temperature and pressure...cu. ft.
(b) Total weight of air delivered...lb.

Hourly Quantities

(22) Total water fed to boilers, or drawn from surface condenser, per hour.lb.

(23) Total dry steam consumed for all purposes (Item 20 $\div$ Item 8)....lb.

(24) Steam consumed per hour for all purposes foreign to main engine....lb.

(25) Dry steam consumed by engine or turbine per hour (Item 23 — Item 24) ....lb.

(a) Circulating water supplied to condenser per hour........................lb.

(26) Volume of compressed air delivered per hour, as measured.......cu. ft.

(a) Volume of compressed air delivered per hour, reduced to atmospheric temperature...cu. ft.
(b) Volume of compressed air delivered per hour, reduced to atmospheric temperature and pressure..cu. ft.
(c) Weight of air delivered per hour.......................................lb.

* In the case of compressors or blowers having more than one stage, additional data should be given covering pressures and temperatures in the different stages, the quantity of water used for cooling, and temperatures of the air and water entering and leaving the intercooler.

Hourly Heat Data

(27) **Heat units consumed by engine or turbine per hour (Item 25 multiplied by total heat of 1 lb. of steam at pressure of Item 9, less heat in 1 lb. of water at temperature of Item 14)..............B.t.u.**

Indicator Diagrams

	1st. cyl.	2d Cyl.
(28) **Mean effective pressure, each steam cylinder..lb. per sq. in.**	.....	.....
(a) **Mean effective pressure, each air cylinder..lb. per sq. in.**	.....	.....

Speed

(29) **Revolutions per minute.....................................r.p.m.**

(a) **Number of single strokes per minute...............................strokes**

Power

(30) **Indicated horsepower of steam end............................i.h.p.**

(31) **Gross air horsepower as indicated in air cylinders.................h.p.**

(a) **Brake horsepower consumed by blower or fan.........................br.h.p**

(32) **Net air horsepower (see ¶ 103a)..............................h.p.**

(33) **Friction horsepower (Item 30 — Item 31).......................h.p.**

(34) **Percentage of i.h.p. lost in friction of machine...............per cent**

Capacity

(35) Compressed air delivered per minute as **measured...............cu. ft.**

(a) Compressed air delivered **per minute, reduced to atmospheric temperature ....cu. ft.**

(b) Compressed air delivered **per minute at 100 lb. pressure, reduced to 62 degrees ....cu. ft.**

(36) Compressed air delivered per minute, reduced **to atmospheric** temperature and pressure (free air)...........................cu. ft.

Economy Results

(37) Heat units consumed per i.h.p-hr..............................B.t.u.

(38) Heat units consumed per net h.p-hr. of Item 32................B.t.u.

(39) Dry steam consumed per i.h.p-hr................................lb.

(40) Dry steam consumed per net air h.p-hr. of Item 32.................lb.

Efficiency Results

(41) Thermal efficiency referred to i.h.p. [(2546.5 ÷ **Item 37) × 100]per cent**

(42) Thermal efficiency referred to net air h.p. [(2546.5 ÷ **Item 38)** × 100]...**per cent**

(43) Efficiency of compression [(Item 32 ÷ Item 31) × 100]....per cent

(a) Mechanical efficiency of machine [(Item 31 ÷ Item 30) ×100]..........per cent

(b) Volumetric efficiency [(Item 36 ÷ displacement in cu. ft. per minute of first compressor) ×100]..per cent

Work Done per Heat Unit

(44) Net work per B.t.u. (1,980,000 ÷ Item 38)....................ft-lb.

Sample Diagrams

(45) Sample indicator diagrams from each cylinder.......................

Note:—The items relating to indicator diagrams and indicated horsepower are to be used only in cases where the machine is of the reciprocating type.

PART IX

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM POWER PLANTS

INTRODUCTION

106 The steam power plants to which this code applies are assumed to be plants embracing one or more boilers using coal for fuel, one or more engines or turbines, and the auxiliaries concerned, the power being utilized for any industrial purpose, such as mill driving, generation of electricity, pumping water or compressing air. The object of a test of such a plant is the determination of the performance of the plant as a whole and that of its component parts, and the efficiency of the plant based on coal and steam consumption.

If the plant contains a number of power-generating units, especially if the units are of different types or classes, the test when practicable should determine the performance of each unit, in addition to that of the plant as a whole.

If the boilers supply steam for heating or other industrial purposes in addition to the steam required for power, the various quantities of steam thus used should be measured and the complete distribution of the steam output ascertained.

107 For methods to be followed in testing the component parts of a complete plant, such as boilers, engines, turbines, pumping machinery, and air machinery, reference may be made to the respective Codes which apply thereto.

OBJECT AND PREPARATIONS

108 Read the general instructions given in ¶ 1 to 20, determine the object of the test, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

FUEL

109 Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Boiler Code.

APPARATUS AND INSTRUMENTS

110 The apparatus and instruments required for a performance test of a steam plant are:

- (*a*) Platform scales for weighing coal, ashes and water, and tanks (or water meters calibrated in place).
- (*b*) Graduated scales attached to the water glasses of the boilers.
- (*c*) Pressure gages, vacuum gages, and thermometers.
- (*d*) Coal calorimeter.
- (*e*) Steam calorimeter.
- (*f*) Steam engine indicators.
- (*g*) Planimeter.
- (*h*) Barometer.
- (*i*) Tachometer, revolution and stroke counters, or other speed measuring apparatus.
- (*j*) Friction brake or dynamometer, if available.
- (*k*) Voltmeter, ammeters, wattmeters, and watt-hour meter (in electric plants).
- (*l*) Steam meters.

111 In case of complete pumping machinery or air machinery plants the following additional instruments are required:

- (*m*) Weir, pitot tube, or other apparatus for measuring the discharge.
- (*n*) Stroke scales for direct-acting pumps.

112 Directions regarding the use and calibration of these appliances are given in ¶ 7 to 9 and ¶ 24 to 33.

113 If the test involves the determination of boiler performance and engine or turbine performance, additional instruments should be used as pointed out in the respective Codes referring to such tests.

OPERATING CONDITIONS

114 Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out in ¶ 14.

DURATION

115 The duration of a plant test should be not less than one day of 24 hours, and preferably in some cases, such as ice-making plants, a full week of seven days, including Sunday.

116 Tests to determine the steam consumption of the individual parts of a plant should be of such length as to secure a number of consecutive hourly or half-hourly records showing uniformity within the desired limits of accuracy.

117 In cases where the engine or other power developing machine is in operation only a part of the day, the duration on which the hourly results are computed should be considered the length of time that the engine or other power developing machine is in operation at its working speed.

STARTING AND STOPPING

118 In a plant operating continuously day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and then burned low, say to a thickness of 4 in. to 6 in. if practicable. When this condition is reached the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure. Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low, and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

119 In a plant running only a part of the day, and during the balance of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

RECORDS

120 The general data should be recorded as pointed out in Part I, ¶ 15 to 18, under the head of Records. Half-hourly readings of the various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

121 Further directions may be found in Part I, ¶ 9s, under the heading Indicators and in Appendix No. 18.

SAMPLING AND DRYING COAL

122 During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, ¶ 24 to 26.

ASHES AND REFUSE

123 The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

CALORIFIC TESTS AND ANALYSES OF COAL

124 The quality of the fuel should be determined by calorific tests and analysis of the representative sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, ¶ 9n and ¶ 9p.

CALCULATION OF RESULTS

125a Heat Consumption

The number of heat units consumed by the engine and its auxiliaries per hour is found by multiplying the total weight of the feed water consumed per hour by the total heat of one lb. of steam supplied to the engine, (corrected for moisture or superheat), less the heat in one lb. of water at the temperature of the water entering the boiler (or economizer, if any). If the water is supplied from a number of sources and at various temperatures, the weight of each supply of water is to be taken and the heat in the water at each temperature of supply, the various individual quantities thus obtained being added together to obtain the total heat consumption. (See Appendix No. 25.)

125b Miscellaneous

Directions for calculating the results pertaining individually to the boilers and engine, or other power consuming machinery alone, may be found in the individual codes referring thereto.

DATA AND RESULTS

126 The data and results should be reported in accordance with the form (Table 15) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted; or if only the principal data and results pertaining to coal and power consumption are desired, the subjoined abbreviated table (Table 16) may be used Unless otherwise indicated, the items should be the averages of the data.

REPORT

127 The report should clearly set forth the more important findings and conclusions bearing on the work, taking care that these are not obscured by minor considerations and details. (Read ¶ 1 and ¶ 20).

CHART

128 It is desirable to plot the principal data on a chart, as stated in ¶ 19, especially the indicated horsepower, kilowatt or other load, and steam and coal consumption.

TABLE 15 DATA AND RESULTS OF STEAM POWER PLANT TEST

Code of 1915

(1) Test of ....plant located at ....
To determine ....
Test conducted by ....

DATE, DURATION, ETC.

(2) Number and kind of boilers (superheaters, if any), engines, turbines, etc.
(3) Rated capacity of boilers in lb. of steam per hour from and at 212 deg. lb.
(a) Kind of furnace ....
(b) Grate surface ....sq. ft.
(c) Percentage of area of openings to area of grate ....per cent.
(d) Water heating surface ....sq. ft.
(e) Superheating surface ....sq. ft.

(4) Rated power of engines or turbines................................
(a) Dimensions of cylinders of engine.......................................
(b) Dimensions of turbine...
(c) Type of engines or turbines and class of service ....
(d) Name of builders...

(5) Type of auxiliaries* ....
(a) Dimensions of auxiliaries*...

(6) Type and capacity of condenser ....
(7) Capacity of generators, pumps, or other apparatus consuming power of engine or turbine ....

DATE, DURATION, ETC.

(8) Date ....
(9) Duration. Length of time engine or turbine was in motion with throttle open ....hr.
(a) Length of time engine or turbine was running at normal speed.............hr.
(b) Elapsed time from start to finish.......................................hr.

(10) Kind and size of coal ....

AVERAGE PRESSURES, TEMPERATURES, ETC.

(11) Boiler pressure by gage..............................lb. per sq. in.
(a) Steam pipe pressure near throttle, by gage................lb. per sq. in.
(b) Barometric pressure ....in. of mercury
(c) Steam chest pressure by gage...........................lb. per sq. in.
(d) Pressure in receivers and reheaters by gage...............lb. per sq. in.
(e) Pressure in turbine stages by gage.......................lb. per sq. in.
(f) Pressure in exhaust pipe near engine or turbine...........lb. per sq. in.

(12) Vacuum in condenser...............................in. of mercury
(a) Corresponding absolute pressure.........................lb. per sq. in.
(b) Absolute pressure in exhaust chamber....................lb. per sq. in.

*For full particulars see text of Report.

(13) Temperature of steam, if superheated (taken at boiler or superheater) .... deg.

- (a) Temperature of steam, if superheated (taken at throttle) .... deg.
- (b) Normal temperature of saturated steam at boiler pressure .... deg.
- (c) Normal temperature of saturated steam at throttle pressure .... deg.
- (d) Temperature of steam leaving receivers, if superheated .... deg.
- (e) Temperature of steam in exhaust pipe near engine or turbine .... deg.
- (f) Temperature of condensed water in hot-well or feed tank .... deg.
- (g) Temperature of circulating water entering condenser .... deg.
- (h) Temperature of circulating water leaving condenser .... deg.
- (i) Temperature of air in boiler room .... deg.
- (j) Temperature of air in engine or turbine room .... deg.

(14) Temperature of feed water entering boilers (average) .... deg.

- (a) Temperature of each feed supply (if more than one) .... deg.
- (b) Temperature of feedwater entering economizer, if any .... deg.
- (c) Increase in temperature of water due to economizer .... deg.

(15) Temperature of escaping gases leaving boiler .... deg.

- (a) Temperature of escaping gases leaving economizer .... deg.
- (b) Decrease in temperature of gases due to economizer .... deg.
- (c) Temperature of furnace .... deg.

(16) Force of draft in main boiler flue .... in. of water

- (a) Force of draft at base of chimney .... in. of water
- (b) Force of draft at each end of economizer .... in. of water
- (c) Force of draft at individual boiler dampers .... in. of water
- (d) Force of draft in individual furnaces .... in. of water
- (e) Force of draft or blast in individual ash pits* .... in. of water

(17) State of weather ....

- (a) Temperature of external air .... deg.

Quality of Steam

(18) Percentage of moisture in steam, or number of degrees of superheating .... per cent or deg.

- (a) Factor of correction for quality of steam ....

Total Quantities of Coal and Water

(19) Total weight of coal as fired .... lb.

- (a) Percentage of moisture in coal .... per cent
- (b) Total weight of dry coal .... lb.
- (c) Total ash, clinkers, and refuse (dry) .... lb
- (d) Weight of clinkers contained in total ash .... lb.
- (e) Percentage of ash and refuse in dry coal .... per cent
- (f) Total combustible burned (Item 19b—19c) .... lb.

(20) Total weight of water fed to boiler from all sources† .... lb.

- (a) Total water evaporated corrected for quality of steam (Item 20 × Item 18a) .... lb.
- (b) Factor of evaporation based on average temperature of water entering boiler ....
- (c) Total equivalent evaporation from and at 212 degrees (Item 20a × Item 20b) ... lb.

* If artificial draft or blast is employed, the force of draft or blast at the fan should also be given.

† If there are a number of supplies of feed water, the weight and temperature of each supply is to be given, and total weight and average temperature ascertained.

Hourly Quantities of Coal, Water, and Steam, and Rates

(21) Coal, as fired, per hour (Item 19 ÷ Item 9) .... lb.

- (a) Dry coal per hour (Item 19b ÷ Item 9) .... lb.
- (b) Dry coal per sq. ft. of grate surface .... lb.

(22) Water evaporated per hour (Item 20 ÷ Item 9) .... lb.

- (a) Equivalent evaporation per hour from and at 212 deg .... lb.
- (b) Equivalent evaporation per sq. ft. of water heating surface .... lb.

(23) Dry steam generated per hour (sum of sub-items *a* to *g*) (Item 20 less moisture in steam ÷ Item 9) .... lb.

- (a) Moisture formed per hour between boiler and engine .... lb.
- (b) Dry steam consumed per hour by engine cylinders or turbine .... lb.
- (c) Dry steam consumed per hour by reheaters and jackets, if any .... lb.
- (d) Dry steam consumed per hour by air and circulating pump of condenser ..... lb.
- (e) Dry steam consumed per hour by boiler feed pump .... lb.
- (f) Dry steam consumed per hour by other steam driven auxiliaries .... lb.
- (g) Dry steam consumed per hour to supply leakage of boilers and piping between boilers and engine (including steam supplied for foreign purposes, if any) .... lb.
- (h) Live steam supplied for heating, or miscellaneous purposes .... lb.
- (i) Injection or circulating water supplied condenser per hour .... lb.

Calorific Value of Coal

(24) Calorific value of 1 lb. of coal as fired, by calorimeter test .... B.t.u.

- (a) Calorific value of 1 lb. of dry coal .... B.t.u.
- (b) Calorific value of 1 lb. of combustible .... B.t.u.

Hourly Heat Data

(25) Heat units in coal as fired generated per hour (Item 21 × Item 24) .... B.t.u.

(26) Heat units consumed by engine and auxiliaries per hour (Item 22 × total heat of 1 lb. of steam at pressure of Item 11 less heat in 1 lb. of water at temperature of feed water supplied to boiler, or economizer, if any) .... B.t.u.

- (a) Heat converted into work per hour .... B.t.u.
- (b) Heat rejected to condenser per hour .... B.t.u.
- (c) Heat rejected in steam withdrawn from receivers or turbine-stages not used by feed water .... B.t.u.
- (d) Heat lost by radiation from engine and auxiliaries, including piping between boilers and condenser .... B.t.u.
- (e) Heat lost in operation of boiler, including economizer, (if any) (Item 25—Item 26) .... B.t.u.

Indicator Diagrams

(27) Mean effective pressure, each cylinder .... lb.

- (a) Commercial cut-off (in per cent of stroke) each cylinder .... per cent
- (b) Initial pressure, above atmosphere, each cylinder .... lb. per sq. in.
- (c) Back pressure at lowest point above or below atmosphere, each cylinder .... lb. per sq. in.
- (d) Steam accounted for per i.h.p. per hour at point near cut-off, each cylinder .... lb.
- (e) Steam accounted for per i.h.p. per hour at point near release .... lb.

ELECTRICAL DATA

(28) Average kilowatt output, gross ....kw.

(a) Volts each phase ....volts
(b) Amperes each phase ....amperes
(c) Kilo-volt-amperes ....kv-a.
(d) Power factor ....

(29) Current used by exciter ....kw.

(30) Net kilowatt output (Item 28 — Item 29) ....kw.

SPEED

(31) Revolutions per minute ....r.p.m.

(a) Variation of speed between no load and full load ....r.p.m.

POWER

(32) Indicated horsepower ....i.h.p.

(33) Brake horsepower ....br.h.p.

CAPACITY

(34) Water evaporated per hour from and at 212 degrees (same as Item 22*a*) ....lb.

(a) Percentage of rated boiler capacity developed (Item 34 ÷ Item 3 × 100) ...per cent

(35) Percentage of rated engine or turbine capacity developed (Item 32 ÷ Item 4 × 100) ....per cent

ECONOMY RESULTS

(36) Coal as fired per i.h.p. of engine per hour ....lb.

(37) Coal as fired per brake h.p. of engine or turbine per hour ....lb.

(a) Dry coal per i.h.p. per hr ....lb.
(b) Dry coal per brake h.p-hr ....lb.
(c) Dry coal per kw-hr ....lb.

(38) Heat units in coal consumed per i.h.p. of engine per hour ....B.t.u.

(39) Heat units in coal consumed per brake h.p. of engine or turbine per hour (Item 37 × Item 24) ....B.t.u.

(a) Heat units consumed by engine (including auxiliaries) per i.h.p-hr..B.t.u.
(b) Heat units consumed by engine or turbine (including auxiliaries) per brake h.p-hr. (Item 26 ÷ Item 33) ....B.t.u.
(c) Heat units consumed by engine per kw-hr ....B.t.u.

(40) Heat units in coal consumed per kw-hr ....B.t.u.

(41) Water evaporated per lb. of coal as fired ....lb.

(a) Water evaporated per lb. of dry coal ....lb.
(b) Equivalent evaporation from and at 212 deg. per lb. of dry coal......lb.
(c) Equivalent evaporation from and at 212 deg. per lb. of combustible....lb.

(42) Dry steam consumed by engine alone per i.h.p-hr ....lb.

(a) Dry steam consumed by auxiliaries per i.h.p-hr ....lb.
(b) Dry steam consumed by combined engine and auxiliaries per i.h.p-hr..lb.

(43) Dry steam consumed by engine or turbine alone per brake h.p-hr....lb.

(a) Dry steam consumed by auxiliaries per brake h.p-hr.................lb.

(b) Dry steam consumed by combined engine or turbine and auxiliaries per brake h.p-hr...lb.

Efficiency Results

(44) Thermal efficiency of plant referred to i.h.p. [(2546.5 ÷ Item 38) × 100]...

(45) Thermal efficiency of plant referred to brake h.p. [(2546.5 ÷ Item 39) × 100]...

(a) Efficiency of boilers (Item 41b × 970.4 × 100 ÷ Item 24a)................

(b) Efficiency of engine referred to i.h.p. [(2546.5 ÷ Item 39a) × 100]...........

(c) Efficiency of engine or turbine referred to brake h.p. [(2546.5÷39b) × 100]....

Fuel Cost of Power

(46) Cost of coal per ton of ——— lb...............................dollars

(47) Cost of coal per i.h.p-hr......................................cents

(48) Cost of coal per brake h.p-hr..................................cents

HEAT BALANCE OF STEAM POWER PLANT

(See Appendix 35 for Example)

	Per lb. coal as fired	Per cent
(49) Heat units in coal (same as Item 24)............		
(50) Boiler losses..................................		
(a) Loss due to evaporation of moisture in coal...........		
(b) Loss due to heat carried away by steam formed by the burning of hydrogen..............................		
(c) Loss due to heat carried away in the dry flue gases.....		
(d) Loss due to carbon monoxide.......................		
(e) Loss due to combustible in ash and refuse.............		
(f) Loss due to heating moisture in air..................		
(g) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for.....................		
(h) Heat supplied steam driven appliances for operating boilers less that recovered by heating feed water......		
(i) Total boiler losses................................		
(51) Engine consumption............................		
(a) Radiation from steam pipe..........................		
(b) Radiation from engine or turbine....................		
(c) Heat rejected to condenser..........................		
(d) Heat withdrawn from engine receivers or turbine stages or other use than heating feed water................		
(e) Heat lost by leakage of steam piping..................		
(f) Heat converted into work...........................		

	Per lb. coal as fired	Per cent
(52) Heat in steam supplied for purposes foreign to engine or turbine		
Totals (same as Item 49)........................		

SAMPLE DIAGRAMS

(53) Sample indicator diagrams from each cylinder of engine. Also sample steam pipe diagrams........

The boiler output, (Item 49—Item 50*i*) may be divided into		
(*a*) Heat units absorbed by water in boiler................		
(*b*) Heat units absorbed by water in economiser..........		
The quantity representing the sum of Items 51*b*, *c*, and *f* may be divided according to the steam distribution into		
(*c*) Heat consumed by engine cylinders or turbine alone (including reheaters or jackets, if any), i.e., total heat supplied to engine or turbine alone less heat recovered therefrom by heating feed water..................		
(*d*) Heat consumed by steam-driven auxiliaries, i.e., total heat supplied to auxiliaries less heat recovered therefrom by heating feed water......................		
The same quantity may be divided according to the distribution of work done by engine or turbine into......		
(*e*) Heat consumed in supplying power lost in friction of engine or turbine................................		
(*f*) Heat consumed in supplying frictional, electrical, or other losses of power delivered by engine or turbine shaft...		
(*g*) Heat consumed in supplying useful power delivered by engine or turbine, whether mechanical, electrical, or otherwise......................................		

NOTE:—In the case of pumping and air machinery plants add lines under the various items as follows:

For Item (13)

(*k*) Pressure in delivery main by gage......................lb. per sq. in.
(*l*) Vacuum or pressure in suction main by gage. lb. per sq. in. or in. of mercury
(*m*) Correction for difference in elevation of the two gages.......lb. per sq. in.
(*n*) Total head expressed in lb. pressure per sq. in............lb. per sq. in.
(*o*) Total head expressed in ft...ft.

For Item (20)

(*d*) Temperature of delivery..deg.
(*e*) Total weight of water discharged, by measurement...........................lb.
(*f*) Total weight of water discharged, by calculation from plunger displacement, correctedlb
(*g*) Total volume of air delivered, by measurement.............................cu. ft.
(*h*) Total volume of air delivered, reduced to atmospheric pressure and temperature. cu. ft.

For Item (23)

(*j*) Weight of water discharged per hour, by measurement.........................lb.
(*k*) Weight of water discharged per hour, by plunger displacement, corrected.........lb
(*l*) Volume of water or air delivered per hour, by measurement..................cu. ft.
(*m*) Volume of air delivered per hour, reduced to atmospheric pressure and temperature cu. ft.

For Item (31)

(*b*) Length of pump stroke .... ft.

For Item (33)

(*a*) Water (or air) h.p. .... h.p.

For Item (35)

(*a*) Gal. of water discharged in 24 hr. as measured .... gal.

(*b*) Volume of air delivered per minute, reduced to atmospheric pressure and temperature .... cu. ft.

For Item (36)

(*a*) Dry coal per water (or air) h.p-hr .... lb.

For Item (39)

(*a*) Duty per 1,000,000 B.t.u. ....

For Item (45)

(*a*) Thermal efficiency of plant referred to water (or air) h.p. ....

For Item (48)

(*a*) Cost of coal per water (or air) h.p. .... dollars

TABLE 16 PRINCIPAL DATA AND RESULTS OF STEAM POWER PLANT TEST

(1) Dimensions of boilers ....

(2) Dimensions of engine or turbine ....

(3) Date ....

(4) Duration .... hr.

(5) Boiler pressure .... lb. per sq. in.

(6) Throttle pressure .... lb. per sq. in.

(7) Pressure in receiver or stages .... lb. per sq. in.

(8) Vacuum in condenser .... in. of mercury

(9) Percentage of moisture in steam near throttle or number of degrees of superheating .... per cent or deg.

(10) Temperature of feed water entering boilers .... deg.

(11) Temperature of escaping gases .... deg.

(12) Force of draft .... in. of water

(13) Coal, as fired, per hour .... lb.

(14) Percentage of moisture in coal .... per cent

(15) Percentage of ash in coal .... per cent

(16) Water evaporated per hour .... lb.

(17) Equivalent evaporation per hour from and at 212 deg .... lb.

(*a*) Equivalent evaporation per hour from and at 212 deg. per sq. ft. water heating surface .... lb.

(18) Steam consumed per hour by engine .... lb.

(19) Steam consumed per hour by engine or turbine and auxiliaries .... lb.

(20) Mean effective pressure in each cylinder of engine .... lb. per sq. in.

(21) Revolutions per minute .... r.p.m.

(22) Indicated horsepower .... i.h.p.

(23) Brake horsepower* .... brake h.p.

(24) Coal as fired per i.h.p-hr. . . . . lb.
(25) Coal as fired per brake h.p-hr.* . . . . lb.
(26) Steam per i.h.p-hr . . . . lb.
(27) Steam per brake h.p-hr.* . . . . lb.
(28) Heat consumed per i.h.p-hr. . . . . B.t.u.
(29) Heat consumed per brake h.p-hr.* . . . . B.t.u.

* For pumping engine (water or air) use Water or Air h.p. in place of Brake h.p.

PART X

RULES FOR CONDUCTING TESTS OF LOCOMOTIVES

CODE FOR LABORATORY TESTS

INTRODUCTION

129 Locomotive tests are of two leading kinds, laboratory tests and road tests. The former are made under conditions quite similar to those of a stationary testing plant in which the power is absorbed by a brake. The latter are made under conditions of service on the road, the locomotive hauling a train of cars. Being a complete steam plant in itself, embracing boiler, engine, and certain auxiliaries, a locomotive equipped for a laboratory test should be considered in the same class as a stationary plant, and the code of rules for such tests are therefore similar to those given in Part IX.

130 In view of the high rate of combustion which usually occurs in a locomotive fire-box, and the impossibility of starting and stopping a laboratory test under the conditions of the working load by using the method of thin fires required for correct coal measurements, as in tests of stationary boilers, it is necessary whenever one of the objects of the test is to determine the water evaporated per pound of coal, to precede the main trial by taking preliminary measurements of coal and water during the period employed in building up the fire from the starting condition to the working condition; and at its close to continue these measurements during the period required to bring the fire at the end of the test to the same condition as that of starting. The complete run, thus made, furnishes data for determining the evaporative performance of the boiler, while the intermediate run, which constitutes the main trial, determines the steam consumption of the engine.

The weight of coal fired during the main trial is not an exact measure of the coal actually consumed during that period, owing to the difficulty of insuring the same condition and thickness of fire-bed at the end of the period as at its beginning. If it were possible to make the thickness the same, it would nevertheless be impossible to make the condition of the fire the same, because of the presence of an indeterminate quantity of ash and clinkers at the end which was not present at the beginning.

OBJECT AND PREPARATIONS

131 The object of a laboratory test, as covered by this code, is the determination of the coal and steam consumption per unit of power when the locomotive is operated under fixed conditions. There are other less prominent objects, such as a determination of the performance of the engine alone under different conditions, or of the boiler running under various rates of combustion, but tests having these objects are substantially covered by the Boiler and Engine Codes of Parts IV and V, to which reference may be made.

132 Note the general instructions given in ¶ 1 to 33 so far as they pertain to the work in hand. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and make preparations for the test accordingly.

133 It is assumed that a testing plant is available where the driving wheels can be mounted upon the supporting wheels of a friction brake apparatus for suitably disposing of the power. See Appendix No. 33.

FUEL[1]

134 Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired the fuel should be some kind of coal that is regarded as a standard for the locality where the test is made, as noted in the Boiler Code.

APPARATUS AND INSTRUMENTS

135 The apparatus and instruments required for a laboratory test of a locomotive are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Coal calorimeter.
(*c*) Tanks and scales for weighing water.
(*d*) Graduated scale attached to the water glass.
(*e*) Pressure gages, draft gages, pyrometers, and thermometers.
(*f*) Steam calorimeter.
(*g*) Steam engine indicators.
(*h*) Planimeter.

[1] See ¶ 147 for memoranda relating to oil fuel.

(*i*) Tachometer, or other speed-measuring apparatus.
(*j*) Friction brake apparatus for absorbing the power.
(*k*) Dynamometer for determining the pull on the drawbar.
(*l*) Gas analysis apparatus.

136 Directions regarding their use and calibration are given in ¶ 7 to 9, ¶ 24 to 33, and Appendix No. 34.

OPERATING CONDITIONS

137 Determine what the operating conditions and method of firing should be as explained in ¶ 14, and see that they prevail throughout the trial. It is important that the firing should be in the hands of skilled firemen.

DURATION

138 The duration of the main trial of a laboratory test depends upon the character of the fuel, upon the rate of combustion, and upon the working limitations of the revolving parts of the locomotive and brake apparatus. Whenever practicable, the main trial should continue not less than two hours.

139 On account of liability of error in estimating the difference in the amount of coal in the fire-box at beginning and end, the results of a single test should not be relied upon. If practicable, repeated tests should be made under the required conditions and the results averaged.

STARTING AND STOPPING

140 Having raised steam in the boiler by starting a new fire or opening and spreading the bank of an old one, burn the fire as low as practicable, and clean it (unless the fire is already new and clean), then remove the ash and refuse from the ash-pan. Note the thickness of the fire-bed, the water levels, and the steam pressure, observe the time and consider this the starting time of the boiler trial. Fire with weighed coal, and proceed with the measurements of coal, water, steam pressure, and feed temperature, which pertain to the boiler work. At the same time start the engine and gradually put on the load; building up the fire to its working thickness and establishing the desired operating conditions as rapidly as the nature of the fuel and other circumstances will permit. Then note the amount of coal thus far fired and the thickness of the fire-bed. Note the water levels, the steam pressure and the time, and consider this the starting time of the main trial. Thereafter proceed with the work of the test in full. Maintain throughout the run the thickness of fire noted, using the rocking grates as needed, keeping also a uniform load and uniform water level. When the test has continued until the end of the period determined on, discontinue firing, take the time, and consider this the stopping time of the main trial. Note again the thickness of the fire, the water levels, and steam pressure. Thereafter burn the fire down, reduce the load, and finally stop the engine. Then clean the fire and leave as near as may be the same amount of live coal on the grate as that noted at the beginning of the boiler trial. Again note the thickness of the fire, water levels, and the steam pressure, observe the time, and consider this the stopping time of the boiler trial. Finally, remove the ash and refuse from the ash-pan, the cinders from the smoke-box, and the sparks from the spark collector.

RECORDS

141 The general data should be recorded as pointed out in Part I, ¶ 15 to 18. Readings of the instruments concerned and a set of indicator diagrams should be obtained every ten minutes. Directions on the subject of indicating may be found in ¶ 9s, ¶ 77c, and Appendix No. 18.

SAMPLING AND DRYING COAL

142 During the progress of the trial the coal should be regularly sampled and its moisture determined in the manner pointed out in ¶ 24 to 26.

ASHES AND REFUSE

143 The ashes and refuse withdrawn from the ash-pan and smoke-box and the sparks withdrawn from the spark collector at the end taken if desired for analysis and for calorimeter test.

CALORIFIC TESTS OF COAL

144 The quality of the coal should be determined by a calorimeter test of the sample above referred to. Methods of making this test are described in Appendix No. 12.

CALCULATION OF RESULTS

145 For methods of calculating the results, reference may be made to ¶ 58 of the Boiler Code and ¶ 77 of the Engine Code.

DATA AND RESULTS

146 The data and results should be reported in accordance with the form (Table 17) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

TESTS WITH OIL FUEL

147 When oil fuel is used the rules governing the tests may be modified to conform to the characteristics of liquid fuel. The "flying" method of starting and stopping may be employed, and the duration reduced.

TABLE 17 DATA AND RESULTS OF LABORATORY TEST OF LOCOMOTIVE

Code of 1915

(1) Test of ....locomotive, located at ....
To determine ....:..........................
Test conducted by ....

DIMENSIONS, ETC.

(2) Type and class of locomotive ....

(a) Number of locomotive..
(b) Driving wheels, number of pairs and diameter.............................
(c) Truck wheels, (engine) number of pairs and diameter.......................
(d) Trailing wheels, number of pairs and diameter.............................

Hourly Quantities and Rates, Main Trial

(24) Water evaporated per hour (Item 21 ÷ Item 7) (dry steam consumed by engine) .... lb.
- (a) Equivalent evaporation per hour from and at 212 deg. .... lb.
- (b) Equivalent evaporation per hour from and at 212 deg. per sq. ft. waterheating surface .... lb.

(25) Coal as fired per hour (Item 22 ÷ Item 7) .... lb.
- (a) Dry coal per hour (Item 22b ÷ Item 7) .... lb.
- (b) Dry coal per hour per sq. ft. of grate .... lb

Indicator Diagrams

(26) Mean effective pressure, each cylinder .... lb. per sq. in.
- (a) Commercial cut-off .... per cent
- (b) Initial pressure above atmosphere .... lb. per sq. in.
- (c) Back pressure at lowest point above atmosphere .... lb. per sq. in.
- (d) Steam accounted for per i.h.p-hr. at point near cut-off .... lb.
- (e) Steam accounted for per i.h.p-hr. at point near release .... lb.

Speed

(27) Revolutions per minute .... r.p.m.
- (a) Piston speed in ft. per minute .... ft.
- (b) Equivalent speed of locomotive in miles per hour .... miles

Power

(28) Indicated horsepower .... i.h.p.

(29) Drawbar pull .... lb.

(30) Dynamometer h.p. .... h.p.
- (a) Brake h.p. .... h.p

(31) Friction h.p. (Item 28 — Item 30) .... h.p.
- (a) Percentage of friction .... per cent

Economy Results, Main Trial

(32) Coal as fired per i.h.p-hr. (Item 25 ÷ Item 28) .... lb.
- (a) Dry coal per i.h.p-hr .... lb.
- (b) Dry coal per dynamometer h.p-hr .... lb.

(33) Heat units in coal as fired per i.h.p-hr .... B.t.u.

(34) Dry steam per i.h.p-hr .... lb.
- (a) Dry steam per dynamometer h.p-hr .... lb.

(35) Heat units consumed by engine per i.h.p-hr .... B.t.u.
- (a) Heat units consumed by engine per dynamometer h.p-hr .... B.t.u.

(36) Water evaporated per lb. of dry coal .... lb.
- (a) Equivalent evaporation from and at 212 deg. per lb. dry coal .... lb.
- (b) Equivalent evaporation from and 212 deg. per lb. combustible .... lb.

EFFICIENCY MAIN TRIAL

(37) Thermal efficiency of locomotive referred to i.h.p. [(2546.5 ÷ Item 33) × 100] ....per cent

(38) Thermal efficiency of engine alone referred to i.h.p. [(2546.5 ÷ Item 35) × 100] ....per cent

(a) Thermal efficiency of engine alone referred to dynamometer h.p. [(2546.5 ÷ Item 35a) × 100] ....per cent

(39) Efficiency of boiler [(Item 36a × 970.4) ÷ (Item 23a × 100)] ....per cent

SMOKE DATA

(40) Percentage of smoke as observed..per cent

FIRING DATA

(41) Approximate thickness of fire-bed..in.

(a) Average interval between firings..

ANALYSIS OF DRY GASES BY VOLUME

(42) Carbon dioxide (CO_2)..per cent

(a) Oxygen (O)..per cent

(b) Carbon monoxide (CO)..per cent

HEAT BALANCE OF LOCOMOTIVE

(If a heat-balance is desired, the form given in the table for Complete Steam Power Plants, Table 15, may be followed, using such modifications as may be needed to adapt it to locomotives).

CODE FOR ROAD TESTS

OBJECT AND PREPARATIONS

148 This code relates to road-tests in which the principal object is the determination of the coal and steam consumption of a locomotive per unit of power under the practical conditions of railroad service. The preparation for a road test are the same as those noted for the laboratory test, except for restrictions imposed by the regular operation of the locomotive, and in addition those required for data to be obtained on the road. See Appendix No. 34.

149 It is assumed that a dynamometer car is available for registering the amount of pull on the drawbar, in the manner referred to in Appendix No. 20.

150 To facilitate the work of the men who operate the indicators

Hourly Quantities and Rates, Main Trial

(24) Water evaporated per hour (Item 21 ÷ Item 7) (dry steam consumed by engine) .... lb.
- (a) Equivalent evaporation per hour from and at 212 deg .... lb.
- (b) Equivalent evaporation per hour from and at 212 deg. per sq. ft. waterheating surface .... lb.

(25) Coal as fired per hour (Item 22 ÷ Item 7) .... lb.
- (a) Dry coal per hour (Item 22b ÷ Item 7) .... lb.
- (b) Dry coal per hour per sq. ft. of grate .... lb

Indicator Diagrams

(26) Mean effective pressure, each cylinder .... lb. per sq. in.
- (a) Commercial cut-off .... per cent
- (b) Initial pressure above atmosphere .... lb. per sq. in.
- (c) Back pressure at lowest point above atmosphere .... lb. per sq. in.
- (d) Steam accounted for per i.h.p-hr. at point near cut-off .... lb.
- (e) Steam accounted for per i.h.p-hr. at point near release .... lb.

Speed

(27) Revolutions per minute .... r.p.m.
- (a) Piston speed in ft. per minute .... ft.
- (b) Equivalent speed of locomotive in miles per hour .... miles

Power

(28) Indicated horsepower .... i.h.p.

(29) Drawbar pull .... lb.

(30) Dynamometer h.p. .... h.p.
- (a) Brake h.p. .... h.p

(31) Friction h.p. (Item 28 — Item 30) .... h.p.
- (a) Percentage of friction .... per cent

Economy Results, Main Trial

(32) Coal as fired per i.h.p-hr. (Item 25 ÷ Item 28) .... lb.
- (a) Dry coal per i.h.p-hr .... lb.
- (b) Dry coal per dynamometer h.p-hr .... lb.

(33) Heat units in coal as fired per i.h.p-hr .... B.t.u.

(34) Dry steam per i.h.p-hr .... lb.
- (a) Dry steam per dynamometer h.p-hr .... lb.

(35) Heat units consumed by engine per i.h.p-hr .... B.t.u.
- (a) Heat units consumed by engine per dynamometer h.p-hr .... B.t.u.

(36) Water evaporated per lb. of dry coal .... lb.
- (a) Equivalent evaporation from and at 212 deg. per lb. dry coal .... lb.
- (b) Equivalent evaporation from and 212 deg. per lb. combustible .... lb.

EFFICIENCY MAIN TRIAL

(37) Thermal efficiency of locomotive referred to i.h.p. [(2546.5 ÷ Item 33) × 100]..per cent
(38) Thermal efficiency of engine alone referred to i.h.p. [(2546.5 ÷ Item 35) × 100].................................... .per cent
(a) Thermal efficiency of engine alone referred to dynamometer h.p. [(2546.5 ÷ Item 35a) ×100]...per cent
(39) Efficiency of boiler [(Item 36a × 970.4) ÷ (Item 23a × 100)] ....per cent

SMOKE DATA

(40) Percentage of smoke as observed...........................per cent

FIRING DATA

(41) Approximate thickness of fire-bed..............................in.
(a) Average interval between firings...

ANALYSIS OF DRY GASES BY VOLUME

(42) Carbon dioxide (CO_2)...................................per cent
(a) Oxygen (O)...per cent
(b) Carbon monoxide (CO)..per cent

HEAT BALANCE OF LOCOMOTIVE

(If a heat-balance is desired, the form given in the table for Complete Steam Power Plants, Table 15, may be followed, using such modifications as may be needed to adapt it to locomotives).

CODE FOR ROAD TESTS

OBJECT AND PREPARATIONS

148 This code relates to road-tests in which the principal object is the determination of the coal and steam consumption of a locomotive per unit of power under the practical conditions of railroad service. The preparation for a road test are the same as those noted for the laboratory test, except for restrictions imposed by the regular operation of the locomotive, and in addition those required for data to be obtained on the road. See Appendix No. 34.

149 It is assumed that a dynamometer car is available for registering the amount of pull on the drawbar, in the manner referred to in Appendix No. 20.

150 To facilitate the work of the men who operate the indicators and read the instruments at the front end of the locomotive, and to protect them from wind and rain and jolting of the locomotive, a suitable housing or pilot box should be provided extending back to the cylinders and securely fastened to the bumper beam.

FUEL[1]

151 Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired, the fuel should be some kind that is regarded as a standard for the road on which the test is made, as noted in the Boiler Code, ¶ 35.

APPARATUS AND INSTRUMENTS

152 The apparatus and instruments required for a road test of a locomotive are:

(*a*) Scales for weighing coal and ashes located at the terminals, the coal being weighed preferably into sacks duly marked.
(*b*) Coal calorimeter.
(*c*) Water meter suitably calibrated for measuring the feed-water.
(*d*) Graduated scale attached to the water glass of the boiler, and a water glass with suitable scale attached to each corner of the cistern of the tender.
(*e*) Suitable levels or plumb-lines to show the inclination of the boiler.
(*f*) Pressure gages, draft gages, pyrometers and thermometers.
(*g*) Steam calorimeter.
(*h*) Steam engine indicators.
(*i*) Planimeter.
(*j*) Tachometer or other speed-measuring apparatus.
(*k*) Dynamometer car for determining the pull on the drawbar.

153 Directions for their use and calibration are given in ¶ 7 to 9, and ¶ 24 to 33, and in Appendix No. 34.

154 The steam used for auxiliary purposes other than the main engine, embracing air pump, train lighting, heating, etc., may be estimated from data obtained by testing them one by one either before or after the trial, using the water-glass method described in Appendix No. 3.

[1]See ¶ 147 in Code for Laboratory Tests for memoranda relating to oil fuel.

OPERATING CONDITIONS

155 In a road test the operating conditions are those pertaining to the regular service on the railroad, embracing principally the regular stops at stations, the usual speeds, and other conditions required for making schedule time, including those pertaining to the methods of firing, position of throttle valve, reverse lever, etc. If it is desired to make the test under other conditions, these should all be determined and maintained accordingly.

DURATION

156 The duration of a road test depends upon the length of the run between locomotive terminals. In fast passenger service the run should, if practicable, be at least 100 miles long. In service requiring frequent stops and in freight service the distance may be much shorter.

157 The length of time upon which the hourly rates of combustion and evaporation are based is the total time that the throttle valve is open, and not the elapsed time between the so-called starting and stopping times.

STARTING AND STOPPING

158 The fire having been thoroughly cleaned and the ashes and refuse removed from the ash-pan and smoke-box, burn down the fire as low as practicable before the locomotive leaves the round house, note its average thickness, the steam pressure, the water levels, the reading of the meter, and the time, and consider the latter the starting time of the boiler trial. Thereafter cover the fire with weighed coal and proceed with the measurements of coal, water, steam pressure, and feed temperature, which pertain to the boiler work, meanwhile building up the fire to its working thickness and condition, and finally taking the engine from the round house to the train and making ready for the schedule run. The period of time elapsing between the beginning of the boiler trial in the round house and the beginning of the schedule run, should be as short as the service will permit.

159 Just before the train starts note the amount of coal thus far fired and the amount of water fed. Note also the thickness of the fire-bed, the water levels, and steam pressure. Thereafter proceed with the schedule run. Maintain the fire at the working thickness throughout the run, using the rocking grates as often as need be, leaving at the end of the route the same amount of fire, as near as can be estimated, as when the train started; and observing the pressure, water levels, and meter readings.

160 Then take the locomotive to the round house or other terminal, burn down the fire and clean it, leaving as near as possible the same amount of live coal on the grate as that observed at the beginning of the boiler trial. Finally observe the average thickness, note the steam pressure, water levels, meter readings, and the time, and consider the latter the stopping time of the boiler trial. Then clean the ashes and refuse from the ash-pan and remove the cinders from the smoke-box.

RECORDS

161 The data should be recorded in the general manner pointed out in ¶ 15 to 18, bearing in mind the extremely fluctuating character of the load which often obtains, and the great variations in the class of service, whether passenger or freight. Readings of the instruments concerned, so far as they can be observed when the locomotive and train are in motion, should be taken every five minutes. Indicator diagrams should be obtained one after another as fast as practicable, say a set every three minutes; and at the same time the corresponding pressure, position of throttle valve and reverse lever, speed, and force of draft should be observed. The intervals of time here given may be lengthened when the service is such that the conditions are substantially uniform.

162 Special readings of the meter, water levels, and total number of sacks of coal fired should be taken at specified stopping and passing points. Careful observations should be made throughout the trial of the time of passing each mile-post, the time that the throttle valve is opened and closed, not only at each stop but also when it is necessary to shut off steam on down grades, and the time of arriving and leaving each station; also the length of time the safety valve, whistle, blower, train heating system, lighting system and other steam using apparatus are in operation. A record should be taken of the number of injector applications, and the overflow water should be measured or estimated and allowed for.

SAMPLING AND DRYING COAL

163 While the coal is being weighed into the sacks, or otherwise weighed, it should be systematically sampled, and its moisture determined in the manner pointed out in ¶ 24 to 26.

ASHES AND REFUSE

164 The ashes and refuse taken from the ash-pan and smoke-box at the close of the test should be weighed in a dry state, and a sample analyzed if desired for unburned carbon.

CALORIFIC TESTS OF COAL

165 The quality of the coal should be determined by a calorimeter test of the sample above referred to. Methods of making this test are described in ¶ 9n and in Appendix No. 12.

CALCULATION OF RESULTS

166 For methods of calculating the principal results, reference may be made to ¶ 58 of the Boiler Code and ¶ 77 of the Engine Code.

167 To determine the average drawbar pull, the dynamometer record should be averaged either by the use of a planimeter or by direct measurement at a sufficient number of points to give an equivalent result.

168 The maximum pull should be measured at the point of the record which shows the highest sustained pull, and not at the highest point which happens to be reached by the momentary fling of the marking pen.

DATA AND RESULTS

169 The data and results should be reported in accordance with the form (Table 18) given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

TABLE 18 DATA AND RESULTS OF ROAD TEST OF LOCOMOTIVE

Code of 1915

(1) Test of ....locomotive ....
To determine ....
Test conducted by ....

Dimensions, etc.

(2) Type and class of locomotive ....
(a) Number of locomotive..
(b) Driving wheels, number of pairs and diameter..........................
(c) Truck wheels (engine), number of pairs and diameter.....................
(d) Trailing wheels, number of pairs and diameter..........................
(e) Wheel base, driving wheel and total......................................
(f) Weight of locomotive and tender (including coal and water)...............lb.
(g) Weight of locomotive not including tender...............................lb.
(h) Number of cars, including dynamometer car...............................
(i) Weight of cars, including dynamometer car...........................tons
(j) Length of route, from———to———.............................miles
(k) Train haul, not including locomotive..............................ton-miles
(l) Total haul, including locomotive..................................ton-miles

(3) Type of boiler (and superheater, if any) ....
(a) Grate surface———wide———long.................................sq. ft.
(b) Percentage of air openings in grates...............................per cent
(c) Area of air inlets to ash-pan...sq. ft.
(d) Diameter of shell..in.
(e) Number and diameter of boiler tubes.......................................
(f) Length of boiler tubes..
(g) Number and diameter of superheater tubes...................................
(h) Length of superheater tubes...
(i) Heating surface, boiler...sq. ft.
(j) Heating surface, superheater..sq. ft.

(4) Diameter of cylinders and stroke of pistons........................
(a) Diameter of piston rods and tail rods.....................................
(b) Type of valves..
(c) Area of valve ports...
(d) Valve travel (maximum)..
(e) Steam lap...
(f) Exhaust lap...
(g) Clearance, in per cent of piston displacement........................per cent
(h) Dimensions of air pump..
(i) Dimensions of other auxiliaries...

(5) Exhaust nozzles, number and area ....

Date, Duration, etc.

(6) Date ....
(7) Duration of main trial, (length of time throttle valve is open between terminals) ....hr.
(a) Duration of boiler trial...hr.
(b) Actual time between terminals...hr
(c) Schedule time between terminals...hr

(*d*) Number of stops..
(*e*) Time consumed in stops..hr.
(*f*) Running time (Item *b*—Item *e*)..hr.

(8) Kind and size of coal ....

Average Pressures, Temperatures, etc. Main Trial

(9) Boiler pressure by gage..lb. per sq. in.

(*a*) Steam chest pressure..lb. per sq. in.
(*b*) Barometric pressure ....in. of mercury

(10) Temperature of steam at chest, if superheated ....deg.

(*a*) Normal temperature of saturated steam at chest pressure..deg.
(*b*) Temperature corresponding to pressure in exhaust pipe..deg.
(*c*) Temperature of air..deg.
(*d*) Weather, wind, etc..

(11) Temperature of feed water in tank ....deg.

(*a*) Temperature of feed water entering and leaving heater, if any......deg.

(12) Temperature of smoke box ....deg.
(13) Force of draft in smoke box, front of diaphragm..........in. of water

(*a*) Force of draft in smoke box, back of diaphragm..in. of water
(*b*) Force of draft in fire-box..in. of water
(*c*) Force of draft in ash-pan..in. of water

Quality of Steam

(14) Percentage of moisture in steam or number of degrees of superheating ....per cent or deg.

(*a*) Factor of correction for quality of steam..

Boiler Trial—Principal Data and Results

(15) Total coal, as fired, on boiler trial, entire run ....lb.

(*a*) Total ash, clinkers, and refuse from ash-pan..lb.
(*b*) Total cinders in smoke-box..lb.
(*c*) Sum of Items *a* and *b*..lb.
(*d*) Percentage of Item *c* to Item 15..per cent

(16) Total water evaporated on boiler trial, entire run ....lb.
(17) Water per lb. of coal as fired ....lb.

MAIN TRIAL, Total Quantities

(18) Total water fed on main trial (corrected for leakages)..lb.

(*a*) Total water evaporated, corrected for quality of steam (Item 18 × Item 14*a*)..lb.

(19) Total water evaporated, corrected for moisture in steam, (dry steam consumed by engine)..lb.

(*a*) Factor of evaporation ....
(*b*) Total equivalent evaporation from and at 212 degrees (Item 18*a* × Item 19*a*)..lb.

(20) Total computed coal as fired during main trial (Item 18 ÷ Item 17) ....lb.

(a) Percentage of moisture in coal .... per cent

(b) Total computed dry coal during main trial $\left(\text{Item } 20 \times \left(1 - \frac{\text{Item } 20a}{100}\right)\right)$ ...... lb.

(c) Percentage of ash in dry coal, by analysis .... per cent

(d) Total combustible fired on main trial (Item 20b × [1—(Item 20c ÷ 100)]) .... lb.

(e) Total combustible burned on main trial (Item 20b × [1—(Item 15a ÷ 100)]) ... lb.

(f) Coal actually fired during main trial .... lb.

Calorific Value and Analysis of Coal

(21) B.t.u. per lb. of coal as fired, by calorimeter .... B.t.u.

(a) B.t.u. per lb. of dry coal, by calorimeter .... B.t.u.

(b) B.t.u. per lb. of combustible, by calorimeter .... B.t.u.

(c) Moisture in coal as fired .... per cent

(d) Volatile matter .... per cent

(e) Fixed carbon .... per cent

(f) Ash .... per cent

(g) Sulphur .... per cent

(h) Ultimate analysis, if made ....

Hourly Quantities and Rates, Main Trial

(22) Water evaporated per hour (Item 19 ÷ Item 7) (dry steam consumed by engine) .... lb.

(a) Equivalent evaporation per hour from and at 212 deg. .... lb.

(b) Equivalent evaporation per hour and at 212 deg. per sq. ft. water-heating surface .... lb.

(c) Estimated weight of dry steam used per hour by air pump and other auxiliaries .... lb.

(d) Percentage of Item 22 c to Item 22 .... per cent

(e) Weight of dry steam used per hour by engine alone (Item 22—Item 22c) .... lb.

(23) Coal as fired per hour (Item 20 ÷ Item 7) .... lb.

(a) Dry coal per hour (Item 20b ÷ Item 7) .... lb.

(b) Dry coal per hour per sq. ft. of grate .... lb.

Indicator Diagrams

(24) Mean effective pressure, each cylinder .... lb. per sq. in.

(a) Commercial cut-off .... per cent

(b) Initial pressure above atmosphere .... lb. per sq. in.

(c) Back pressure at lowest point above atmosphere .... lb. per sq. in.

(d) Steam accounted for per i.h.p-hr. at point near cut-off .... lb.

(e) Steam accounted for per i.h.p-hr. at point near release .... lb.

Speed

(25) Revolutions per minute .... r.p.m.

(a) Piston speed in ft. per minute .... ft.

(b) Average train speed in miles per hour .... miles

(c) Maximum train speed in miles per hour .... miles

Power

(26) Indicated horsepower (average) .... i.h.p.

(a) Indicated horsepower (maximum) .... i.h.p.

(27) Drawbar pull (average)..lb.

(a) Drawbar pull (maximum)..lb.

(28) Dynamometer horsepower ....h.p.

Economy Results, Main Trial

(29) Coal as fired per i.h.p-hr. (Item 23 ÷ Item 26)....................lb.

(a) Dry coal per i.h.p-hr. (Item 28a ÷ Item 26)......................lb.
(b) Dry coal per dynamometer h.p-hr...................................lb.
(c) Coal as fired per ton-mile..lb.
(d) Coal as fired per car-mile..lb.

(30) Heat units in coal as fired per i.h.p-hr.......................B.t.u.

(31) Dry steam per i.h.p-hr..lb.

(a) Dry steam per dynamometer h.p-hr..................................lb.
(b) Dry steam per ton-mile..lb.
(c) Dry steam per car-mile..lb.

(32) Heat units consumed by engine per i.h.p-hr....................B.t.u.

(a) Heat units consumed by engine per dynamometer h.p-hr..........B.t.u.

(33) Water evaporated per lb. of dry coal (entire run)................lb.

(a) Equivalent evaporation from and at 212 deg. per lb. dry coal..........lb.

Efficiency, Main Trial

(34) Thermal efficiency of locomotive referred to i.h.p. [(2546.5 ÷ Item 30) × 100]..per cent

(35) Thermal efficiency of engine alone referred to i.h.p. [(2546.5 ÷ Item 32) × 100].....................................per cent

(a) Thermal efficiency of engine alone referred to dynamometer h.p. [(2546.5 ÷ Item 32a) ×100]...per cent

(36) Efficiency of boiler [(Item 33a × 970.4) ÷ Item 21a × 100]..per cent

PART XI

RULES FOR CONDUCTING TESTS OF GAS PRODUCERS

OBJECT AND PREPARATIONS

170 Determine the object, take the dimensions, note the physical condition of the producer and its appurtenances, install the testing appliances, etc., as pointed out in the general instructions given in ¶ 1 to 6 and ¶ 10 to 13, and make preparations for the test accordingly.

FUEL[1]

171 Determine the character of the fuel to be used. If an untried fuel is selected and a test-producer is available, make a preliminary trial of the fuel in this apparatus and ascertain its working characteristics and the proper methods of handling it.

172 In tests of maximum efficiency and capacity of a producer for conparison with other producers, the fuel should be some kind of coal which is commercially regarded as a standard for such use in the locality where the test is made. The coal selected for such tests should be the best of its class and free from unusual slag-forming impurities.

173 The size of the coal should be determined by screening a sample, using the screens referred to in ¶ 9 f, or in Appendix No. 7.

APPARATUS AND INSTRUMENTS

174 The apparatus and instruments required for producer tests are:

(*a*) Platform scales for weighing coal and ashes.
(*b*) Coal calorimeter.
(*c*) Gas calorimeter.
(*d*) Gas analyzing apparatus and appliances for determining tar and soot.
(*e*) Gas meter, venturi meter, pitot tube, or other suitable apparatus for measuring the gas output.

[1] This code is primarily intended for producers using coal. If other fuel, such as wood or oil, is burned, the rules may be modified accordingly.

(*f*) Manometers or pressure gages.
(*g*) Water meters for measuring feed and scrubber water, and steam meters for measuring steam used by the apparatus.
(*h*) Thermometers.

In addition to these instruments a continuous indicating calorimeter showing the quality of the gas furnishes a valuable adjunct both for testing and operating purposes.

175 Full directions regarding the use and calibration of the above noted appliances are given in ¶ 7 to 9, and in various appendices there referred to.

176 The location of the pitot tube, if used, should be in the delivery pipe at a point near the producer or just beyond the scrubber, or at both points, according to the use made of the gas, either for fuel or power, and other requirements.

OPERATING CONDITIONS

177 Determine what the operating conditions should be to conform to the object in view, as pointed out in ¶ 14, and see that they prevail throughout the trial.

DURATION

178 The duration of both efficiency and capacity tests of a producer, with the exceptions noted below, should be such that the total consumption of fuel is at least ten times the weight of the fuel contained in the producer when in normal operation, estimating this weight in the case of coal at 45 lb. per cu. ft.

179 In cases which require the fuel bed to be entirely removed and rebuilt at regular intervals, and in producers where a complete cleaning and renewal occurs before the total consumption above stipulated has been reached, the duration should be that of the regular commercial operating cycle, or the time elapsing between two successive renewals of the fuel bed.

STARTING AND STOPPING

180 The conditions regarding the temperature of the producer and its contents, and the quantity and quality of the latter, should be as nearly as possible the same at the end as at the beginning of the trial. To secure the desired equality of conditions, the starting and stopping should occur at times of regular cleanings, and they should be preceded for a period of not less than 10 hours by the same regular working conditions as are intended to characterize the test as a whole. The operations of starting and stopping should then be carried on as follows:

180a Continuous Producers with Grate and No Ash Bed

Remove the ash and clinkers from the grate and the lower part of the furnace space, taking care that the crust or closely-united layer which supports the coal above is not unduly disturbed. Then break open the crust and allow the mass to drop into the space left vacant. Introduce a poker rod through the poke holes in the upper head and stir up the coal within, thereby causing it to settle and fill the remaining spaces. As a final step, quickly replenish the producer with coal to the working depth, fill the hopper level-full, take the time, and consider this the starting time. Then clean the ash-pit, and thereafter proceed with the regular work of the test, using weighed coal.

When the time arrives for bringing the trial to a close, the cleaning operations described above are repeated, ending with filling the hopper, taking the time, and considering this the stopping time; finally hauling the ashes and refuse from the ash-pit.

180b Continuous Producers with Supporting Ash Beds

Remove the ashes until the top of the ashbed is lowered to the normal working point. Introduce the poker-rod and break down any bridge or crust that may have formed, at the same time closing up the channels that run through the fuel bed, thereby making the bed homogeneous. Then replenish the producer with coal to the working depth, fill the hopper level-full, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test, using weighed coal.

When the time approaches for closing the test, the operations above described are repeated, ending with replenishing the producer and filling the hopper with weighed coal, taking the time, and considering this the stopping time. The ashes and refuse finally removed are to be dried before weighing, unless already dried, or a sample should be taken and the moisture, as determined therefrom, allowed for.

180c Intermittent Producers

Thoroughly clean the producer of its entire contents. Introduce a weighed supply of coke or coal, start the fire, and build up the fuel bed to its working condition, using weighed coal. When this point is reached, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test.

When the time approaches for closing the test, burn the fuel bed as low as practicable to prepare for cleaning, note the time, and consider this the stopping time. Then completely empty the producer, quench the fire remaining in the live coals, separate and weigh the coke and ash, and deduct the weight of the former from that of the coke as charged. Finally dry the ash and refuse, or take a sample and allow for the moisture determined therefrom.

NOTE:—Some idea of the depth of the ash bed may be gained by the following method: Insert a long poker-rod through the fuel bed, and determine how many minutes it takes to become red hot. Cool it, and insert it again through two or more poke holes successively for the determined time, cooling it after each trial. Measure the distance from the top of the producer to the lower end of the red portion in each case, and subtract the average of the distances thus found from the total depth of the producer. The result gives the approximate depth of the ash bed. The length of the red portion furnishes also some idea of the depth of the zone of burning fuel.

The distance from the top of the producer to the surface of the fuel bed may be found by direct measurement with the poker-rod, noting by sense of touch when the end of the rod reaches the fuel.

RECORDS

SAMPLING AND DRYING COAL

ASHES AND REFUSE

CALORIFIC TESTS AND ANALYSES OF COAL

181 The directions pertaining to the above divisions of the subject are practically the same as those given under the corresponding headings in the Boiler Code, and reference may be made to ¶ 49 to 53, of that Code for these directions.

CALORIFIC TESTS AND ANALYSES OF GAS OUTPUT

182 The quality of the gas should be determined by calorific tests and analyses, continuous samples for this purpose being taken from the delivery pipe at a point near the producer and at other points as may be needed.

183 The calorific test should be made with the Junker calorimeter, described in Appendix No. 13, or its equivalent. Unless otherwise required the "higher value" should be employed in calculating the results of the test.* For an approximate determination of the composition of the gas, the Orsat apparatus may be used, and for complete determination, the Hempel apparatus or its equivalent. Both of the two named are described in Appendix No. 15. The frequency with which these determinations should be made depends on the uniformity of the output, but the intervals, where practicable, should not be more than one-half hour, the time taken for collecting each sample being not less than one-half hour.

*If the lower value is used in place of the higher value the fact should be so stated.

CALCULATION OF RESULTS

184a Total Volume of Gas Delivered

The volume of gas in the case of pitot-tube measurement is determined by multiplying the area of the delivery pipe in sq. ft. at the tube by the velocity of the gas in ft. per minute, and the product by the duration of the trial in minutes.

The equivalent volume at atmospheric pressure (30 in. barometer) and temperature of 60 deg. fahr., is obtained by multiplying the measured volume by the absolute pressure of the gas in lb. per sq. in. (gage pressure plus 14.7) and by the constant 35.3 (520 ÷ 14.7), and dividing the product by the absolute temperature of the gas (temperature by thermometer plus 460 deg.).

The gas as it leaves the scrubber is saturated with water vapor at the pressure due to the temperature, and this pressure is to be deducted from the absolute pressure as observed, to obtain the net pressure of dry gas. The volume of dry gas in that case is obtained by the following formula:

$$V = V_e \times \frac{P - P_w}{P_a} \times \frac{520}{t + 460}$$

in which V = equivalent volume of dry gas at 60 deg. and 30 in. leaving scrubber

V_e = observed volume of gas leaving the scrubber

P = absolute pressure at the point where the volume is measured

P_w = pressure of saturated water vapor at temperature

P_a = atmospheric pressure (14.7 lb. per sq. in. or 30 in. of mercury)

t = observed temperature of gas leaving scrubber

P and P_W are to be taken in the same units as P_a.

The volume of dry gas delivered by a gas producer can be obtained by calculation from the analyses of the coal and of the dry gas, by the following method:

$$\text{Dry gas per lb. carbon} = \frac{11\ CO_2 + 8\ O + 7\ (CO + N + C_2H_4) + 4\ CH_4 + 0.5H}{3\ (CO_2 + CO + CH_4 + 2\ C_2H_4)}$$

in which CO_2, O, CO, H, CH_4, C_2H_4, and N are percentages of the dry gas by volume. Multiplying the lb. of gas per lb. of carbon by the percentage of carbon in the coal and dividing by 100 gives the weight of dry gas per lb. of coal, and the volume of gas per lb. of coal is found by dividing this weight by the weight of one cubic foot of the gas calculated from the analysis as shown in Paragraph 184c below.

The weight of carbon per 100 cu. ft. of gas can be found directly from the analysis by multiplying the per cent by volume of each gas containing carbon by the pounds of carbon in 1 cu. ft. of the gas as given below:

$CO + CO_2$, 0.03173; CH_4, 0.03176; C_2H_4, 0.06345
and adding the results together.

184b Net Volume of Dry Gas Delivered

The net volume of dry gas delivered is found by subtracting from the total volume the volume of gas that would be required for furnishing steam or power for any purpose concerned in the operation of the producer and its auxiliaries.

184c Weight of Gas

The weight of dry gas delivered is found by multiplying the volume in cu. ft. reduced to 60 deg. and 30 in. pressure, corrected for moisture, by the weight per cubic foot, which is found by multiplying the percentage of each component gas as found by analysis by its weight in lb. per cu. ft. at 60 deg. and 30 in., as given in the following table, (calculated from Landolt and Börnstein's figures at 32 deg.), and dividing the sum of the products by 100.

H	0.005335	CO_2	0.116333
O	0.084608	CH_4	0.042434
N	0.074082	C_2H_4	0.074113
CO	0.074029	SO_2	0.169400

184d Calorific Value of Gas

The calorific value of the dry gas per cu. ft. is obtained by means of the Junker calorimeter, (Appendix No. 13) but it may also be obtained from the analysis by multiplying the percentage of each combustible constituent gas by its heating value per cu. ft. at 60 deg. and 30 in. as given below:

	B.t.u. per cu. ft.		B.t.u. per cu. ft.
CO	317.8	CH_4	1002.1
H	329.9	C_2H_4	1595

These figures are the product of the above values of weight of gas per cu. ft. by the heating value of one pound of gas according to Thomsen.

184e Moisture in Gas

The moisture in the gas leaving producer, is found by passing a measured sample of the gas through a chloride of calcium tube and weighing the amount of moisture absorbed.

The moisture in the gas leaving the scrubber is best found by calculation assuming that the gas is saturated with moisture. The calculation is made in the manner pointed out in Paragraph 184 a.

184f Percentage of Tar and Soot in Gas

The percentage of tar and soot is found by comparing the total weight determined, including that collected from the various tar drips with the total weight of dry fuel used.

184g Efficiency

The efficiency is the relation between the calorific value of the dry gas per lb. of dry fuel charged or combustible, and the calorific value of 1 lb. of dry fuel or combustible. The former is ascertained by multiplying the B.t.u. per cu. ft. of dry gas as determined by the calorimeter test (higher value) by the cu. ft. of dry gas delivered, and dividing the product by the total weight of dry fuel charged or combustible.

The "combustible" is determined by subtracting from the weight of coal charged the moisture in the coal and the weight of ash refuse and unburned coal withdrawn from the producer or ash-pit during the progress of the trial. The "combustible" used for determining the calorific value is the weight of the coal less the moisture and ash found by analysis.

The efficiency of "conversion and cleaning" or "gross efficiency" in the above calculation is found by using the total volume of gas delivered. The "efficiency of the plant" or "net efficiency" is found by using the net volume of gas delivered.

184h Heat Balance

The various quantities showing the distribution of heat in the heat balance given in Table 19, are computed in the following manner:

The calorific value of the dry gas is found by multiplying the cubic feet of gas at 60 deg. and 30 in. per lb. of dry coal by the calorific value of 1 cu. ft. of gas at 60 deg. and 30 in. (higher value).

The sensible heat in the dry gas is found by multiplying the weight of gas per pound of coal by the mean specific heat of the gas and by its temperature measured above 60 deg.

The heat carried away by the scrubber is obtained by multiplying the weight of water fed to the scrubber by the number of degrees rise of temperature, and dividing the product by the total weight of dry coal consumed.

The heat contained in the moisture leaving the producer is found by multiplying the total weight of dry gas per lb. of dry coal by the proportion of moisture in the gas and by the total heat of 1 lb. of superheated steam at the temperature of the gas leaving the producer reckoned from 60 deg.

The loss due to combustible matter in the ash is found by multiplying the proportion that this combustible bears to the whole amount of dry coal by 14,600 B.t.u.

DATA AND RESULTS

185 The data and results should be reported in accordance with the form given herewith (Table 19), adding lines for data not provided for or omitting those not required as may conform to the object in view. If a shorter form is desired, the items in fine print designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

186 If a preliminary trial of the fuel is made in a test-producer, add to the table the general results obtained.

CHART

187 In trials having for an object the determination and exposition of the complete performance from beginning to end, the entire log of readings and data should be plotted on a chart and represented graphically.

TABLE 19 DATA AND RESULTS OF GAS PRODUCER TEST

Code of 1915

(1) Test of ....producer located at ....
To determine ....
Test conducted by ....

Dimensions, etc.

(2) Outside diameter and height of producer ....ft.
(3) Inside diameter of producer ....ft.
(4) Area of grate diameter ....sq. ft.
(a) Percentage of air space in grate ....per cent
(b) Area of blast inlet ....sq. ft.
(c) Area of exit flue ....sq. ft.
(5) Area of fuel bed (at maximum diameter) ....sq. ft.
(a) Area of water heating surface in vaporizer ....sq. ft.
(6) Rated capacity of producer in lb. of coal per hour ....lb.

Date, Duration, etc.

(7) Date ....
(8) Duration ....
(9) Kind and size of coal* ....

Average Pressures, Temperatures, etc.

(10) Steam pressure in vaporizer by gage....................lb. per sq. in.
(11) Gas pressure in main at point where gas is measured.......in. of water
(a) Pressure at top of producer............................in. of water
(b) Pressure beyond scrubber..............................in. of water
(c) Pressure beyond purifier...............................in. of water
(12) Force of blast or draft in ashpit, or bottom of producer....in. of water

* If other fuel than coal is used the lines may be changed to read accordingly.

(a) Barometric pressure ....in. of mercury
(b) Relative humidity of air..................................per cent
(c) Depth of fuel bed..
(d) Intervals between cleaning ....
(e) Intervals between poking..

(13) Temperature of feedwater entering vaporizer ....deg.

(a) Temperature of gas in exit flue at producer...........................deg.

(14) Temperature of gas in main at point where gas is measured.......deg.

(a) Temperature of air in room..deg.
(b) Temperature of water entering scrubber..............................deg.
(c) Temperature of water leaving scrubber...............................deg.

TOTAL QUANTITIES

(15) Weight of coal charged* ....lb.
(16) Percentage of moisture in coal.............................per cent
(17) Total weight of dry coal..lb.
(18) Total ash and refuse..lb.
(19) Percentage of ash and refuse in dry coal...................per cent
(20) Total combustible (Item 17 — Item 18)...........................lb.
(21) Total number of cu. ft. of gas delivered....................cu. ft.

(a) Total weight of dry gas delivered.....................................lb.

(22) Moisture in gas leaving producer, mixed with 1 lb. of dry gas......lb.
(23) Moisture in gas leaving scrubber, mixed with 1 lb. of dry gas......lb.
(24) Equivalent cu. ft. of dry gas at temperature of 60 deg. and pressure of atmosphere of 30 in...................................cu. ft.
(25) Net cu. ft. of dry gas at 60 deg. and 30 in...................cu. ft.
(26) Percentage of tar and soot in gas referred to total fuel.......per cent
(27) Total water fed to vaporizer....................................lb.

(a) Total water evaporated in vaporizer...................................lb.
(b) Total weight of steam supplied to producer............................lb.
(c) Total weight of water fed to scrubber.................................lb.

HOURLY QUANTITIES AND RATES

(28) Dry coal per hour...lb.

(a) Dry coal per hour per sq. ft. of main fuel bed........................lb.

(29) Equivalent cu. ft. of dry gas per hour at 60 deg. and 30 in. (Item 24 ÷ Item 8)...cu. ft.
(30) Net cu. ft. of dry gas delivered per hour at 60 deg. and 30 in.....cu. ft.
(31) Water fed per hour to vaporizer.................................lb.

(a) Water evaporated per hour in vaporizer................................lb
(b) Steam supplied to producer per hour...................................lb.

(32) Water fed to scrubber per hour..................................lb.

PROXIMATE ANALYSIS OF COAL

(33) Fixed carbon ....per cent
(34) Volatile matter ....per cent

* Corrected for difference in estimated quantity of coal in producer at beginning and end of test.

(35) Moisture .... per cent
(36) Ash .... per cent
(37) Sulphur, separately determined .... per cent

100 per cent

Ultimate Analysis of Dry Coal

(38) Carbon (C) .... per cent
(39) Hydrogen (H) .... per cent
(40) Oxygen (O) .... per cent
(41) Nitrogen (N) .... per cent
(42) Sulphur (S) .... per cent
(43) Ash .... per cent

100 per cent

Analysis of Ash and Refuse

(44) Carbon .... per cent
(45) Earthy matter .... per cent
(46) Fusing temperature of ash .... deg.
(47) Nature and texture of ash ....

Analysis of Gas by Volume

(48) Carbon dioxide (CO_2) .... per cent
(49) Carbon monoxide (CO) .... per cent
(50) Oxygen (O) .... per cent
(51) Hydrogen (H) .... per cent
(52) Marsh Gas (CH_4) .... per cent
(53) Olefiant gas (C_2H_4) .... per cent
(*a*) Sulphur dioxide (SO_2) .... per cent
(*b*) Hydrogen sulphide (H_2S) .... per cent
(*c*) Nitrogen (N) by difference .... per cent

100 per cent

(*d*) Total combustible gases .... per cent

Calorific Values by Calorimeter

(54) Calorific value of 1 lb. of dry coal .... B.t.u.
(55) Calorific value of 1 lb. of combustible .... B.t.u.
(56) Calorific value of 1 cu. ft. of dry gas at 60 deg. and 30 in. (higher value) .... B.t.u.

Economy Results

(57) Equivalent cu. ft. of dry gas at 60 deg. and 30 in. per lb. of dry coal .... cu. ft.
(*a*) Equivalent cu. ft. of dry gas at 60 deg. and 30 in. per lb. of combustible .... cu. ft.

(58) Net cu. ft. of dry gas at 60 deg. and 30 in. per lb. of dry coal .... cu. ft.
(*a*) Net cu. ft. of dry gas at 60 deg. and 30 in. per lb. of combustible .... cu ft.

Efficiency

(59) Gross efficiency of producer, based on dry coal..............per cent

(a) Net efficiency of producer, based on dry coal........................ per cent

(60) Gross efficiency of producer, based on combustible............per cent

(a) Net efficiency of producer, based on combustible......................per cent

Cost of Coal

(61) Cost of coal per ton of————lb. delivered.....................dollars

(62) Cost of coal required for producing 1000 net cu. ft. of gas at 60 deg. and 30 in. ....dollars

(a) Cost of coal for producing 1,000,000 B.t.u.............................dollars

Heat Balance Based on 1 Lb. of Dry Coal

	B.t.u.	per cent
(63) Total calorific value of 1 lb. of dry coal, same as Item 54................................	....	....
(a) Calorific value of dry gas..................	....	....
(b) Sensible heat in hot dry gas above 60 deg. fahr.	....	....
(c) Total heat of moisture in gas above 60 deg.	....	....
(d) Heat lost in scrubber......................	....	....
(e) Heat lost by combustible in ash..............	....	....
(f) Heat lost by radiation, and unaccounted for (difference between the sum of items a, b, c, d, e and item 63)	....	....

Note:—If steam is supplied to the producer from an outside source the data and results should be modified accordingly.

PART XII

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES

OBJECT AND PREPARATIONS

188 Determine the object, take the dimensions, note the physical condition of the engine and its appurtenances, install the testing appliances, etc., as explained in the general instructions given in ¶ 1 to 6 and ¶ 10 to 13, and make preparations for the test accordingly.

APPARATUS AND INSTRUMENTS

189 The apparatus and instruments required for performance tests of gas and oil engines are:

(*a*) Tanks and platform scales for weighing oil.
(*b*) Calorimeter for determining the heat of combustion of oil.
(*c*) Gas meter or other apparatus for measuring gas.
(*d*) Gas calorimeter.
(*e*) Pressure gages and thermometers.
(*f*) Gas engine indicators.
(*g*) Planimeter.
(*h*) Tachometer or other speed-measuring apparatus.
(*i*) Gas analyzing apparatus.
(*j*) Water meter for measuring jacket water.
(*k*) Friction brake or dynamometer.

190 Full directions regarding the use and calibration of these appliances are given in ¶ 7 to 9, and in various appendices there referred to.

OPERATING CONDITIONS

191 Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out in Part I, ¶ 14.

DURATION

192 The test of a gas or oil engine with substantially constant load should be continued for such time as may be necessary to obtain a number of successive records covering periods of half an hour or less during which the results are found to be uniform. In such cases a duration of three to five hours is sufficient for all practical purposes.

STARTING AND STOPPING

193 The engine having been set to work under the prescribed conditions, the test is begun at a certain predetermined time by commencing to weigh the oil, or measure the gas, as the case may be, and take other data concerned; after which the regular measurements and observations are carried forward until the end. When the stopping time arrives the test is closed by simply taking the final readings.

RECORDS

194 The general data should be taken and recorded in the same manner as that described in the Steam Engine Code, in ¶ 75 and 76, to which reference may be made.

CALORIFIC TESTS AND ANALYSES

195 The quality of the oil or gas should be determined by calorific tests and analyses made on representative samples.

196 Directions for these tests or analyses are given in ¶ 9 n to 9 q under the headings Coal Calorimeters, Gas Calorimeters, Coal Analysis Apparatus, and Gas Analysis Apparatus, and in Appendices Nos. 12, 13, 14, and 15 there referred to. See also ¶ 184d and ¶ 184e.

CALCULATION OF RESULTS

197a Volume of Gas at 60 deg. and 30 in.

The equivalent volume at a temperature of 60 deg. and at atmospheric pressure of 30 in. is obtained in the manner pointed out in ¶ 184a.

197b Heat Consumption.

The number of heat units consumed by the engine is found by multiplying the heat units per lb. of oil or per cu. ft. of gas (higher value), as determined by calorimeter test, by the total weight of oil in lb. or volume of dry gas in cu. ft. consumed.

197c Horsepower and Efficiency

The indicated horsepower, brake horsepower, and efficiency are computed by the same methods as those explained in the Steam Engine Code, in ¶ 77, to which reference may be made.

197d Heat Balance

The various quantities showing the distribution of heat in the heat balance given in Table 20 are computed in the following manner:

The heat converted into work per i.h.p-hr. (2546.5 B.t.u.) is found by dividing the work representing 1 h.p., or 1,980,000 ft-lb., per hour by the number of ft-lb. representing 1 B.t.u., or 777.5.

The heat rejected in the cooling water is obtained by multiplying the weight of water supplied by the number of degrees rise of temperature, and dividing the product by the indicated horsepower.

The heat rejected in the dry exhaust gases per i.h.p-hr. is found by multiplying the weight of these gases per i.h.p-hr. by the sensible heat of the gas reckoned from the temperature of the air in the room and by its specific heat. The weight of the dry exhaust gases per i.h.p-hr. is the product of the weight of fuel per i.h.p-hr. by the weight of the dry gases per lb. of fuel. The latter is the product of the proportion of carbon in 1 lb. of fuel by the weight of the dry gases per lb. of carbon, which may be found by the formula

$$\frac{11\ CO_2+8\ O+7\ (CO+N)}{3\ (CO_2+CO)}$$

in which CO_2, O, CO, and N are percentages of the dry exhaust gases by volume.

When the weight of air supplied per lb. of fuel is determined the weight of dry gas per pound of fuel may be found by the formula

$$1 + \text{lb. air per lb. fuel} - 9H$$

in which H is the proportion of hydrogen in 1 lb. of fuel.

The heat lost in the moisture formed by the burning of hydrogen in the fuel gas is found by multiplying the total heat of 1 lb. of superheated steam at the temperature of the exhaust gases, reckoning from the temperature of the air in the room, by the proportion of the hydrogen in the fuel as determined from the analysis, and multiplying the result by 9.

The heat lost in superheating the moisture contained in the gas and air is determined by multiplying the difference between the temperature of the exhaust gases and that of the gas and air by the average specific heat of superheated steam for the range of temperature and pressure.

The heat lost through incomplete combustion is obtained by analyzing the exhaust gases and computing the heat of the unburned products which would have been produced by their combustion.

The above rules do not apply to engines with hit-and-miss governors.

DATA AND RESULTS

198 The data and results should be reported in accordance with the form given herewith (Table 20), adding lines for data not provided for, or omitting those not required, as may conform to the object in view.[1] If a shorter form is desired, items given in fine print and designated by letters of the alphabet may be omitted. Unless otherwise indicated, the items should be the averages of the data.

TABLE 20 DATA AND RESULTS OF GAS OR OIL ENGINE TEST

Code of 1915

(1) Test of ....engine, located at....................
To determine ....
Test conducted by ....

Dimensions, etc.

(2) Type of engine, whether oil or gas ....
(3) Class of engine, (mill, marine, motor for vehicle, pumping, or other)....
(a) Number of strokes of piston for one cycle, and class of cycle..................
(b) Method of ignition..
(c) Single or double acting..
(d) Arrangement of cylinders..
(e) Vertical or horizontal..

(4) Rated power ....h.p.
(a) Name of builder..

(5) Number and diameter of working cylinders.......................in.
(a) Number and diameter of compression cylinders....................in.
(b) Diameter of piston rods..in.

(6) Stroke of pistons..ft.
(a) Compression space referred to piston displacement....................per cent
(b) Stroke of compression piston..ft.
(c) H.p. constant for 1 lb. m.e.p. and 1 r.p.m..............................h.p.

Date, Duration, etc.

(7) Date ....
(8) Duration ....hr.
(9) Kind of oil or gas..
(a) Physical properties of oil (specific gravity, burning point, flashing point)........

Average Pressure and Temperature

(10) Pressure of gas near meter..........................in. of mercury
(a) Barometric pressure ....in. of mercury

(11) Temperature of gas near meter ....deg.

[1] See note Table 9 for reference to engines driving electric generators and other machinery.

(a) Temperature of cooling water, inlet....deg.
(b) Temperature of cooling water, outlet....deg.
(c) Temperature of air by dry-bulb thermometer....deg.
(d) Temperature of air by wet-bulb thermometer....deg.
(e) Temperature of exhaust gases at cylinder....deg.

Total Quantities

(12) Gas or oil consumed....cu. ft., lb.
(13) Moisture in gas, in per cent by weight, referred to dry gas....per cent
(14) Equivalent dry gas at 60 deg. and 30 in....cu. ft.
(a) Air supplied....cu. ft.
(15) Cooling water supplied to jackets....lb.
(a) Water or steam fed to cylinder....lb.
(16) Calorific value of oil per lb., or of dry gas per cu. ft. at 60 deg. and 30 in. by calorimeter test (higher value)....B.t.u.

Hourly Quantities

(17) Gas or oil consumed per hour....cu. ft., lb.
(18) Equivalent dry gas per hour at 60 deg. and 30 in....cu. ft.
(19) Cooling water supplied per hour....lb.
(20) Heat units consumed per hour (Item 16 × Item 18)....B.t.u.

Analysis of Oil

(21) Carbon (C)....per cent
(22) Hydrogen (H)....per cent
(23) Oxygen (O)....per cent
(24) Sulphur (S)....per cent
(a) Moisture....per cent
(b) Result of fractional distillations....

Analysis of Fuel Gas by Volume

(25) Carbon dioxide (CO_2)....per cent
(26) Carbon monoxide (CO)....per cent
(27) Oxygen (O)....per cent
(28) Hydrogen (H)....per cent
(29) Marsh gas (CH_4)....per cent
(30) Heavy hydrocarbon $C_n H_m$....per cent
(a) Sulphur dioxide (SO_2)....per cent
(b) Hydrogen sulphide (H_2S)....per cent
(c) Nitrogen (N) by difference....per cent

Analysis of Exhaust Gases by Volume

(31) Carbon dioxide (CO_2)....per cent
(32) Carbon monoxide (CO)....per cent
(33) Oxygen (O)....per cent
(34) Nitrogen (N)....per cent

Indicator Diagrams

(35) Pressure above atmosphere............................lb. per sq. in.

- (a) Maximum pressure ....lb. per sq. in.
- (b) Pressure at beginning of stroke.........................lb. per sq. in.
- (c) Pressure at end of expansion...........................lb. per sq. in.
- (d) Exhaust pressure at lowest point.......................lb. per sq. in.

(36) Mean effective pressure..............................lb. per sq. in.

Speed

(37) Revolutions per minute....................................r.p.m.

(38) Average number of explosions or firing strokes per minute............

- (a) Variation of speed between no load and full load................r.p.m.
- (b) Momentary fluctuation of speed on suddenly changing from full load to half load...r.p.m.

Power

(39) Indicated horsepower ....i.h.p.

(40) Brake horsepower ....br. h.p.

(41) Friction horsepower by difference (Item 39 — Item 40)*.....fr. h.p.

- (a) Friction horsepower by friction diagrams.......................fr. h.p.

(42) Percentage of indicated horsepower lost in friction Item 41....per cent

Economy Results

(43) Heat units consumed by engine per i.h.p. per hour†..............B.t.u.

(44) Heat units consumed by engine per br-h.p......................B.t.u.

(45) Dry gas at 60 deg. and 30 in. consumed per i.h.p-hr........lb., cu. ft.

(46) Pounds of oil or cubic feet of dry gas per br-h.p-hr..........lb., cu. ft.

Efficiency

(47) Thermal efficiency referred to indicated horsepower............per cent

(48) Thermal efficiency referred to brake horsepower..............per cent

Work Done per Heat Unit

(49) Net work per B.t.u. consumed (1,980,000 ÷ Item 40)...........ft.lb.

HEAT BALANCE

(50) Heat balance, based on B.t.u. per i.h.p. per hour.....................

* In two cycle engines this includes the power required for compression.

† If these results, in the case of a gas engine, are based on the low value of the heat of combustion that fact should be so stated.

		B.t.u.	Per cent
(*a*)	Heat converted into work	2546.5	....
(*b*)	Heat rejected in cooling water	....	....
(*c*)	Heat rejected in the dry exhaust gases	....	....
(*d*)	Heat lost due to moisture formed by burning of hydrogen	....	....
(*e*)	Heat lost in superheating moisture in gas and air	....	....
(*f*)	Heat lost by incomplete combustion	....	....
(*g*)	Heat unaccounted for, including radiation	....	....
(*h*)	Total heat consumed per i.h.p-hr., same as Item 38	....	....

Sample Diagrams

(51) Sample indicator diagrams from each cylinder and if possible a stop-motion light-spring diagram showing inlet and exhaust pressures ....

Note:—For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code.

PART XIII

RULES FOR CONDUCTING TESTS OF WATERWHEELS

INTRODUCTION

199 Waterwheel tests may be divided into two classes, one of which may be termed "shop" tests and the other "field" tests. The former refer to those which are conducted in a plant devoted exclusively to testing work, and the latter to tests of the wheel in its permanent location. The Holyoke Water Power Company's testing flume is an example of a shop-testing plant, being one which is equipped for turbine water wheels of any size up to 300 h.p. at 18-ft. head. This plant, it is understood, is also at present the only one of the kind in the country which is available for commercial work. Under these circumstances there seems to be no call at the present time (1915) for a general code of rules applying to tests of that character. The tests to which the following code refers are therefore limited to field tests, the wheel being in place, and operating so far as possible under the conditions of service for which it was installed.

OBJECT AND PREPARATIONS

200 The usual object of a waterwheel test in the field is the determination of the capacity and efficiency of the wheel at various gate openings and if practicable at various speeds, as compared with standard or guaranteed performance. Having determined the object, whatever it may be, take the dimensions, note the physical condition of the wheel and of the plant throughout, install the testing appliances, etc., following the general instructions given in ¶ 1 to 20, so far as they pertain to the work in hand, and make preparations for the test accordingly.

201 The most important preparations are those which relate to the determination of the power developed by the wheel, and the quantity of water which it consumes. The nature of these preparations is governed altogether by the character of the equipment. As regards power determination, the simplest method is the one applying to a case where the wheel drives an electric generator and the power is measured by calculation from the electrical output. Another simple method is one which may be used where the wheel serves as an auxiliary to steam power, and the load is reasonably constant, in which case the output is determined by ascertaining the difference between the indicated horsepower developed by the engine when the wheel is on and that developed when the wheel is off. Another method which is applicable to almost any situation where there is room, although the most difficult of the three, is the use of a friction brake attached to the waterwheel shaft, being arranged so as to take the place of a section of the shaft which may temporarily be removed. As to preparation for water measurement, the desirability of preserving the maximum head of water usually makes it necessary to gage the stream supplied to the wheel or the stream leaving it, and to select or prepare for this purpose a sufficient length of canal having a uniform cross-section to determine the required velocity by float measurement. Another method consists in the use of current meters or pitometers which have been properly calibrated. In cases where some part of the head may be sacrificed either in the head race or tail race, the measurement may be made by the insertion of a suitable weir.

APPARATUS AND INSTRUMENTS

202 The apparatus and instruments required for a capacity and efficiency test of a waterwheel are:

(*a*) **A friction brake, steam engine indicators, or electrical instruments, depending on the character of the equipment.**
(*b*) **Graduated scales showing the heights of water in the flume above the wheel and in the discharge pit beneath.**
(*c*) **One or more current meters or other apparatus for ascertaining the velocity of the water; or a weir.**

203 Directions for the use of these appliances may be found in ¶ 9 and in Appendices Nos. 18, 19, 21, and 5.

204 It is of the greatest importance that the water measured is that which is consumed wholly by the wheel. If water leaks by without going through the wheel, the quantity of leakage should be determined by independent measurement when the wheel is entirely shut off, in which case the gross quantity is corrected accordingly.

DURATION

205 The duration of a simple efficiency test of a waterwheel depends mainly upon the method of water measurement employed, and the time required to obtain a sufficient number of observations to insure a reliable average. After the desired load and other conditions have been obtained, continuous observations and measurements for a period of 15 minutes is sufficient for all practical purposes, provided the water is measured by a weir, but a longer time is necessary when other methods of measurement are used.

RECORDS

206 The records should be obtained in a manner conforming to the principles explained in ¶ 15 to 18. Readings of the weight on the brake arm, levels of water in the flume and discharge pit, indications of the current meters, and revolutions per minute, should be taken every five minutes, and at more frequent intervals if they show much fluctuation. In case of float measurement, repeated observations should be made one after the other throughout the whole period of the trial.

CALCULATION OF RESULTS

207 The total average head of water on the wheel is obtained by adding together the reading of the scale in the flume and the vertical distance between the zero of this scale and that of the scale in the discharge pit, and subtracting the reading of the latter scale, both readings being taken in reasonably still water. The velocity of water in the measuring canal is found by averaging the readings obtained at several points extending over the whole width of the canal. The cubic feet of water flowing per second is obtained by multiplying the cross-section of the stream in square feet by the velocity of the water in feet per second, determined as stated above. The total power of water available is obtained by multiplying the net weight of water in pounds discharged per second by the total average head in feet on the wheel, and dividing the product by 550. The brake horsepower developed by the wheel is found by multiplying the net weight on the brake arm in pounds by the circumference of the corresponding circle in feet and by the number of revolutions per minute, and dividing the final product by 33,000.

In the case of a wheel supplied through a penstock the head is found by adding together the pressure at the intake to the wheel case, the velocity head at this point, and the elevation of the point above the surface of the tail water, all expressed in feet.

In an impulse wheel, the head is the sum of the pressure at the nozzle in feet and the velocity head at that point in feet.

DATA AND RESULTS

208 The data and results should be reported in accordance with the form given herewith (Table 21), adding lines for data not provided for or omitting those not required, as may conform with the object in view:

TABLE 21 DATA AND RESULTS OF WATERWHEEL TEST ADAPTED TO BRAKE MEASUREMENT OF POWER

Code of 1915

(1)	Test of ....water wheel located at.....................
	To determine ....
	Test conducted by ....
(2)	Type of wheel and class of service ....
(3)	Type of generator, if any, kind of current, etc. ....
(4)	Rated power of wheel ....h.p.
(5)	Cross-section of stream where velocity of water is measured......sq. ft.

GENERAL DATA

(6)	Date ....
(7)	Duration of period covered by test ....hr.
(8)	Average net weight on brake arm ....lb.
(9)	Average revolutions per minute ....r.p.m.
(10)	Total average head of water on wheel ....ft.
(11)	Average velocity of water per second in measuring canal............ft.
(12)	Volume of water flowing per second.......................cu. ft.
(13)	Weight of water flowing per second (Item 12 × 62.35)............lb.
(14)	Leakage per second...lb.
(15)	Net water discharged by wheel per second (Item 13 — Item 14)....lb.

POWER

(16)	Total power of water available ....h.p.
(17)	Brake horsepower developed by wheel.......................br. h.p.

EFFICIENCY

(18)	Efficiency of wheel, (Item 17 ÷ Item 16) × 100...........per cent

THIRD SECTION

PART XIV

APPENDICES

APPENDIX NO. 1

CLEARANCE MEASUREMENT BY WATER

209 To measure the clearance by actual test, the engine is carefully set on the center, with the piston at the end where the measurement is to be taken. Assuming, for example, a Corliss engine, the best method is to remove the steam valve so as to have access to the whole steam port, and then fill up the clearance space with water, which is poured into the open port through a funnel. The water is drawn from a receptacle containing a quantity previously measured. When the whole space, including the port, is completely filled, the quantity left is measured, and the difference shows the amount which has been poured in. The measurement can be easily made by weighing the water, and the corresponding volume determined by calculation, making proper allowance for its temperature. The proportion of clearance, both in steam cylinders and in cylinders of gas and oil engines, is the volume in cu. in. thus found, divided by the volume of the piston displacement, also in cu. in., and the result expressed as a decimal. In this test care should be taken that no air is retained in the clearance space when it is filled with water.

210 The only difficulty in measuring the clearance in this way is that occurring when the exhaust valves and piston are not tight, and the water poured in flows away and is lost. If the leakage is serious, no satisfactory measurement can be made, and it is better to depend upon the volume calculated from the drawing. If not too serious, however, an allowance can be made by observing the length of time consumed in pouring in the water; then, after a portion of the water has leaked out, fill up the space again, taking the time and measuring the quantity thus added, determining in this way the rate of leakage. Data will thus be obtained for the desired correction.

APPENDIX NO. 2

LEAKAGE TESTS OF ENGINES, INCLUDING TIME-METHOD

211 The method of testing the valves and pistons for leakage in a Corliss engine, or one in which the admission valves can be operated independently of the exhaust valves, is as follows:

212 First close the two steam valves, open the two indicator cocks, and admit a full pressure of steam into the chest by opening the throttle valve. The movement of the starting bar, first one way and then the other, so as to close one exhaust valve and then the other, causes the leakage through the steam valves to escape from the open indicator cock, where it becomes visible. The quantity of leakage is judged by the force of the current of steam blowing out.

213 To test the exhaust valves and piston, block the flywheel so that the piston will be at a short distance from the end of the stroke, and turn on the steam. The leakage escapes to the exhaust pipe, and can be observed at the open atmospheric outlet. If the outlet is not visible, and there is a valve in the exhaust pipe, this can be shut and the indicator cock opened, thereby deflecting the steam which leaks, and causing it to appear at the indicator cock. In a condensing engine where no atmospheric pipe is provided, and there is no opening that can be made in the exhaust pipe in front of the condenser, some idea can be obtained in regard to the amount of leakage by observing how rapidly the condenser is heated.

214 It is well to make these tests with the piston in different positions, so as to cover the whole range of the length of the stroke.

215 Another method of testing leakage is called the "time method." Instead of observing the steam that actually blows through the valves or piston to be tested, they are subjected to full steam pressure, and when the parts are thoroughly heated, the throttle valve is shut and the length of time observed which is required for the pressure to disappear. In testing the piston and exhaust valves, the flywheel is blocked as before, and, preferably, an indicator is attached, and a line drawn on a blank card at intervals of, say, one-quarter of a minute after the valve is shut, thereby making a record of the fall of the pressure. In a tight engine the fall of the pressure is slow, whereas in a leaky engine it is sometimes very rapid. The relative condition of the engine as compared with a tight engine must be judged by the observer, who must, of course, have had experience in tests of this kind on engines in various conditions.

216 The leakage of a piston can always be determined by removing the cylinder head and observing what blows through the open end with the pressure of steam behind it. The advantage of the time-method is that it saves the labor and time required in removing the cylinder head and replacing it, which, in cases of large engines, is considerable.

217 Leakage tests of single-valve engines cannot be made as satisfactorily as those of the Corliss type and other four-valve engines. The best that can be done as regards the valve is to place it at or near the center of its travel, covering both ports, and then make the test under full pressure. The valve and piston can be tested as a whole by blocking the flywheel and opening the throttle valve in the same way as in other engines. In locomotives leakage will be revealed by the escape of steam at the top of the smoke stack.

218 In testing compound engines for leakage, the work is somewhat simplified in case of any one cylinder, as compared with a simple engine. For example, leakage of the high-pressure cylinder can be revealed by opening the indicator cock on the proper end of the low-pressure cylinder, the steam valve of that cylinder being open. The test of leakage of the low-pressure exhaust valves and piston when the time-method is used may be based on the indications

of the receiver gage, instead of using an indicator. In that case the fall of the pressure due to leakage is read directly from the gage.

219 The tests thus far referred to are qualitative, and not quantitative. It is practical in some cases to determine the quantity of leakage under any set of conditions by collecting the steam which passes through, condensing it and weighing it. This can be readily done when there is a surface condenser, and it can be done in the absence of such a condenser by attaching a small pipe to the exhaust, and carrying the steam which escapes into a tank of water and condensing it. How much dependence can be placed upon the results of such a quantitative test as showing the actual quantity of leakage which occurs when the valves and pistons are in motion must be left to the judgment of the person who makes the test.

220 In Corliss engines the leakage of the piston with the engine in operation can be observed by removing the cylinder head, disconnecting the steam and exhaust valves at the head end, and setting the engine to work with steam admitted at the crank end.

APPENDIX NO. 3

GENERAL PRECAUTIONS REGARDING LEAKAGE AND METHODS OF MEASURING LEAKAGE

(a) Leakage

221 It is not always necessary to blank off a connecting pipe to make sure that there is no leakage through it. If satisfactory assurance can be had that there is no chance for leakage, this is sufficient. For example, where a straightway valve is used for cutting off a connecting pipe, and this valve has double seats with a hole in the bottom between them, this being provided with a plug or pet cock, assurance of the tightness of the valve when closed can be had by removing the plug or opening the cock. Likewise, if there is an open drip pipe attached to an unused or empty section of pipe beyond the valve, the fact that no water escapes here is sufficient evidence of the tightness of the valve. The main thing is to have positive evidence in regard to the tightness of the connections, such as may be obtained by the means suggested above; but where no positive evidence can be obtained, or where the leakage that occurs cannot be measured, it is of the utmost importance that the connections should be broken and blanked off.

222 Leakage of relief valves which are not tight, drips from traps, separators, etc., and leakage of tubes in the feedwater heater must all be guarded against or measured and allowed for.

223 It is well, as an additional precaution, to test the tightness of the feedwater pipes and apparatus concerned in the measurement of the water by running the pump at a slow speed for, say, fifteen minutes, having first shut the feed valves at the boilers and making sure they are tight. Leakage will be revealed by disappearance of water from the supply tank. In making this test, a gage should be placed on the pump discharge to guard against undue or dangerous pressure.

(b) Water Glass Tests of Leakage

224 To determine the leakage of steam and water from a boiler and steam pipes, etc., the water-glass method may be satisfactorily employed. This consists of shutting off all the feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approximately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well, in this test, as in other work of this character, to make observations every ten minutes, and to continue them for such length of time that the differences between successive readings attain a constant rate. In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of water due to leakage of steam and water become approximately constant. It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hourly periods. When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure. The water columns should not be blown down during the time a water-glass test is going on, nor for a period of at least one hour before it begins.

225 If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable, and properly allowed for.

(c) Surface Condenser Tests

226 In making an engine test where the steam consumption is determined from the amount of water discharged from a surface condenser, leakage of the piston rods and valve rods should be guarded against; for if these are excessive, the test is of little use, as the leakage consists partly of steam that has already done work in the cylinder and of water condensed from the steam when in contact with the cylinder. If such leakage cannot be prevented, some allowance should be made for the quantity thus lost. The weight of water as shown at the condenser must be increased by the quantity allowed for this leakage.

Leakage of the condenser itself may be determined by operating the condenser when all steam from the engine or turbine is shut off, and observing the rate at which the water, if any, is discharged by the air pump, correcting the results in the case of turbines for any steam or water used for shaft seals.

When salt water is used for circulating water, leakage may be determined by testing the condensate with silver nitrate.

APPENDIX NO. 4

CALIBRATING WATER METERS

227 Referring to Fig. 2, two tees *A* and *B* are placed in the feed pipe,

and between them two valves *C* and *D*. The meter is connected between the outlets of the tees *A* and *B*, and the valves *E* and *F* are placed one on each side of the meter. When the meter is running, the valves *E* and *F* are opened, and the valves *C* and *D* closed. A small bleeder *G* is kept open to make sure that there is no leakage. A gage is attached at *H*. When the meter is tested, the valves *C*, *D* and *F* are closed, and the valves *E* and *I* are opened. The water flows from the valve *I* to a tank on platform scales. In testing the meter, the water is throttled at the valve *I* to obtain the desired rate of discharge, the gage meanwhile showing the working pressure. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be readily turned into the tank or away from it. When the desired speed has been secured, the end of the pipe is swung into the tank at the instant the pointer of the meter is opposite

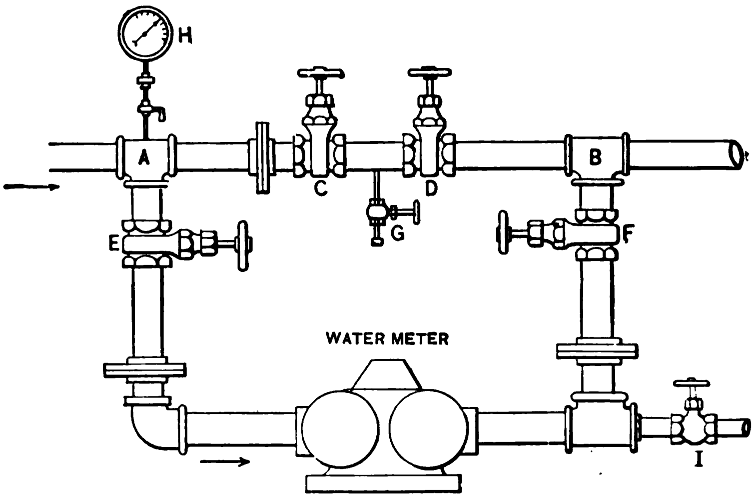


FIG. 2 METER CALIBRATION

some graduation mark on the dial. When the required number of cubic feet are discharged, the pipe is swung away. The tests should start and stop at the same graduation mark on the first dial, and continued until at least 10 or 20 cu. ft. are discharged for one test. The tank is weighed before and after filling.

228 The water passing the meter should always be under pressure so that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no unnecessary air drawn into the feed water. The meter should be tested before and after the trial, and repeated calibrations should be made to obtain confirmative results.

229 Fig. 2* and the description apply to a piston meter, but any other type of meter carrying water under pressure may be calibrated in the same manner.

* Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 724, fig. 118.

APPENDIX NO. 5

MEASUREMENTS OF WATER BY MEANS OF WEIRS, VENTURI METERS, ETC.

Weirs

230 The measurement of water by the use of weirs may be based on Francis' experiments, an account of which is given in Lowell Hydraulic Experiments.[1] These resulted in the formula

$$Q=3.33\ (L-0.2H)\times H^{\frac{3}{2}}$$

in which Q is the discharge in cu. ft. per second, L the length of the weir in ft., and H the depth of water on the weir in ft. The coefficient, 3.33, was obtained from the mean of 88 experiments, the greatest variation from the mean in any individual case being 1 per cent. The length of the weir, in all but six of the experiments, was approximately 10 ft. The depth of water on the weir varied from 7 to 19 in. The formula applies to that type of weir having perfect contraction at each end, which was the form used in 65 experiments.

231 The weir was of rectangular cross-section, with a horizontal crest and vertical ends. The upper edge was made of cast iron, and the corner presented to the current was square and sharp. The horizontal part of the crest was ¼ in. wide, and the remaining part was bevelled off at an angle of 45 deg. The ends were of similar cross-section to the crest. The depth on the weir was taken by means of hook gages, 6 ft. from the weir, these gages being placed in wooden boxes situated on the sides of the canal and communicating with the water through small openings. Vertical gratings, for overcoming eddies in the current, were provided above the gages. The distance from the side of the canal to the end of the weir was about 2 ft., and the depth of the canal below the crest was, in most of the experiments, about 5 ft.

232 The Francis formula is applicable only to cases similar to those described. According to Mr. Francis' statement, it cannot be applied where the depth on the weir exceeds one-third of the length, nor to very small depths. Furthermore, the distance from the side of the canal to the end of the weir should not be less than three times the depth on the weir.

233 In using the formula, the depth should be corrected for the head due to the velocity of approach. The formula then becomes

$$Q=3.33\ (L-0.2\ H)\times[(H+h)^{\frac{3}{2}}-h^{\frac{3}{2}}]$$

in which h is the head due to the velocity of approach. This last may be determined from the formula

[1] D. Van Nostrand.

$$h=\frac{V^2}{64.3}$$

in which V is the velocity of approach in ft. per second, which may be determined by dividing the uncorrected discharge of water, in cu. ft. per second, by the area of the cross-section of the stream flowing through the canal, in sq. ft.

234 Refer also to the experiments on weirs made by Hamilton Smith, Jr., described in Smith's Hydraulics, and to those made by Fteley and Stearns, described in the Transactions of the American Society of Civil Engineers, 1883 and to the review of Bazin's work given in the Deep Water Ways report in the Transactions of the same society, 1900. For paper on V-notch weirs by D. Robert Yarnall, see The Journal Am. Soc. M. E., October 1912. See also Williams and Hazens' "Hydraulic Tables."

Venturi Tubes

235 According to Herschel's experiments, described in the Transactions of the American Society of Civil Engineers, November 1887 and January 1888, a venturi tube inserted in a force main may be used for determining the quantity of water discharged by a pumping engine. Such a tube, applied, for example, to a 24-in. main, has a total length of about 20 ft. At a distance of 4 ft. from the end nearest the engine, the inside diameter of the tube is contracted to a throat having a diameter of about 8 in. A pressure gage is attached to each of two chambers, one surrounding and communicating with the entrance or main pipe, and the other with the throat. Experiments made upon two tubes of this kind, one of which was 4 in. in diameter at the throat and 12 in. at its entrance, and the other about 36 in. in diameter at the throat and 9 ft. at its entrance, showed that the quantity of water which passes through the tube is very nearly the theoretical discharge through an opening having an area equal to that of the throat and a velocity that due to the difference in head shown by the two gages. Mr. Herschel states that the coefficient for these two widely varying sizes of tubes and for a wide range of velocity through the pipe was found to be within 2 per cent, either way, of 98 per cent. In other words, the quantity of water flowing through the tube in cu. ft. per second is expressed within 2 per cent by the formula

$$Q=0.98\times A\times\sqrt{2gh}$$

in which A is the area of the throat of the tube, and h the head, in ft., corresponding to the difference in the pressure of the water entering the tube and that found at the throat.

236 For accurate calculation the following formula may be used:

$$W=9{,}900\times d^2\times\sqrt{\frac{R^2}{R^2-1}}\times\sqrt{ML}$$

in which

W = weight in pounds of water discharged per hour.
d = diameter of throat in inches.
R = ratio of the areas of inlet end and throat.
= area of inlet end divided by area of throat.

ML = difference in inches of mercury levels in a U-tube attached to the two pressure chambers of the meter tube.

The water should enter the meter through a straight section of pipe, without excessive pulsations of pressure.

237 The substantial reliability of venturi meters of small size (6 in. entrance and 2 in. at throat), such as are readily calibrated, furnishes strong evidence of the practicability of this method of measurement for pumping engines.

Nozzles

238 The measurement of water by computation from its discharge through the nozzles of fire hose, furnishes a means of determining the quantity of water delivered by a pumping engine which can be applied without much difficulty. Freeman's investigations upon fire nozzles, described in the Transactions of the American Society of Civil Engineers, November, 1889, which covered a wide range of pressures and sizes, showed that the coefficient of discharge for a smooth nozzle of ordinary good form was within ½ of 1 per cent, either way, of 0.977, the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gage attached to a suitable piezometer at the base of the play pipe.

239 To use this method the water should be conducted to a pressure box and as many nozzles attached to the box as would be required to carry off the water. Four 1¼-in. nozzles, with a pressure of 80 lb. per sq. in., discharge the full capacity of a 2,500,000 gal. engine.

Orifice Formula

240 The quantity of water flowing through an orifice into the open air, expressed in cu. ft. per second, may be calculated by the use of the formula

$$8.02\,AC\sqrt{H}$$

in which A is the area of the orifice in sq. ft., C, a coefficient depending on the form of the orifice and velocity of approach, and H, the head over the center of the orifice measured in ft. For an orifice located in the side of a tank, and consisting of a circular opening in a thin metal plate with a smooth sharp edge, the value of the coefficient is very close to 0.6. (See Hamilton Smith, Jr.'s, work on Hydraulics.)

241 For description of "Brauer" method where a number of similar orifices are used and calibration made of a single one, see Zeitschrift des Verein Deutscher Ingenieure, 1892, p. 1493.

Float Rods, Current Meters, etc.

242 The velocity of water flowing in a stream may be obtained indirectly by the use of float rods, and directly by means of current meters.

243 Float rods may be patterned after those used in Francis' experiments, which consisted of tin tubes, 2 in. in diameter, the lower ends being loaded so as to float upright, and the upper ends standing a few inches out of water. They were used at each foot of width of the canal, the length of run being

70 ft., and the time was recorded on a chronograph. The mean velocity ranged from 0.5 ft. to 5 ft. per second, the number of experiments being 115. It was found that if the lower ends of the rods did not extend to near the bottom of the canal, the observed velocity should be multiplied by a coefficient obtained from the formula

$$1-0.116\,(\sqrt{D}-0.1)$$

in which D is the proportionate part of the total depth of the water not reached by the float. If for example the immersed part of the float is 0.9 of the total depth, D is 0.1 and the coefficient becomes $1 - 0.024 = 0.976$.

244 A current meter consists essentially of a small screw propeller which is operated by the current of flowing water in which it is immersed, the revolutions being shown on a dial placed at some convenient point to which the motion is transmitted electrically. The speed of revolution of the propeller, referred to a rating table, shows the velocity of the water. The Price electric meter is an example of a satisfactory instrument for this purpose. Whatever type of meter is used, its rating should be checked by calibration under conditions which are substantially the same as those occurring on the waterwheel test, especially those pertaining to depth of immersion and average velocity. The method of calibration considered most reliable is that which is carried on by moving the meter at a known velocity through still water.

245 Another method of measuring the discharge of turbines is the chemical or titration method, which consists of pouring into the head race at a constant rate a small stream of a concentrated solution of common salt, and taking samples of water from the tail race and analyzing or titrating them to determine the percentage of salt therein. To secure accuracy there must be (1) a constant discharge of the initial solution, (2) a perfect mixture, (3) a precise titration of both the initial solution and of the tail-race water. The rate of discharge of the initial solution to the discharge of the turbine is inversely proportional to their concentration. The amount of initial solution injected should be approximately 0.0001 of the turbine discharge.

For example, if 0.1 liter per second of initial solution containing 300 grams of salt per liter is poured into the stream, and after thorough mixing the water in the stream contains only 0.3 grams per liter, then the discharge is $300 \div 0.3 = 1000$ liters or 35.314 cubic feet per second. See Engineering and Contracting, Sept. 16, 1914; Engineering Record, Jan. 31, Aug. 22 and 29, 1914; Proc. Engrs. Society of Western Pennsylvania, May 1914; Electrical World, Sept. 5, 1914.

246 For further information regarding these and other methods of measuring the velocity of water, reference may be made to Water Supply and Irrigation Paper No. 95, U. S. Geological Survey, published in 1904, on the subject, Accuracy of Stream Measurements. For pitot tube measurement of water, see paper by H. C. Berry, read before Engineers' Club of Philadelphia, July, 1910.

APPENDIX NO. 6

PITOT TUBES FOR GAS AND AIR MEASUREMENT

247 A form of pitot tube suitable for measuring the velocity of gas or air moving in a pipe is illustrated in principle in Fig. 3. This consists of a

small tube *A*, which is inserted in such a position that the open end points toward the current of gas or air, thereby receiving the full impact which the current produces. Surrounding this tube is an annular chamber or jacket *B*, which communicates with the outside space by means of four small holes *C*, *D*, *E*, and *F*, two on each side. By means of the small pipes *G* and *H*,

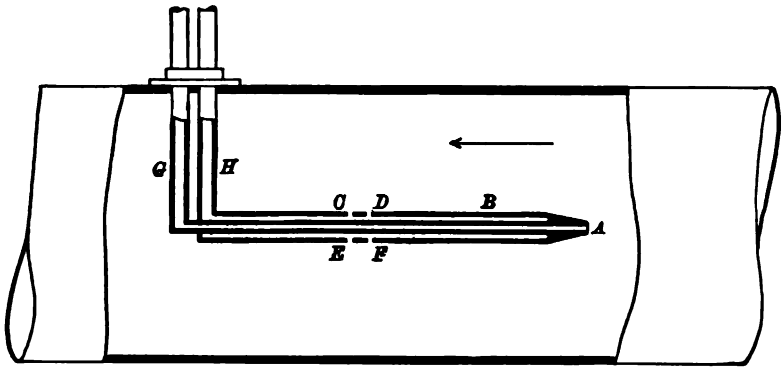


FIG. 3 PITOT TUBE

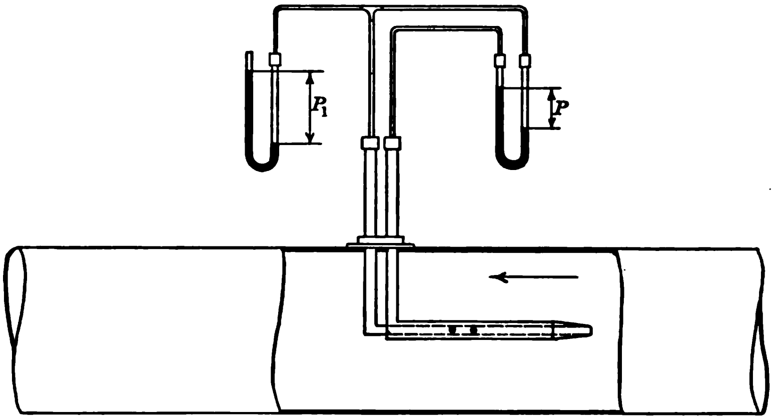


FIG. 4 PITOT TUBE AS INSTALLED

the inner tube and the outer jacket are connected respectively to the two legs of a U-tube manometer, as shown in Fig. 4.

248 The pressure shown by the manometer, or what is called the velocity-pressure, is that due to the difference in pressure at the two points, the inner tube *A* being subjected to the sum of the static and velocity pressures, and the annular jacket *B* to the static pressure alone. The velocity corresponding to the difference of pressure shown is computed by using the formula

$$\text{Velocity per minute} = 60\ \sqrt{2\,gH}$$

in which $H = \dfrac{\text{Velocity-pressure in lb. per sq. ft.}}{\text{Weight of 1 cu. ft. of gas or air.}}$

The velocity-pressure in lb. per sq. ft. is found by multiplying the indication of the manometer P in in. of water by the constant 5.196. By substituting in the above formula, and reducing, we have

$$\text{Velocity per minute} = 1097\ \sqrt{\frac{\text{Reading of manometer in in. of water.}}{\text{Weight of 1 cu. ft. of gas or air.}}}$$

The weight of 1 cu. ft. of gas or air is obtained by multiplying the weight of 1 cu. ft. at a temperature of 32 deg. and pressure of 29.92 in. (0.0807 lb. for dry air) by the proportion obtained from the expression

$$\frac{492}{460 + t} \times \frac{P + 29.92}{29.92}$$

in which t is the temperature of the gas or air as observed and P the observed pressure in in. of mercury.

249 For low velocities the U-tube shown in Fig. 4 may be replaced by one containing gasolene and having the two legs inclined so as to multiply the ordinary indications to any extent desired. In this case the reading may be converted into inches of water by multiplying it by the proportion of the vertical height to the inclined height and by the specific gravity of gasolene.

250 For high pressures, the manometer attached to the impact tube can be replaced by a mercury column or steam gage attached to a receiver in the main where the velocity is largely reduced.

251 To determine the average velocity of the gas or air throughout the whole cross-section of the pipe, readings should be obtained at a number of points and the results averaged. Two sets of readings should be taken, one set along the horizontal diameter and the other set on the vertical diameter. The various points should be selected so that the different velocities will apply to equal areas. If there are ten points in each diameter this equality will be obtained if they are located on the four radii at distances of 32, 55, 71, 84, and 95 per cent, respectively, of their length, measured from the center. In pipes carrying high pressures two pitot tubes may be used, one for each diameter, and they should be mounted so as to be readily moved to the desired locations, the required points being indicated by suitable marks on the outside supports.

252 The point selected for the attachment of a pitot tube should be as far removed as practicable from abrupt bends, or from the inlet opening. The distance should be equal at least to fifteen diameters of the pipe.

253 Another form of tube may be used, when properly calibrated, especially for high pressures, which is inserted diametrically across the main in the manner of a sampling pipe. In this case the impact is obtained by a number of short tubes attached to the side and pointing toward the current.

254 To get reliable indications from a pitot tube used for gas measure-

ment, it is important that the tube should be frequently examined to see that the passages are not obstructed by particles of dust. The pitot tube is not adapted to cases where the gas contains a large amount of dust.

APPENDIX NO. 7

BITUMINOUS COAL SIZES

255 Bituminous coals in the Eastern States may be graded and sized as follows:

(*a*) Run of mine coal; the unscreened coal taken from the mine after the impurities which can be practicably separated have been removed.

(*b*) Lump coal; that which passes over a bar-screen with openings 1¼-in. wide.

(*c*) Nut coal; that which passes through a bar-screen with 1¼-in. openings and over one with ¾-in. openings.

(*d*) Slack coal; that which passes through a bar-screen with ¾-in. openings.

256 Bituminous coals in the Western States may be graded and sized as follows:

(*e*) Run of mine coal; the unscreened coal taken from the mine.

(*f*) Lump coal; divided into 6-in., 3-in. and 1¼-in. lump, according to the diameter of the circular openings over which the respective grades pass; also 6 by 3 lump and 3 by 1¼ lump, according as the coal passes through a circular opening having the diameter of the larger figure and over one of the smaller diameter.

(*g*) Nut coal; divided into 3-in. steam nut, which passes through an opening 3-in. diameter and over 1¼-in.; 1¼-in. nut, which passes through a 1¼-in. diameter opening and over a ¾-in. diameter opening; and ¾-in. nut, which passes through a ¾-in. diameter opening and over a ⅝-in. diamter opening.

(*h*) Screenings; that which passes through a 1¼-in. diameter opening.

(*i*) Washed sizes; those passing through or over the circular openings of the following diameters, in inches:

Number	Through	Over
1	3	1¾
2	1¾	1⅛
3	1⅛	¾
4	¾	¼
5	¼	..

APPENDIX NO. 8

APPARATUS FOR PUMPING ENGINES

257 To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over com-

puted. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact scale marks, which are liable to creep in, may thereby be avoided.

258 The apparatus referred to under the heading *Pressure Gages*, ¶ 9g, consisting of a small air chamber for the attachment of a gage to the force main is shown in Fig. 5.*

APPENDIX NO. 9

HUMIDITY TABLE

259 Tables 22*a*, 22*b* and 22*c* show the percentage of humidity in moist air referred to total saturation, lb. of moisture per 1000 lb. of dry air, and weight of saturated air per cu. ft., for various differences of temperature given by dry and wet bulb thermometers, the latter instrument being of the sling type.

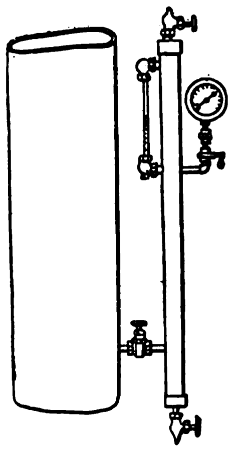


FIG. 5 AIR CHAMBER FOR ATTACHMENT OF GAGE TO FORCE MAIN

APPENDIX NO. 10

OPTICAL, PNEUMATIC, AND RADIATION PYROMETERS

260 Wanner's optical pyrometer, Uehling's pneumatic pyrometer, and Féry's radiation pyrometer, are among those referred to in ¶ 9k as being useful for measuring high temperatures. Full descriptions of these instruments may be found in the catalogues of various dealers in chemical and physical apparatus.

APPENDIX NO. 11

DESCRIPTION OF STEAM CALORIMETERS

261 The instrument referred to in ¶ 9m is shown in Figs. 6*a* and 6*b*. It consists of brass pipe and fittings of the ½-in. size in which the leading parts are the two tees, a pair of flanges, an orifice plate between the flanges,

* Reproduced from Trans. Am. Soc. M. E., vol. 12, p. 548, fig. 109.

HUMIDITY TABLES

TABLE 22-A RELATIVE HUMIDITY, in per cent (Total Saturation = 100%) Barometer 29.92"

Dry Thermom. Deg. Fahr.	Difference between Dry and Wet Thermometers, Deg. Fahr.								
	1	2	3	4	5	6	7	8	9
0	66.8	34.0	1.5	..	..	..	..	..	..
10	78.1	56.6	35.3	14.3	..	..	..	..	..
20	84.9	70.0	55.2	41.0	26.9	12.9	..	..	..
30	89.1	78.3	67.5	56.8	46.5	36.4	26.3	16.5	6.8
40	91.6	83.4	75.3	67.5	59.9	52.4	45.0	37.7	30.5
50	93.5	87.0	80.6	74.3	68.0	61.9	55.8	50.0	44.3
60	94.5	89.0	83.6	78.3	73.1	68.1	63.1	58.3	53.6
70	95.3	90.6	86.0	81.6	77.2	72.9	68.6	64.4	60.4
80	95.8	91.7	87.7	83.7	79.9	76.1	72.3	68.6	65.0
90	96.1	92.3	88.7	85.1	81.7	78.3	75.0	71.7	68.5
100	96.5	93.0	89.7	86.4	83.2	80.0	77.0	74.0	71.0
110	96.7	93.5	90.3	87.2	84.2	81.2	78.3	75.6	72.9
120	97.0	94.0	91.0	88.0	85.1	82.3	79.6	76.9	74.3
130	97.1	94.2	91.3	88.5	85.7	83.1	80.6	78.1	75.7

	10	11	12	13	14	15	16	17	18
40	23.5	16.5	9.7	3.0	..	..	..	..	..
50	38.7	33.2	27.8	22.4	17.2	12.1	7.0	2.0	..
60	49.1	44.6	40.1	35.7	31.4	27.1	22.8	18.6	14.5
70	56.4	52.5	48.7	44.9	41.1	37.4	33.8	30.3	26.9
80	61.5	58.1	54.8	51.5	48.2	44.9	41.7	38.6	35.6
90	65.3	62.1	59.1	56.1	53.2	50.2	47.4	44.7	42.0
100	68.0	65.1	62.3	59.5	56.8	54.2	51.6	49.1	46.7
110	70.2	67.5	65.0	62.5	60.0	57.5	55.1	52.8	50.5
120	71.8	69.4	67.0	64.6	62.3	60.1	57.9	55.7	53.6
130	73.4	71.1	68.8	66.6	64.5	62.4	60.3	58.3	56.3

	19	20	21	22	23	24	25	26	27
60	10.5	6.5	2.6	..	..	..	..	..	..
70	23.5	20.2	17.0	14.0	11.0	8.0	5.0	2.1	..
80	32.6	29.8	27.0	24.3	21.6	19.0	16.4	13.9	11.4
90	39.4	36.8	34.3	31.9	29.5	27.2	24.9	22.6	20.5
100	44.4	42.1	39.8	37.6	35.5	33.4	31.3	29.3	27.4
110	48.3	46.1	44.0	42 0	40.0	35.0	36.1	34.2	32.4
120	51.6	49.6	47.6	45.6	43.7	41.8	40.0	38.2	36.4
130	54.4	52.5	50.6	48.7	46.9	45.1	43.4	41.7	40.0

	28	29	30
80	9.0	6.7	4.4
90	18.3	16.2	14.1
100	25.5	23.6	21.7
110	30.6	28.9	27.2
120	34.7	33.0	31.4
130	38.3	36.7	35.2

TABLE 22-B WEIGHT OF MOISTURE per 1,000 Lb. of Dry Air, in Lb. Barometer 29.92"

Dry Thermom. Deg. Fahr.	Vapor Pressure Inches Mercury	Difference between Dry and Wet Thermometers, Deg. Fahr.							
		0	1	2	3	4	5	6	7
0	0.0383	0.8	0.5	0.3	0.0	..	..	..	..
10	0.0631	1.3	1.0	0.8	0.5	0.2	..	..	..
20	0.1026	2.1	1.8	1.5	1.2	0.9	0.6	0.3	..
30	0.1640	3.4	3.0	2.7	2.3	1.9	1.6	1.2	0.9
40	0.2477	5.2	4.8	4.4	3.9	3.5	3.1	2.7	2.3
50	0.3625	7.7	7.2	6.7	6.2	5.7	5.2	4.7	4.3
60	0.5220	11.0	10.4	9.8	9.2	8.7	8.1	7.5	7.0
70	0.7390	15.8	15.0	14.2	13.5	12.8	12.1	11.4	10.7
80	1.0290	22.2	21.2	20.2	19.3	18.4	17.5	16.7	15.8
90	1.4170	30.9	29.7	28.5	27.3	26.1	25.0	23.9	22.8
100	1.9260	43.3	41.6	40.0	38.4	36.8	35.4	34.0	32.6
110	2.5890	59.6	57.5	55.4	53.4	51.5	49.6	47.8	45.9
120	3.4380	82.5	79.7	76.8	74.1	71.4	68.8	66.3	63.9
130	4.5200	112.5	108.9	105.3	101.7	98.2	94.9	91.7	88.6

Dry Thermom. Deg. Fahr.	8	9	10	11	12	13	14	15	16
30	0.6	0.3	..	..	..	..	..	..	..
40	1.9	1.6	1.2	0.8	0.5	0.2	1.3	..	..
50	3.8	3.4	2.9	2.5	2.1	1.7	3.4	0.9	0.5
60	6.4	5.9	5.4	4.9	4.4	3.9	6.4	2.9	2.5
70	10.1	9.4	8.8	8.2	7.6	7.0	10.4	5.8	5.2
80	15.0	14.2	13.5	12.7	11.9	11.2	16.0	9.7	9.0
90	21.8	20.8	19.8	18.8	17.9	16.9	23.9	15.2	14.3
100	31.2	29.9	28.6	27.3	26.2	25.0	34.5	22.8	21.7
110	44.1	42.4	40.7	39.1	37.6	36.0	49.0	33.0	31.6
120	61.5	59.3	57.1	55.1	53.0	51.0	68.9	47.0	45.1
130	85.7	82.8	79.9	77.1	74.3	71.5	95.3	66.2	63.6

Dry Thermom. Deg. Fahr.	17	18	19	20	21	22	23	24	25
50	0.1	..	..	..	..	..	..	..	..
60	2.0	1.6	1.1	0.7	0.3	..	..	..	..
70	4.7	4.1	3.6	3.1	2.6	2.1	1.6	1.1	0.7
80	8.4	7.7	7.1	6.5	5.9	5.3	4.7	4.1	3.5
90	13.5	12.7	11.8	11.1	10.3	9.6	8.9	8.1	7.4
100	20.7	19.7	18.7	17.7	16.8	15.8	14.9	13.9	13.0
110	30.1	28.8	27.5	26.3	25.0	23.8	22.6	21.5	20.3
120	43.2	41.4	39.7	38.0	36.5	35.0	33.5	32.0	30.5
130	61.1	58.6	56.3	54.1	52.0	50.0	48.0	46.2	44.4

Dry Thermom. Deg. Fahr.	26	27	28	29	30
70	0.2	..	..	..	..
80	2.9	2.4	1.9	1.3	0.8
90	6.7	6.1	5.4	4.8	4.2
100	12.1	11.3	10.5	9.7	8.9
110	19.2	18.1	17.0	16.0	15.0
120	29.1	27.7	26.3	25.1	23.8
130	42.6	40.9	39.2	37.5	35.9

TABLE 22-C WEIGHT IN POUNDS OF ONE CUBIC FOOT OF SATURATED AIR

Dry Thermom. Deg. Fahr.	Barometric Pressure—Inches				
	26	27	28	29	30
0	0.0750	0.07788	0.08077	0.08365	0.08654
10	0.07338	0.07620	0.07903	0.08185	0.08468
20	0.07180	0.07456	0.07733	0.08009	0.08286
30	0 07027	0.07297	0.07569	0.07839	0.08110
40	0.06879	0.07143	0.07409	0.07675	0.07942
50	0.06732	0.06992	0.07252	0.07512	0.07773
60	0.06588	0.06843	0.07098	0.07353	0.07600
70	0.06442	0.06692	0.06943	0.07193	0.07440
80	0.06297	0 06542	0.06789	0.07034	0.07280
90	0.06146	0.06388	0.06629	0.06870	0.07112
100	0.05901	0.06228	0 06465	0.06703	0.06939
110	0.05828	0.06060	0.06293	0.06526	0.06759
120	0.05653	0.05882	0.06111	0.06339	0.06569
130	0.05467	0.05692	0.05917	0.06142	0.06367

and two thermometer cups containing thermometers which are screwed into the tees. The whole is surrounded by suitable non-conducting covering and encased in a box. The percentage of moisture is determined by observing the number of degrees of cooling that the thermometer in the low-pressure steam shows below the ''normal'' reading for dry steam, and dividing that number by the ''constant'' number of degrees representing 1 per cent of moisture.

262 To determine the ''normal'' reading of the low-pressure thermometer corresponding to dry steam, the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upwards to near the top of the pipe, there being no perforations and the steam entering through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and when the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure falls during the time when the observations are being made, the test should be continued long enough to obtain the effect of an equivalent rise of pressure.

263 To find the ''constant'' for 1 per cent of moisture divide the latent heat of the steam supplied to the calorimeter at the observed pressure or temperature by the specific heat of superheated steam at atmospheric pressure (0.46) and divide the quotient by 100.

264 Finally ascertain the percentage of moisture by dividing the number of degrees of cooling by the constant, as above noted.

265 To determine the quantity of steam used by the calorimeter it is usually sufficient to calculate the quantity from the area of the orifice and the absolute pressure, using Napier's formula for the number of lb. which passes through per second; that is, absolute pressure in lb. per sq. in. divided by 70 and multiplied by the area of orifice in sq. in. To determine the quantity by actual test, a steam hose may be attached to the outlet of the calorimeter, and carried to a barrel of water on platform scales. The amount of

steam condensed in a certain time is determined, and thereby the quantity discharged per hour.

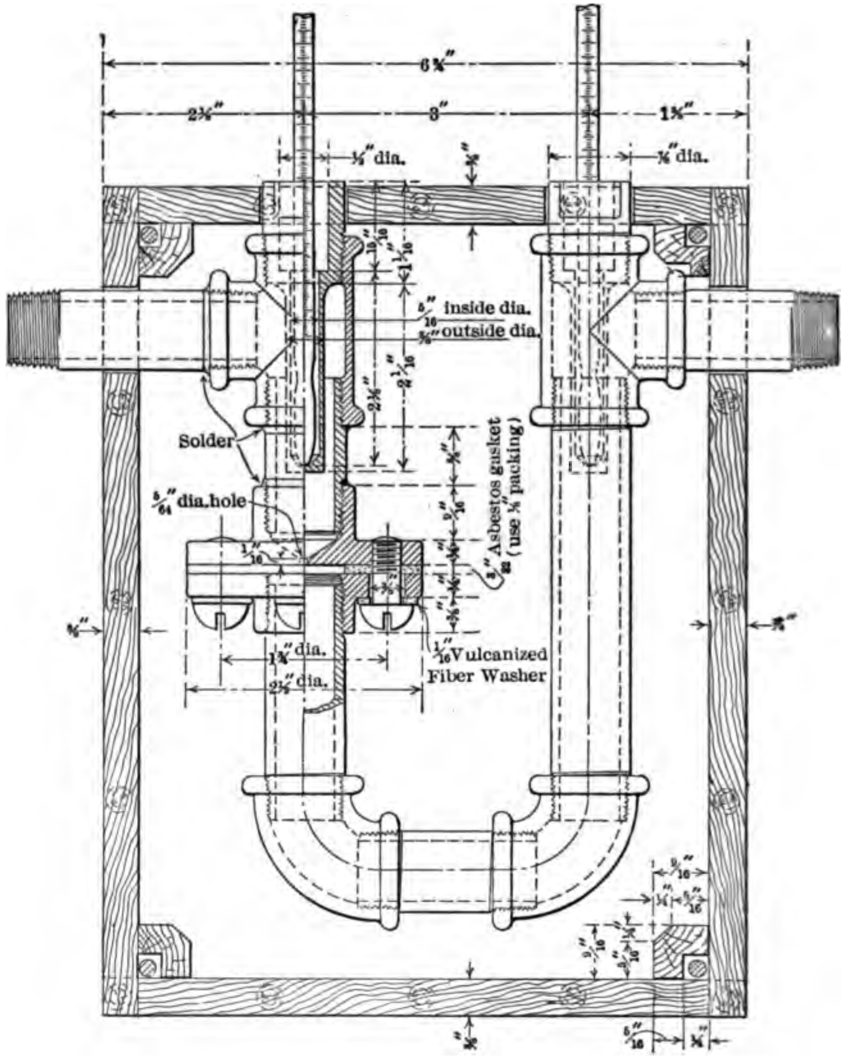


FIG. 6*a* THROTTLING CALORIMETER

266 A combined throttling and separating calorimeter, in which the throttling portion operates on the same principle as the one just described, is shown in Volume 17 of the Transactions.[1]

[1] New Form of Steam Calorimeter. Trans. Am. Soc. M. E., vol. 17, p. 618.

267 For testing low-pressure steam the discharge pipe of the calorimeter may be connected with the condenser of an engine.

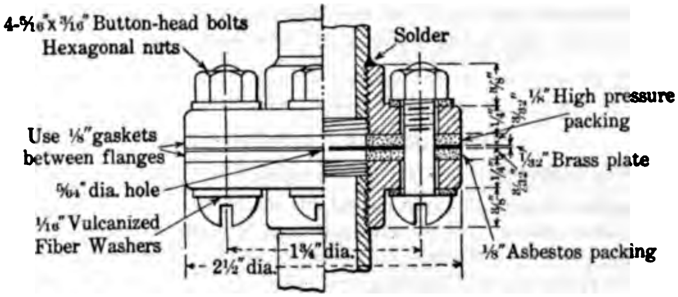


FIG. 6b ORIFICE PLATE FOR THROTTLING CALORIMETER

APPENDIX NO. 12

MAHLER COAL CALORIMETER

268 The Mahler coal calorimeter consists essentially of a strong cylindrical vessel having a capacity of about 800 cu. c., which is closed at the top and filled with oxygen gas, under a pressure of 375 lb. per sq. in. A sample of finely powdered coal, which will pass through a sieve having 100,000 meshes to the sq. in., weighing about 1 gram, is placed in a pan suspended within the interior vessel, and provided with two electrodes, through which an electric current from a battery can be passed. The whole is immersed in an outer vessel containing about 2500 grams of water, thoroughly stirred, the temperature of the water observed, the coal set on fire by completing the electric circuit, the water again stirred, and the temperature observed at intervals of half a minute until the thermometer ceases to rise. The difference between the initial and final temperatures thus determined is corrected for radiation, the latter being found by observing the rate at which the temperature changes before and after the coal is fired.

The weight of water contained in the outer vessel is added to the water equivalent of the apparatus, and the sum of the two is multiplied by the corrected rise of temperature expressed in deg. cent. The heat generated in burning the fuse wire, the heat due to the formation of aqueous nitric acid, and that due to the combustion of sulphur to sulphuric acid, are subtracted from this product. The remainder, divided by the weight of fuel expressed in grams, is the heat of combustion expressed in kilogram-calories per kilogram. This result is multiplied by 1.8 to convert to heat of combustion expressed in B.t.u. per lb.

The correction for iron fuse wire is 1.6 calories per milligram. The correction for nitric acid, which is obtained by titrating the washings with standard ammonia solution (0.00587 grams of NH_3 per cc.), is 5 gram-calories per cc. of the ammonia solution. The correction for sulphur, which is obtained by precipitation as barium sulphate is 13 gram-calories per 0.01 gram of sulphur. (See Technical Paper No. 8, 1913, U. S. Bureau of Mines for the derivation of these figures and for other details.)

269 The sample used for the calorimeter test should be powdered and air-dried at the temperature of the room. A duplicate sample should be taken

for the determination of the moisture in this air-dried coal by heating in a drying oven to 220 deg. fahr. for one hour (or longer if necessary to obtain minimum weight), cooling in a desiccator and weighing. The results obtained on the calorimeter test should be corrected for the moisture thus found and reported as being referred to dry coal and to combustible.

APPENDIX NO. 13

JUNKER GAS CALORIMETER

270 This instrument consists of a vertical cylindrical water chamber containing vertical tubes, which is heated by the gas burned in a Bunsen lamp beneath. The products of combustion pass upward through a combustion chamber and downward through the tubes, while the water passes in at the bottom and out at the top in a continuous current. The quantity of gas is measured by a gas meter, and the quantity of water by collecting the overflow discharged from the apparatus. Thermometers are inserted at the points of entrance and exit. The heat of combustion of a cu. ft. of gas is determined by multiplying the rise of temperature in deg. fahr. by the weight of water in lb., and dividing the product by the volume of gas in cu. ft. The result thus found after being corrected for moisture and reduced to the equivalent at 60 deg. and 30 in., is what is termed the "higher value," and this is the value, unless otherwise stated, which is employed throughout the codes.

271 The "low value" is obtained by multiplying the weight of the condensed vapor resulting from the combustion, expressed in lb., by the total heat of atmospheric steam above the temperature of the condensed vapor, dividing the product by the volume of the gas in cu. ft., and subtracting the quotient from the higher value.

APPENDIX NO. 14

FUEL ANALYSES

Proximate Analysis of Coal

272 The apparatus required for proximate analysis consists of a mill for grinding coal, chemical scales sensitive to 1/1000 of the amount weighed, drying apparatus, a platinum crucible, a Bunsen burner and blast lamp, a supply of oxygen gas, and such chemicals and chemical apparatus as may be required. The elements to be determined are moisture, volatile matter, fixed carbon, ash, and sulphur.

273 Determine the loss from air drying and the total moisture in the ash as received as explained in ¶ 26.

274 To determine volatile matter, place about one gram of the air-dried powdered coal in the crucible and heat in a drying oven to 220 deg. fahr. for one hour (or longer if necessary to obtain minimum weight), cool in a desiccator and weigh. Cover the crucible with a loose platinum plate. Heat 7 minutes with a Bunsen burner giving a 6 to 8 in. flame, the crucible being supported 3 in. above the top of the burner tube and protected from outside air currents by a cylindrical asbestos chimney 3 in. diameter. Cool in a desiccator, remove the cover, and weigh. The loss in weight represents the volatile matter.

In the U. S. Bureau of Mines practice a 1-gram sample of fine (60-mesh) air-dried coal is heated to a temperature of 950° C. in a platinum crucible with a close-fitting cover for seven minutes over a No. 3 Meker burner giving a flame 16 to 18 cm. high. The crucible is placed so that its bottom is 2 cm. above the top of the burner. To protect the crucible from the effects of drafts it is surrounded by a sheet iron chimney of special design. The loss in weight minus the weight of moisture determined at 105° C. represents the volatile matter.

275 To ascertain the ash, expose the residue in the crucible to the blast lamp until it is completely burned, using a stream of oxygen if desired to hasten the process. The residue left is the ash.

The Bureau of Mines determines the ash in the residue from the moisture determination. The moisture is determined by heating 1 gram of the 60-mesh air dried coal in a porcelain crucible for one hour at 105° C. in a constant temperature heating-oven. To determine the ash the crucible is heated slowly in a muffle furnace until the volatile matter is driven off. Ignition in the muffle is continued at a temperature of 750° C. with occasional stirring of the ash until all the particles of carbon have disappeared. The crucible is cooled in a desiccator, weighed, heated again for half an hour, and weighed again. The process is repeated until the variation in weight between two successive ignitions is 0.0005 gram or less.

276 The difference between the residue left after the expulsion of the volatile matter and the ash is the fixed carbon.

277 To determine sulphur by Eschka's method, which is the one commonly used, a sample of 60-mesh coal weighing 1.3736 grams is mixed in a 30 c.c. platinum crucible with about 2 grams of Eschka mixture (2 parts light calcined magnesium oxide, 1 part anhydrous sodium carbonate) and about 1 gram of the Eschka mixture is spread over it as a cover. The mixture is carefully burned out over a gradually increasing alcohol or natural gas flame. When all black particles are burned out the crucible is cooled, the contents digested with hot water, filtered, washed, and the solution treated with saturated bromine water and hydrochloric acid, boiled, and the sulphur precipitated as barium sulphate by adding a solution of barium chloride. (For further particulars see Technical Paper No. 8, 1913 of the Bureau of Mines.)

Ultimate Analysis of Coal

278 The apparatus required for ultimate analysis consists of a mill and other apparatus for grinding and pulverizing the coal; chemical scales sensitive to 1/1000 of the amount weighed; drying apparatus; combustion apparatus, embracing a combustion furnace, a glass combustion tube one end of which is filled with copper oxide and chromate of lead and the other end with a roll of oxidized copper gauze, a porcelain boat, a set of bulbs containing hydrate of potassium, a U-tube filled with chloride of calcium, and a supply of pure oxygen and pure air; together with suitable chemicals and chemical apparatus required for the various processes. The elements to be determined are moisture, carbon, hydrogen, oxygen, sulphur, nitrogen, and ash.

279 The moisture is determined in the manner pointed out in ¶ 25.

280 The carbon and hydrogen are obtained by the use of the combustion apparatus. One-half gram of the pulverized oven-dried coal is placed in the porcelain boat, which is introduced between the copper roll and the copper oxide within the combustion tube. After the contents within have been thoroughly dried out by a sufficient preliminary heating aided by a current of dry air, the furnace is set to work and the coal burned by first passing air through the tube and finally oxygen, conducting the products of combustion

through the potash bulbs and the chloride of calcium tube. The carbon dioxide produced by the combustion of the carbon is absorbed by the potash, and the water formed by the combustion of hydrogen is taken up by the chloride of calcium. The quantity of carbon is determined by weighing the bulbs before and after, thereby obtaining the weight of the carbon dioxide produced, and then calculating the weight of carbon from the known composition of the dioxide. Likewise, the quantity of hydrogen is determined by weighing the calcium tube before and after, which gives the amount of water produced, and, dividing by 9, the amount of hydrogen.

281 Sulphur is found by the method described above under the heading Proximate Analysis, ¶ 277.

282 To determine nitrogen, a certain weight of coal is mixed with strong sulphuric acid and permanganate of potash and heated until nearly colorless. This process converts the nitrogen into ammonia and then into sulphate of ammonia, and the amount of sulphate is determined by making the solution alkaline and then distilling it. The nitrogen is found by calculation from the known composition of ammonia.

(Recent experiments show that the nitrogen thus found in coal is 0.2 to 0.3 per cent too low, and that in order to obtain more accurate results it is, necessary to add mercury and potassium sulphate. See paper by Fieldner and Taylor in Jour. Ind. and Engrg. Chem., February, 1915.)

283 The ash is found by weighing the refuse left in the combustion boat after the coal is completely burned.

284 The oxygen is the difference between the sum of the elements previously determined and the original weight of coal.

285 The ultimate analysis of coal, as will be seen from the above description, requires the use of so much chemical apparatus, and at best it is so complicated that it is not likely to be done except in a fully equipped chemical laboratory. It should not be undertaken by one who is not entirely familiar with all the details of the work.

286 See report of the joint committee on coal of the American Chemical Society and of the American Society for Testing Materials, also Technical Paper No. 8 of the U. S. Bureau of Mines for details of methods of sampling and of analysis.

Analysis of Liquid Fuels

287 The determination of carbon and hydrogen in liquid fuels is made in the same manner as that concerning the solid fuels above described, using special means for preventing loss in the various processes on account of the volatile characteristics of the fuel.

288 To determine the sulphur, the oil or other liquid is heated with nitric acid and barium chloride. The quantity of sulphate of barium thus produced is ascertained by filtering and weighing, and the sulphur calculated from the known composition of the compound.

289 The ultimate analysis of liquid fuel, like that of coal, should be undertaken only by a person familiar with all the necessary details.

APPENDIX NO. 15

GAS ANALYSIS

Orsat Apparatus

290 The Orsat apparatus is a portable instrument contained in a wooden case with removable sliding doors front and back, as shown in its simplest form in Fig. 7.* It consists essentially of a measuring tube or burette, three absorbing bottles or pipettes, and a leveling bottle, together with the connecting tubes and apparatus. The bottle and measuring tube contain pure water; the first pipette, sodium or potassium hydrate dissolved in three times its weight of water; the second, pyrogallic acid dissolved in a like sodium hydrate solution in the proportion of 5 grams of the acid to 100 cc. of the hydrate; and the third, cuprous chloride. These chemicals are sold by most of the large dealers.

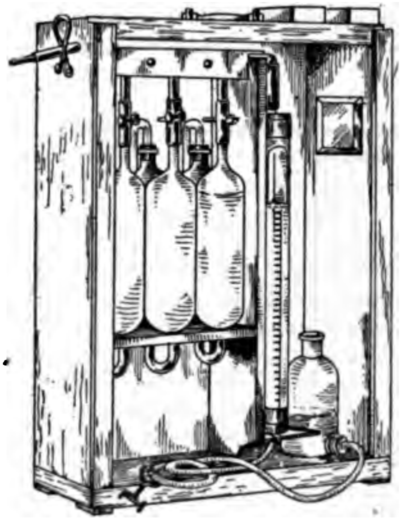


FIG. 7 ORSAT APPARATUS

291 The manipulation of the instrument, which can be carried on after suitable practice by any person familiar with testing work, is as follows:

After completely drawing out the air contained in the supply pipe, a sample of the gas is drawn into the measuring tube by opening the necessary connections and allowing the water to empty itself from the tube and flow into the bottle. The quantity of gas drawn in is adjusted to 100 cc. By opening one by one the connections to the pipettes, and raising and lowering the water bottle, the sample is alternately admitted to and withdrawn from the pipettes, and the ingredients one by one absorbed.

* Reproduced from Trans. Am. Soc. M. E., vol. 18, p. 903, fig. 291.

The first pipette absorbs carbon dioxide (CO_2); the second, oxygen (O); and the third, carbonic oxide (CO). The quantity absorbed in each case is determined by finally returning the sample to the measuring burette and reading the volume. The percentage of CO_2 is read directly, being the first absorption. Those of the other two ingredients are the respective differences between the readings taken after successive absorptions.

292 Various modifications of this apparatus have been developed which enable analyses to be made with greater rapidity than with the form illustrated in Fig. 7.

Hempel Apparatus

293 The Hempel apparatus works on the same principle as the simple form of Orsat apparatus described, so far as the latter is applicable, excepting that the absorption may be hastened by shaking the pipettes bodily, bringing the chemical into most intimate contact with the gas. It is less portable and in some particulars it requires more careful manipulation than the Orsat, while for general analysis it is not adapted unless used in a well equipped

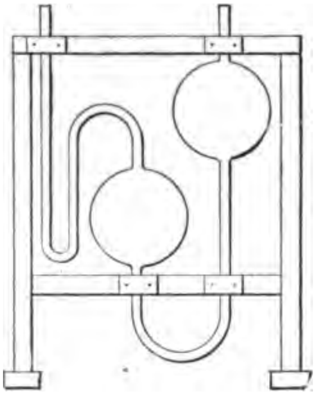


FIG. 8 HEMPEL PIPETTE

chemical laboratory. The absorption pipettes are made in sets which are shaped in the form of globes, and a number of independent sets are required for the treatment of the different constituent gases. A simple pipette of the Hempel type is shown in Fig. 8.

294 The method of carrying on an analysis with the Hempel apparatus is as follows:

A sample of gas measuring 100 cc. is drawn into the burette, and then transferred to the first pipette, which contains potassium hydrate dissolved in two times its weight of water. This pipette absorbs carbon dioxide (CO_2). The gas is then passed into the second pipette, containing saturated bromine water, which absorbs the heavy hydrocarbons (C_2H_4); then into the third pipette, containing a solution of pyrogallic acid and potassium hydrate in the proportion of 5 grams of acid to 100 cc. of hydrate, which absorbs oxygen (O); then into the fourth pipette, containing ammoniacal cuprous chloride, which absorbs carbonic oxide (CO), and finally into the fifth pipette, which is of large size and provided with exploding wires and galvanic battery, for the determination of marsh gas (CH_4) and hydrogen (H). A measured quantity of oxygen gas is added to this pipette and the contents exploded by an electric spark from the battery, resulting in a mixture of carbon dioxide, nitrogen, and free oxygen. The quantity of carbon dioxide is determined by passing the gas into the pipette containing potassium hydrate, and the quantity of oxygen by subsequently passing it into the pipette containing potassium pyrogallate, finally determining the quantity of marsh gas and hydrogen from the known reactions which occur during this process, and the composition of the resulting gases.

For each of these processes the pipettes are shaken to hasten the absorption, and the quantity absorbed is determined by returning the gas into the measuring burette and observing the successive differences.

Tar and Soot

295 The quantity of tar and soot contained in producer gas may be found by drawing a measured volume of gas through filter paper, weighing the paper before and after.

APPENDIX NO. 16

RINGLEMANN SMOKE CHART

296 A Ringelmann smoke chart is shown with full-size spacing in Fig. 9.* To use this chart, four cards are ruled like those shown, though covering a much larger area, and placed in a horizontal row about 50 ft. from the observer, and in line between him and the chimney, together with two other cards, one of which is white and the other solid black. The observer glances rapidly from the chimney to the cards and judges which one corresponds with the color and density of the smoke. He makes these observations every minute, or oftener if desired, recording the number of the card representing the character of the smoke at the instant of observation. The results are then plotted on a chart, and the variations shown graphically.

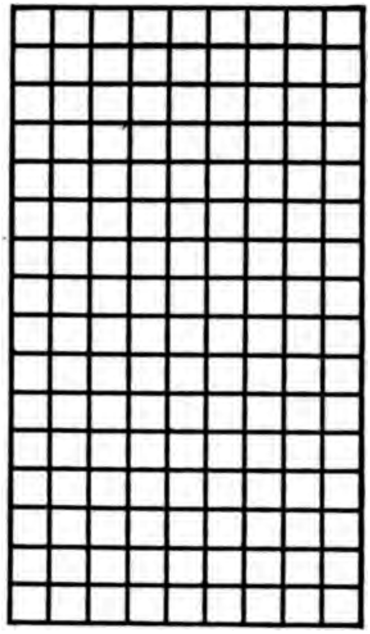
297 The lines in cards 1 to 4 are respectively 1, 2.3, 3.7, and 5.5 mm. thick, and the spaces 9, 7.7, 6.3, and 4.5 mm. The lines should be made with black India ink.

APPENDIX NO. 17

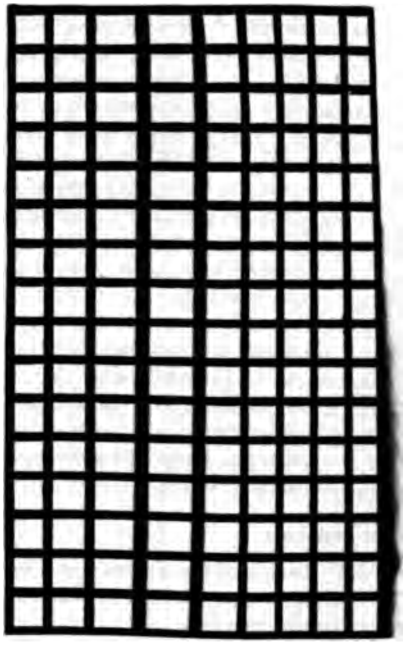
SMOKE METER

298 As an illustration of the soot-collecting method referred to in ¶ 9r, which was carried out in a certain trial, a plate ⅞ in. wide and 24 in. long, representing a surface of 21 sq. in., was inserted through a hole in the top of the flue and suspended by a wire, the hole being covered after the plate was inserted. The plate was temporarily withdrawn every two hours during

* Reproduced from Trans. Am. Soc. M. E., vol. 21, p. 98, fig. 6.



No. 1.



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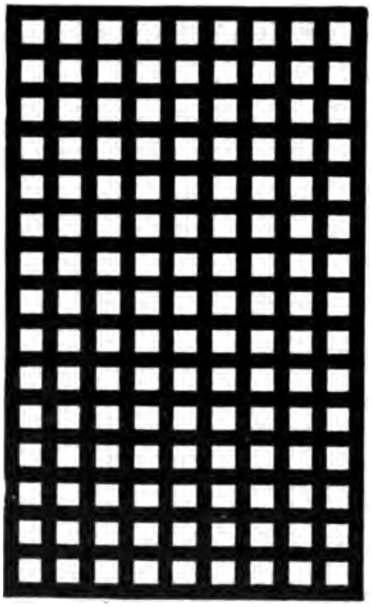

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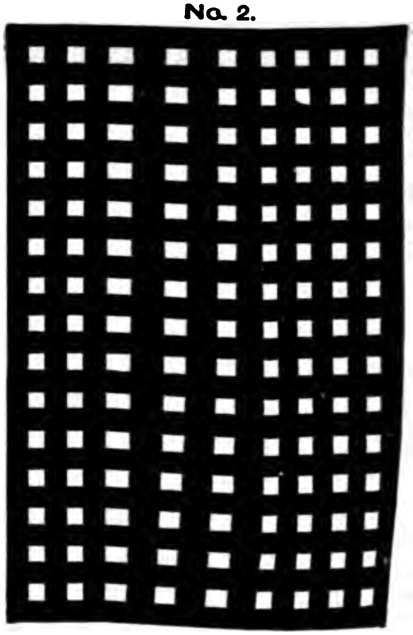


No. 4.

Fig. 9 Ringelmann Smoke Chart

the progress of the test, and the collection of soot removed by a brush. This was tried on a boiler fitted with an automatic stoker and on another fitted with a smoke-burning furnace, and it was found that under various conditions as to character of coal and other changes, the weight of soot collected per hour varied from 9 to 184 mg.

APPENDIX NO. 18

USE AND CALIBRATION OF INDICATORS

299 The indicated horsepower should be determined from the average mean effective pressure of diagrams taken at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes this necessary, this being done for each end of each cylinder. With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of 10 seconds until 2 minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a variation diagram should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test. Continuous indicators may be employed where the load is extremely variable.

It is seldom necessary, as far as average power measurements are concerned, to obtain diagrams at precisely the same instant at the two ends of the cylinder, or at the same instant on all the cylinders when there are more than one. All that is required is to take the diagram at regular intervals. Should the diagrams vary so much among themselves that the average may not be a fair one, it signifies that they should be taken more frequently, and not that special care should be employed to obtain the diagrams of each set at precisely the same time. When diagrams are taken during the time when the engine is working up to speed at the start, or when a study of valve setting and steam distribution is being made, they should be taken at as nearly the same instant as practicable. In cases where the diagrams are to be taken simultaneously, the best plan is to have an operator stationed at each indicator. This is desirable, even where an electric or other device is employed to operate all the instruments at once; for unless there are enough operators, it is necessary to open the indicator cocks some time before taking the diagrams and run the risk of clogging the pistons and heating the high-pressure springs above the ordinary working temperature.

To determine the power developed while an engine is starting from rest and attaining working speed and working load, a number of diagrams should be taken in rapid succession, the speed counted for each one, and the results averaged. If the period of time thus covered on a commercial test is unduly long, so as to be an appreciable percentage of the total running time, this power should be included in the average for the whole run, due regard being had for the time it is in operation. In that event the total duration of the run should be the whole time the throttle valve is open.

300 The most satisfactory driving rig for indicating is some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal length when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

With a perfect working pantagraph, or similar apparatus, the equality in the length of diagrams taken with the same indicator at the two ends is sufficient indication of the substantial reliability of the reduction when the point of cut-off on the diagram is not unusually short, say, not shorter than one-eighth. When the cut-off is unusually short, the error produced by imperfect reducing motion, stretch of cord, or otherwise, becomes a comparatively large item, and one which for accurate work should be allowed for. To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equi-distant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the crosshead has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up by turning sufficient steam into the cylinder to bring a pressure to bear on the piston, but not sufficient to start the flywheel in motion. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger, so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

301 In all cases the pipes leading to indicators should be as short and direct as possible. The use of a three-way cock and a single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced should be determined and allowed for.

In small cylinders, long pipes increase the clearance space, which in air compressors and gas engines is objectionable.

The effect of the error produced by a three-way cock is usually to increase the area of the diagram. This is due to the tardiness of the indicator in responding to the changes of pressure. In an investigation which was carried out both on short-stroke engines running at high speed and long-stroke engines running at comparatively slow speed, it was found that the increased area of the diagram, due to the sluggish action referred to, ranged from 3 to 7 per cent as compared with an indicator having a short and direct pipe.

The point selected for attaching the indicator pipe to the cylinder should never be the drip pipe or any point where the water of condensation will run into the instrument, if this can possibly be avoided. The admission of water with the steam may greatly distort the diagram. If it becomes necessary to place the indicator in such a position, as may happen when it is attached to the lower end of a vertical cylinder, the connection to the indicator must be short and direct, and in some cases it should be provided with a drip chamber arranged so as to collect the water or deflect it from entering the instrument.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of explosion. A special gas-engine indicator with a small piston and a strong spring is required for satisfactory work.

302 In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely, so as to get absolute freedom of motion, than to have a mathematically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams.

No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion of compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. Entire absence of wire drawing of the steam line, and especially a sharp, square corner at the beginning of the steam line, should be looked upon with suspicion, however desirable and satisfactory these features might otherwise be. These are frequently produced by an indicator which is defective owing to want of freedom in the mechanism. An indicator which is free when subjected to a steady steam pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to pressure, the movement should appear smooth to the sense of touch.

303 To make a comparison of indicator springs with standards, the calibration should be made, if this were practical, under the same conditions as those pertaining to their ordinary use. Owing to the fact that for steam work the pressure of the steam in the indicator cylinder and the corresponding temperature are undergoing continual changes, it becomes almost impossible to compare the springs with any standard under such conditions. There must be a constant pressure during the time that the comparison is being made. To bring the conditions for steam work as nearly as possible to those of the working indicator, the steam should be admitted to the indicator as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted therefrom before another pressure is tried. By this means the parts are heated and cooled somewhat the same as under the working conditions. For each required pressure the first step is to open and close the indicator cock a number of times in quick succession, then quickly draw the line for the desired record, observing the gage or other standard at the same instant. A corresponding atmospheric line is taken immediately afterwards.

304 The calibration should be made for at least five equidistant points. For ordinary work the arithmetical mean of the various results should be taken for the average scale.

305 The indicator springs used for gas and oil engines should also be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air is suitable for the actual work of calibration, being used in preference to steam so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use in such engines.

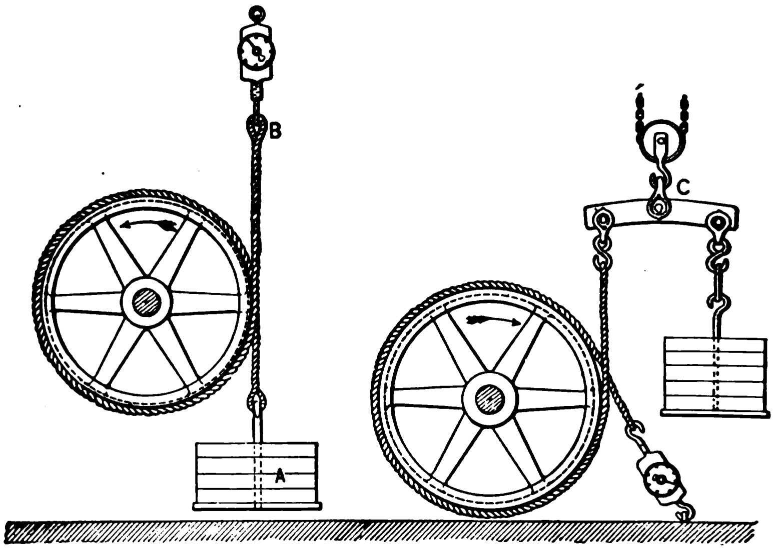
306 The standard of comparison in all cases is a steam gage which has been proved correct by reference to either a mercury column or a dead weight testing apparatus.

307 For engines running at extremely high speeds the optical indicator may be used.

APPENDIX NO. 19

FRICTION BRAKES

308 A self-adjusting rope brake is illustrated in Fig. 10* in which it will be seen that if the friction at the rim of the wheel increases it will lift the weight *A*, which action will diminish the tension in the end *B* of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar, *C*, supported by a chain tackle exactly at its central point, may be used as shown in Fig. 11,* thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more



FIGS. 10 AND 11 ROPE BRAKES

than a certain amount. To compute the horsepower, multiply the difference between the weight *A* and the weight shown on the spring balance by the number of revolutions of the pulley per minute, and by the circumference of a circle passing through the centre of gravity of the rope expressed in ft.; finally dividing the product by 33,000. The radius is measured at the point where that part of the rope under highest tension leaves the rim of the wheel.

If the self-adjusting feature is omitted the weight may be determined by the use of platform scales, the two ends of the rope being attached to a vertical stand supported by the platform, and provision being made for adjusting the tension.

* Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 739, figs. 119, 120.

309 A water-friction brake is shown in Fig. 12*. It consists of two circular disks, *A* and *B*, attached to the shaft *C*, and revolving in a case *E*, between fixed planes. The space between the disks and planes is supplied with running water, which enters at *D* and escapes at the cocks *F*, *G*, and *H*. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water, the greater the amount of power absorbed. By adjusting suitably the amount of water entering and leaving any desired power can be obtained.

310 Another form of water brake is that shown in Fig. 13, which is used by the Westinghouse Machine Company for large turbines. In this brake the shaft is carried in two bearings and is coupled to the turbine shaft. The

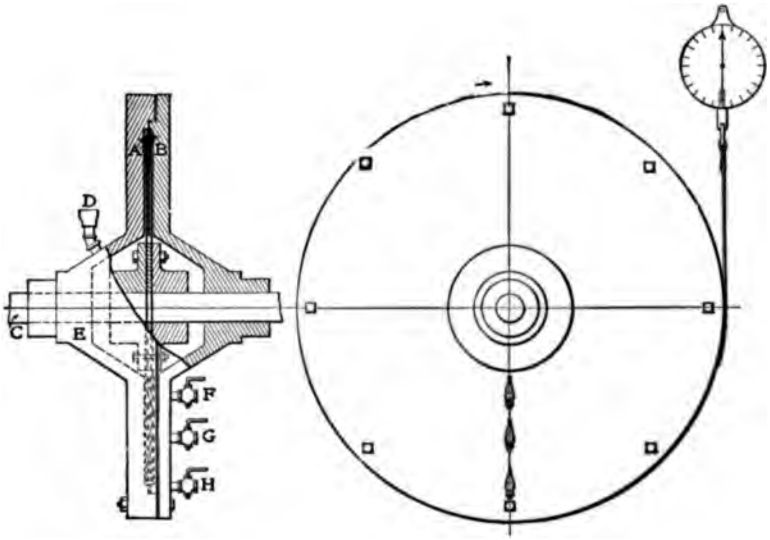


Fig. 12 Webb Water Brake

revolving paddle wheel or "runner" is designed to run in very close clearance with the serrated-rim piece. The outer casing of the brake maintains its accurate position surrounding the runner by means of carefully constructed bearings. To this casing is rigidly attached the lever arm, made of such length as to facilitate ready calculation. The little roller wheel on the end of the arm bears upon platform scales which weigh the load.

311 Water is introduced through a hose connected at the opening marked B. The water enters the interior of the runner, is thrown out by centrifugal force through small holes drilled in the rim into the outer teeth spaces where a resistance is created, and escapes with difficulty through the close clearance

* Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 740, fig. 121.

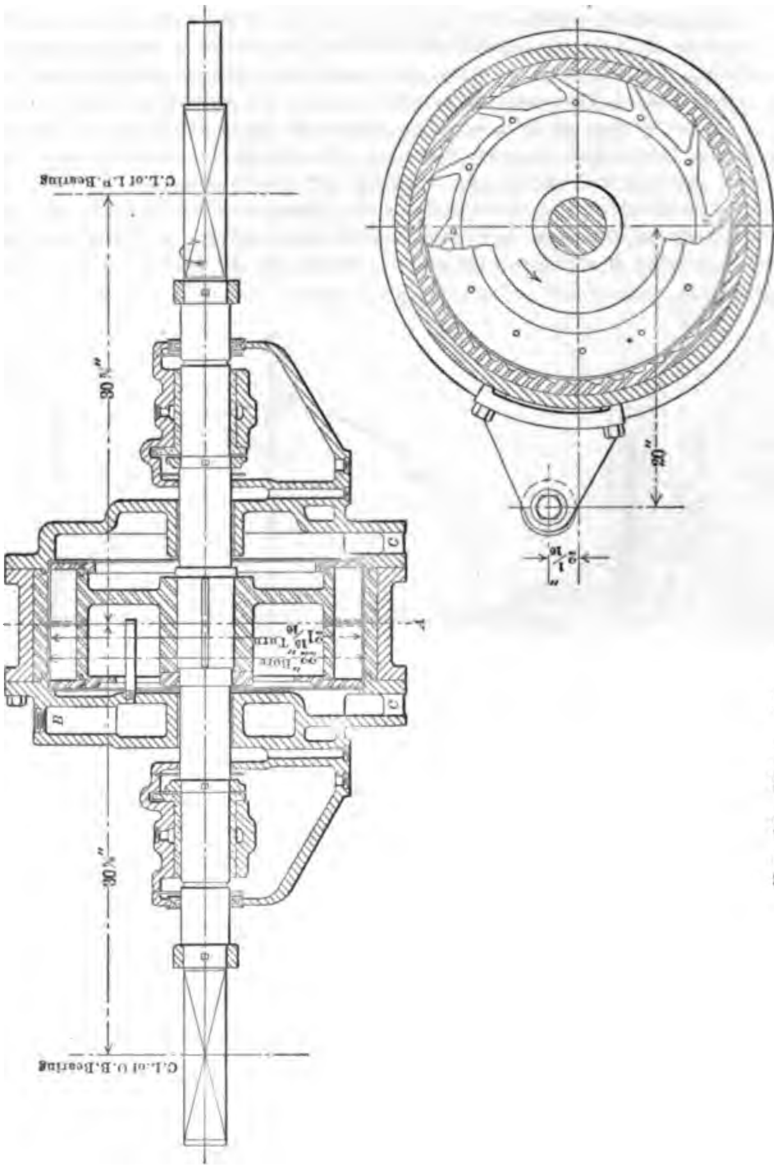


FIG. 13 WATER BRAKE

around the outside of the runner, finally discharging at the bottom through openings *C* (Fig. 13), where it runs to waste.

312 For a description of the Alden water brake, see the paper on An Automatic Absorption Dynamometer, in the Transactions.[1]

313 The application of water brakes for making field tests of waterwheels is shown in Professor Allen's paper on The Testing of Water Wheels After Installation, in the Transactions.[2]

APPENDIX NO. 20

DYNAMOMETERS

314 A form of transmission dynamometer for measuring the power transmitted from an engine to a mill shaft is well illustrated in principle by the Webber balance dynamometer, shown in Vol. 4 of the Transactions.[3] In this apparatus, which consists essentially of a compound gear, the intermediate gear is carried by a weighted arm, which is free to rise and fall according to the amount of force transmitted. The power is determined from the net weight thus lifted.

315 Another form which is useful for mill purposes is the belt dynamometer, the principle of which is shown in Fig. 14. *A* is the driving pulley and *B* the driven pulley. The connecting belt passes over the dynamometer pulleys *C* and *D*. These two pulleys are mounted on a frame carrying a scale beam *F*, all of which turns on the center *E*. The difference in the total strain on the two sides of the belt is computed by multiplying the net weight on the beam by the leverage, the latter being found by dividing the length of the beam by the distance from the center E to the centers of pulleys *C* and *D*.

316 The shaft dynamometer consists of the following essential parts: a long tube encircling the shaft and made fast thereto at one end, being free at the other end and maintained in alignment by adjustable rollers; two radial arms, one attached to the shaft and the other to the free end of the tube, which rotate a slight amount with reference to each other according to the twist of the enclosed length of shaft; and a set of levers which multiply this rotative movement and at the same time convert it into linear motion, which is transmitted to a sleeve and collar mounted upon the shaft and sliding thereon. These parts all revolve with the shaft. An independent indicating apparatus, which is mounted on a stationary frame, is provided, and the sliding movement of the rotating collar is transmitted through it to an index hand. The torsional strain is determined from the reading of the accompanying scale, which is graduated to millimeters.

317 The zero reading is found by disconnecting the propeller and turning the shaft at a slow speed, first in one direction and then in the other, observing the indication in both cases, and fixing the point of zero strain at the mean of the two. When it is impracticable to disconnect the propeller the readings may be taken when the vessel is drifting under her own headway after shutting off steam. The calibration of the instrument, which can best be done when the shaft is in the shop before installation, is carried on by

[1] Geo. I. Alden. Trans. Am. Soc. M. E., vol. 11, p. 958.

[2] C. M. Allen. Trans. Am. Soc. M. E., vol. 32, p. 275.

[3] S. S. and W. O. Webber. Trans. Am. Soc. M. E., vol. 4, p. 227. See also description of Prof. Durand's dynamometer, vol. 28, p. 697.

securing the shaft in a fixed position, and applying a torsional strain by means of weights at the end of a lever attached beyond the dynamometer, taking readings with a number of different weights.

318 The horsepower shown by the dynamometer is determined by multiplying the reading of the instrument expressed in millimeters by the number of revolutions of the shaft per minute, and by a constant determined from the calibration noted, the constant being an expression for the horsepower corresponding to 1 r.p.m. and a reading of 1 mm.

319 A shaft dynamometer requires delicate manipulation, a load of 500 h.p. in some instruments causing a movement at the end of the two arms of only 1/100 of an inch.

320 The locomotive dynamometer, as formerly designed, consists essentially of one or more strong helical springs placed between the drawbar of the locomotive and the train, and the distance which the spring is extended or compressed furnishes a measurement on an appropriate scale of the amount of force transmitted, and thereby the amount of power. The strain on the spring is transmitted through mechanism to a registering apparatus placed in the dynamometer car, and a continuous record is made upon a strip of paper traveling at a definite rate of speed. The apparatuses used by the Pennsylvania

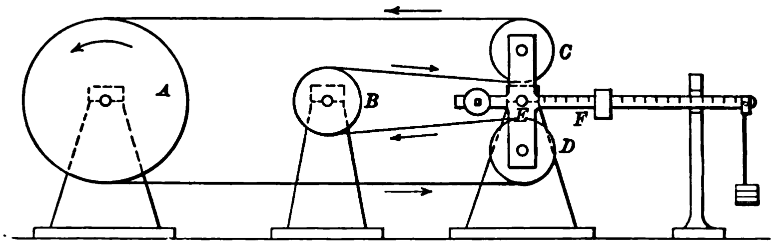


FIG. 14 BELT DYNAMOMETER

R. R.; the Chicago, Burlington and Quincy R. R., and the Chicago, Milwaukee and St. Paul R. R., are shown in the cuts given on pages 1329 to 1334 of Transactions, Vol. 14. The position of the dynamometer car should be such that the weight of the dynamometer car is included in the weight of the train. In pulling tests the measuring end is at the head of the car while in pushing tests it is at the rear of the car.

321 The latest form of dynamometer car makes use of oil plungers in place of springs, and employs oil transmission for the recording devices.

322 An electric dynamometer or brake, as developed by the Sprague Electric Works, is shown in Fig. 15. It consists of the generator *A* which receives the power to be measured and which is coupled to the shaft of the engine or motor *B*, the transformed energy being disposed of in suitable rheostats or otherwise. In the figure a gasolene engine supported on an adjustable frame, is indicated in broken lines. The field of the generator is free to turn on the ball bearings upon which it is mounted, except as it is held in position by connection to the beams of the scale. The effect of the torque acting between the interior revolving armature and the exterior field tends

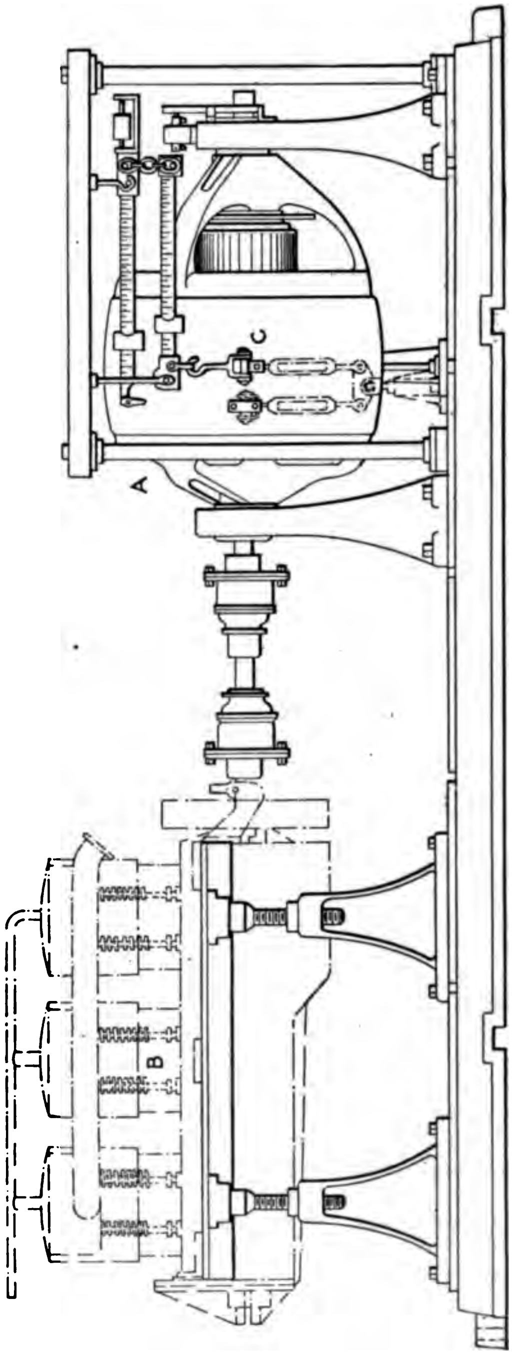


Fig. 15 Electric Dynamometer

to turn the field in the direction of rotation, and the pull thus produced acts either downward or upward on the scale beam according as the rotation is clockwise or counter-clockwise. If the pull is downward as viewed from the driving end, the beam is connected direct to a point C on the field frame. If upward, the beam is connected through a double link and rocker arm as shown by the dotted lines.

APPENDIX NO. 21

USE AND CALIBRATION OF ELECTRICAL INSTRUMENTS

323 To determine whether the readings of an instrument are disturbed by stray fields, the instrument is turned bodily through an angle of 180 deg. If there are stray fields, the change of position will change the reading, in which case the instrument should be moved to some other point where the reading is found to be unaffected.

324 The proper methods of connecting wattmeters in polyphase alternat-

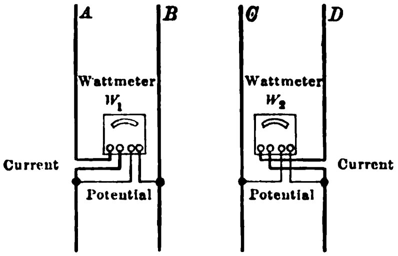


FIG. 16 TWO-PHASE SYSTEM

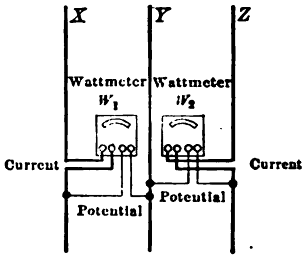


FIG. 17 THREE-PHASE SYSTEM

ing-current systems are shown in Figs. 16 and 17. The former refers to two-phase and the latter to three-phase systems. In Fig. 16 there are four cables, designated A, B, C, and D, arranged in two pairs, or one pair for each phase. Each wattmeter is connected to its respective pair of cables, the current coil being in series with one of the two cables, and the potential coil in parallel with the load or bridging the two cables. In Fig. 17 there are three cables, which for convenience are designated by the letters X, Y, and Z. Meter W_1 is connected to the phase represented by the cables X and Y, and meter W_2 to that represented by the cables Y and Z. The current coil of the former is connected in series with cable X, the potential coil bridging X and Y. The current coil of the latter is connected in series with cable Z, the potential coil bridging Y and Z. The connections shown are elementary, transformers and other equipment usually employed in actual installations being omitted. The total output is the arithmetical sum of the quantities indicated by the two meters for a two-phase system, and the algebraic sum for a three-phase system.

325 The power factor is the proportion borne by the true power or kilowatt output as shown by wattmeters, to the apparent power or kilovolt-ampere (kva) output determined from the readings of the voltmeters and ammeters. The kva output in a single-phase system is found by multiplying the volts

by the amperes and dividing by 1000. In a two-phase system it is found by multiplying the volts by the amperes in each phase, adding the two products together, and dividing by 1000. In a three-phase system it is ascertained by multiplying the average volts of the three circuits by the average amperes of the three circuits, multiplying the product by the constant 1.73, and dividing by 1000.

If the power factor is indicated directly by a power-factor meter, readings of this instrument should check with the determinations made in the manner stated.

326 In a three-phase system which is balanced (that is, when there is an equality of amperes and volts in all three phases), the power factor may also be determined by referring to the curve shown in Fig. 18, in which abscissae represent the proportion borne by the readings of the two wattmeters, and ordinates the corresponding power factor.

327 When the power factor in a three-phase system is 100 per cent,

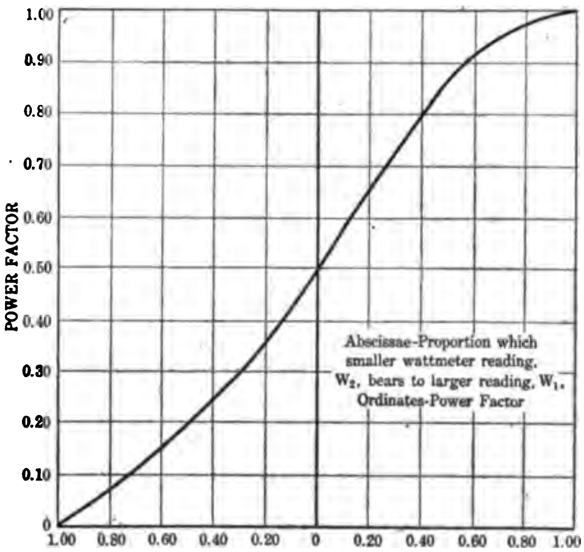


FIG. 18 POWER FACTOR CURVE

both wattmeters indicate the same quantity. When it is less than 100 per cent, one of the meters indicates a larger quantity than the other, the one showing the larger quantity giving positive readings, and the other either positive or negative readings. The latter become negative when the power factor falls below 50 per cent. To determine whether the smaller reading is to be considered negative or not, disconnect the potential circuit of the meter showing the smaller reading from the cable Y (referring to Fig. 17), and connect it to the opposite outside cable so that this circuit will bridge the two outside cables. If the meter reverses, i.e., if the index hand falls back

to the zero point and rests against the stop, the readings taken with the meter in its working position must be considered negative. If it does not reverse, the readings are both positive.

328 A watt-hour meter can best be calibrated by reference to an indicating wattmeter, in accordance with the following directions:

Ascertain first the constants and formula pertaining to the watt-hour meter by referring to the dial of the instrument, and, if need be, to the manufacturer's instruction book; and by this means determine the number of watts corresponding to one revolution of the meter disk per second. Then find the actual speed of the disk by observing the time required for a certain number of complete revolutions, using a stop watch, and compute the actual number of revolutions per second; finally, multiplying the revolutions per second by the watts corresponding to one revolution per second, and the result is the output in watts. Read the indicating wattmeter every five seconds during the period that the speed is observed, and average the readings. A comparison of the watts determined by the watt-hour meter observations with this average shows the error of the watt-hour meter.

Calibrations should be made while the main test is going on. If the load is not substantially constant they should be repeated at regular intervals during its progress so as to obtain the comparison under average conditions.

APPENDIX NO. 22

WATER RHEOSTATS

329 A simple form of water-rheostat, suitable for absorbing the output of a 500-kw. generator of the three-phase alternating-current type, consists of three wine casks, one for each phase, placed side by side, through each of which a sufficient amount of cooling water is allowed to flow to prevent boiling. Salt is added to the water as may be needed to lower its resistance. A fixed terminal is provided consisting of three connected iron plates, one of which is laid on the bottom of each cask. A movable terminal is provided for each phase, consisting of a piece of 2-in. standard iron pipe, 4 to 6 ft. in length, suspended vertically in the water at the center of each cask. Each pipe is arranged to be raised or lowered at will so as to regulate the depth to which it is immersed and thereby the output for that phase. By varying the relative immersion of the pipes, an equal distribution of the current may be obtained between the three phases.

330 Another arrangement consists of a tank, the size of which equals that of the three casks combined, the pipes being located suitable distances apart and a single plate being used for the fixed terminal.

331 When a larger amount of current than 500-kw. is generated, the size of the tank and terminals may be proportionately increased, and at the same time provision made for introducing the cooling water so as to flow along the surface of each pipe and overcome or reduce the tendency to local ebullition.

The changes of resistance due to variations in the strength of the salt solution, depth of immersion of the terminals, and temperature of the water, are such that constant attention is required to maintain uniformity in the output with this kind of rheostat. When uniformity is desired without such attention a rheostat may be constructed which embodies the principles referred to in the description of the apparatus used for a direct current rheostat, which is given in ¶9y.

APPENDIX NO. 23

CHART OF BOILER LOG

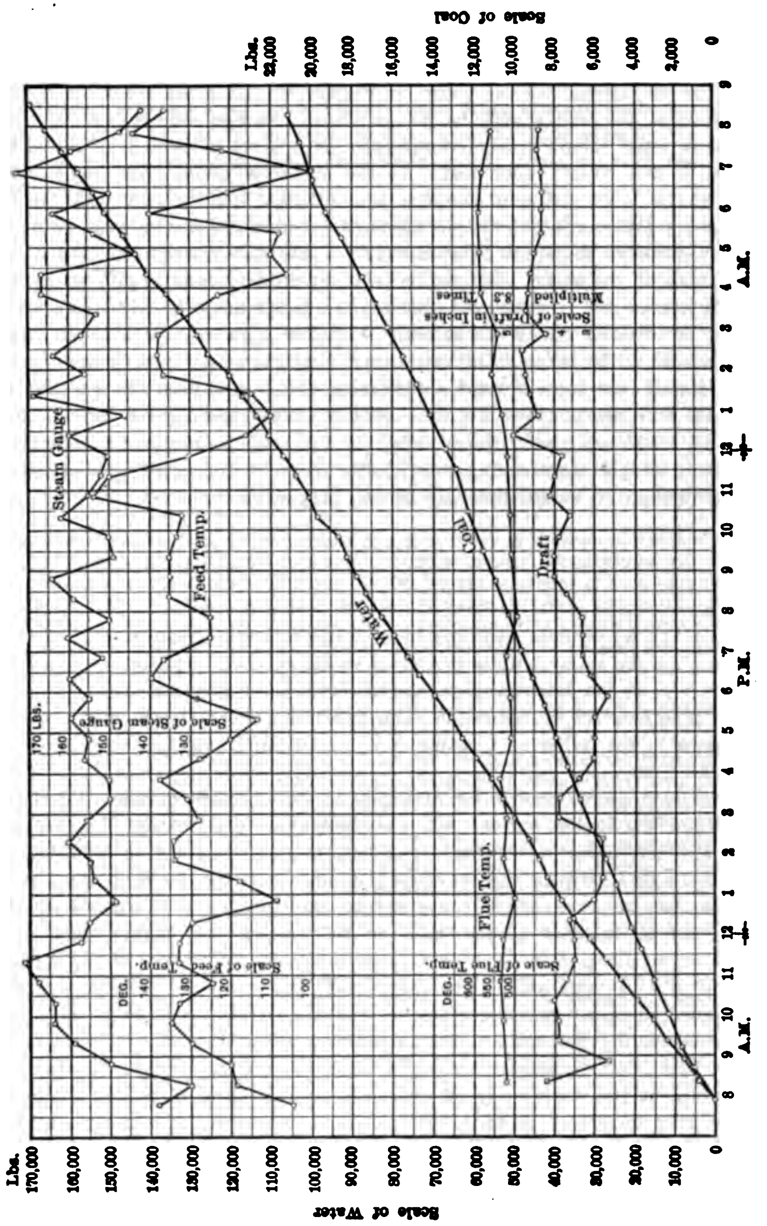


FIG. 19 CHART SHOWING LOG OF BOILER TEST

Reproduced from Trans. Am. Soc. M. E., vol. 21, p. 104, fig. 7.

APPENDIX NO. 24

LOCATION OF INSTRUMENTS FOR BOILER TESTS

332 The feedwater thermometer should be placed in a thermometer well and inserted in the feed pipe. Except in cases where an injector is used, the point selected should be as near as practicable to the boiler. Where an injector is employed, and the water is weighed or measured before it is supplied thereto, the well should be placed on the suction side of the injector, and the injector should receive steam through a short covered pipe connected directly to the boiler under test. If the steam is taken from some other source and it is of different pressure and different quality from that of the boiler under test, correction should be made for such difference, and especially for any excessive moisture thus introduced into the feedwater. When the temperature of the water changes between the injector and boiler, as by the use of a heater or by excessive radiation, the temperature at which the water not only enters and leaves the injector, but that also at which it enters the boiler, should all be taken. In that case, the weight to be used is that of the water leaving the injector, computed from the heat units, if not directly measured, and the temperature, that of the water entering the boiler. The weight of condensed steam to be added to the weight of water entering the injector, to obtain that leaving the injector, may be computed approximately by multiplying the weight entering by the proportion

$$\frac{h_3 - h_1}{h_2 - h_3}$$

in which

h_1 = heat units per lb. of water entering injector.
h_2 = heat units per lb. of steam entering injector.
h_3 = heat units per lb. of water leaving injector.

333 The location of the steam calorimeter and steam thermometer should be as close to the boiler as possible, keeping in mind the directions given in ¶ 28 to 31.

334 Draft gages should be attached to each boiler between the hand damper and the boiler, and as near the damper as practicable. In the case of a plant containing a number of boilers, a gage should also be attached to the main flue between the regulating damper and the boiler plant. It is desirable also to have gages connected to the furnace or furnaces of the boilers, and in cases of forced blast, to the ashpits and blower ducts. If there is an economizer in the flue a gage should be connected to the flue at each end of this apparatus. The same draft gage may be used for all the points noted, provided suitable pipes are run from the gage to each, arranged so as to be readily connected to either point at will.

335 The flue thermometer should be located where it will show the average temperature of the whole body of gas. For an extremely large flue the thermometer may be placed in an oil pot of small diameter, which is suspended in the flue, and the thermometer lifted partially out of the oil when the temperature is read.

336 The sample for flue gas analysis should be taken as noted in ¶ 33.

APPENDIX NO. 25

DETERMINATION OF HEAT CONSUMPTION OF STEAM PLANT

337 The measurement of the heat consumption used in the calculations for a plant test requires the measurement of each supply of feedwater to the boiler—that is, the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and water supplied by gravity or other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam.

338 The temperatures should be those applying to the working conditions. It frequently happens that the measurement of the water requires a change in the usual temperature of supply. For example, where the main supply is ordinarily drawn from a hot-well in which the temperature is, say 100 deg. fahr., it may be necessary, owing to the low level of the well, to take the supply from some source under a pressure or head sufficient to fill the weighing tanks used, and this supply may have a temperature much below that of the hot-well, possibly as low as 40 deg. fahr. The temperature to be used is not the temperature of the water as weighed in this case, but that of the working temperature of the hot-well. The working temperature in cases like this must be determined by a special test, and included in the log sheets.

In determining the working temperatures, the preliminary or subsequent test should be continued a sufficient time to obtain uniform indications, and such as may be judged to be an average for the working conditions. In this test it is necessary to have some guide as to the quantity of work being done, and for this reason the power developed by the engine should be determined by obtaining a full set of diagrams at suitable intervals during the progress of the trial, and in a turbo-generator the output of the generator should be observed. Observations should also be made of the gages connected with the plant and of the water levels in the boilers, the latter being maintained at a uniform point so as to be sure that the rate of feeding during the test is not sensibly different from that of the main test.

When the feedwater is all supplied by one feeding apparatus, the temperature to be found is that of the water in the feed pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejected heat of the engine or turbine, such as that contained in the exhaust steam either of the main cylinders or turbine or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feedwater supply, one the main supply from the hot-well, or from some other source, and the other an auxiliary supply derived say from the water condensed in the jackets of the main engine and in the live-steam reheater if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot-well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

339 The heat to be determined is that used by the entire engine or turbine equipment, embracing the main cylinders or turbine and all auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, and feed pumps, also the jacket and reheater when these are used.

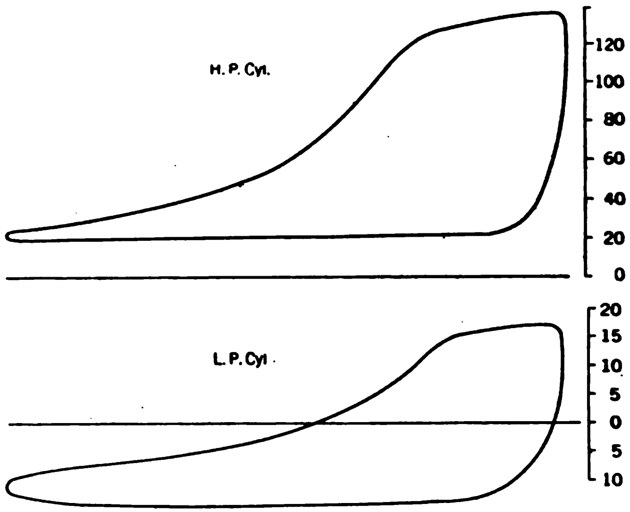


Fig. 20 Actual Diagram Receiver Engine

340 The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of steam used by the calorimeter must be determined and properly allowed for.

APPENDIX NO. 26

THE COMBINED DIAGRAM

341 The Combined Diagram is a hypothetical figure, which in its essential features represents an indicator diagram which would be obtained if the whole process of admission, expansion and exhaust occurred in one cylinder, viz., the low-pressure cylinder. It is a diagram from which the pressure of the steam at any point in the stroke of either cylinder, and the volume of that steam can be measured from one diagram, in the same manner that it can be measured in the case of a single cylinder engine from the actual indicator diagram.

The general method of laying out a combined diagram is shown clearly in Figs. 20 to 23*, the first of which refers to a Corliss compound engine (receiver type) in which the cylinder ratio is 3.72, and the clearances 4 per cent and 8 per cent, respectively; and the second to a Westinghouse compound engine (Woolf type), in which the cylinder ratio is 2.72, and the clearances, 33 per cent and 9 per cent, respectively.

APPENDIX NO. 27

DIAGRAM FACTOR

342 The diagram factor is the proportion borne by the mean effective pressure measured from the actual diagram to that of a hypothetical diagram which represents the maximum power obtainable from the steam accounted for by the actual diagram at the point of cut-off; assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during admission or release; third, that the expansion line is a hyperbolic curve; and, fourth, that the initial pressure is that of the boiler, and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

The method of determining the diagram factor is illustrated in Figs. 24 to 27*, which apply to a simple non-condensing engine, a simple condensing engine, and a compound condensing engine. In Fig. 24 *RS* represents the volume of steam at boiler pressure admitted to the cylinder, *PR* and *OS* being hyperbolic curves drawn through the compression and cut-off points respectively. In Fig. 25 the factor is the proportion borne by the area of the actual diagram to that of the diagram *CNHSK*. In Fig. 26 the factor is the proportion borne to the area of the diagram *CNHSK*. In Fig. 27 the factor is the proportion borne by the area of the two combined diagrams to the area *CNHSK*. In Fig. 25 where the diagram is the same as in Fig. 24 the distance *CN* is laid off equal to *RS* shown in Fig. 24, and the curve *NH* is a hyperbola referred to the zero lines *CM* and *MJ*. In Fig. 26 the distance *CN* is found in a similar way. In Fig. 27 the distance *CN* for the high-pressure cylinder is found in the same manner as in the case of a simple engine. The mean effective pressure of the ideal diagram can readily be obtained from the formula

$$\frac{P}{R}(1+\text{hyperbolic logarithm } R)-p$$

where

P = absolute pressure of steam in boiler

$R=\frac{MJ}{CN}$

p = pressure of atmosphere or that in condenser

APPENDIX NO. 28

HEAT CONSUMPTION OF IDEAL ENGINE OR TURBINE CONFORMING TO RANKINE CYCLE

343 In an engine working according to the Rankine cycle, steam is admitted at constant pressure, expanded adiabatically to the back pressure, and exhausted at that pressure. The engine has no clearance and there are no heat losses, or losses from friction, imperfect expansion, or otherwise, all the en-

* Reproduced from Trans. Am. Soc. M. E., vol. 24, pp. 746, 748, 753, 754, figs. 123-126, 130-133.

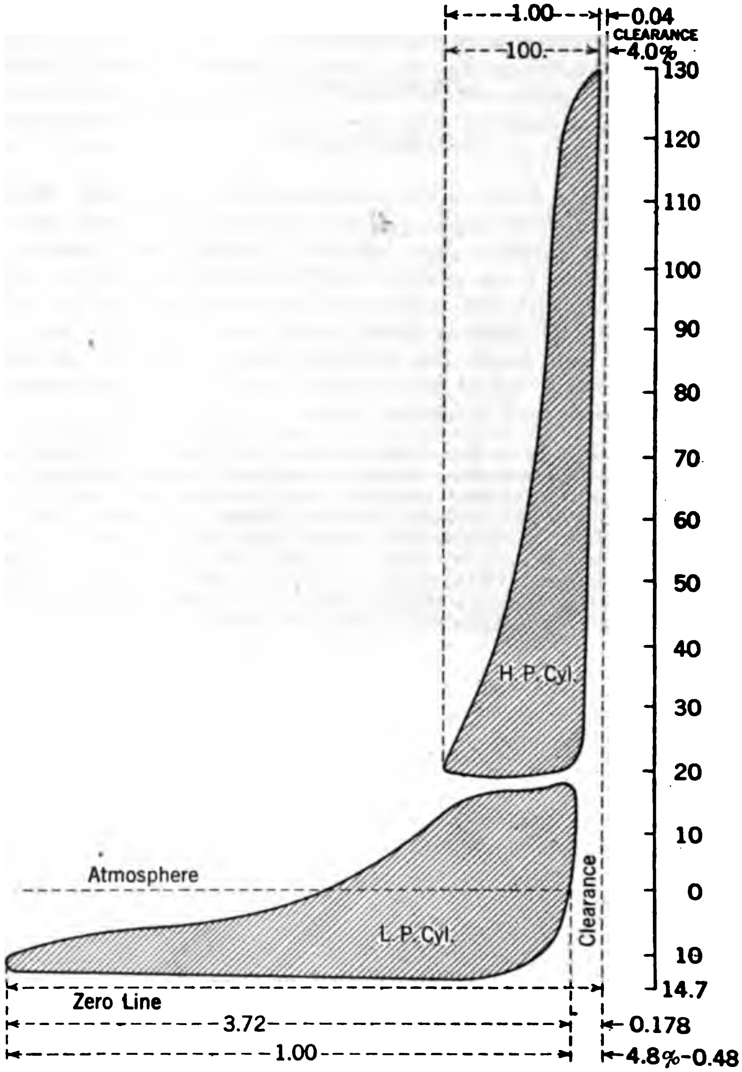


FIG. 21 COMBINED DIAGRAM RECEIVER ENGINE

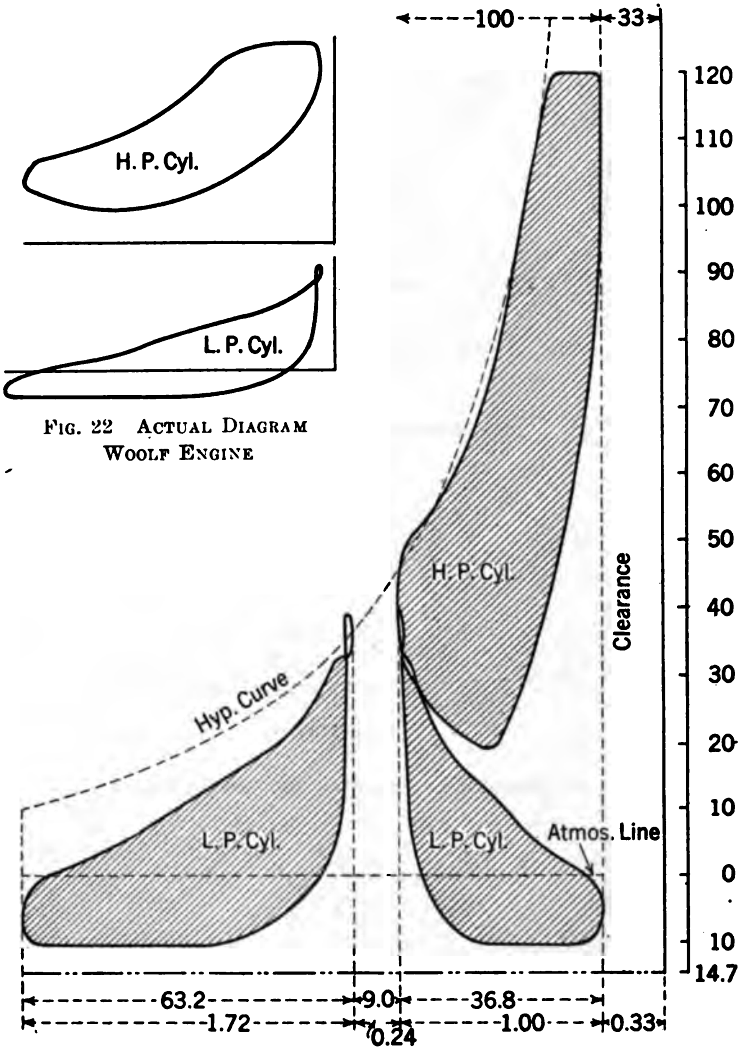


FIG. 22 ACTUAL DIAGRAM WOOLF ENGINE

FIG. 23 COMBINED DIAGRAM WOOLF ENGINE

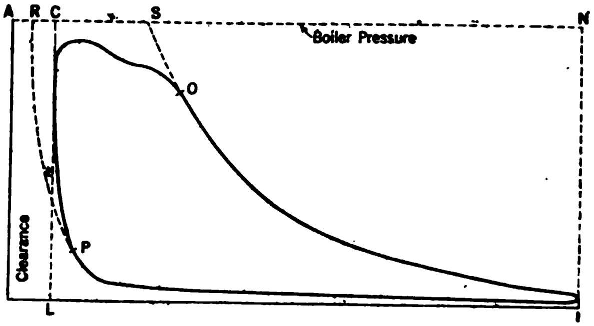


Fig. 24 Diagram Factor Net Volume

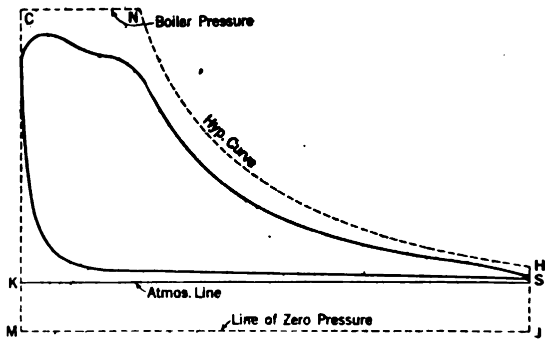


Fig. 25 Diagram Factor Non-Condensing Engine

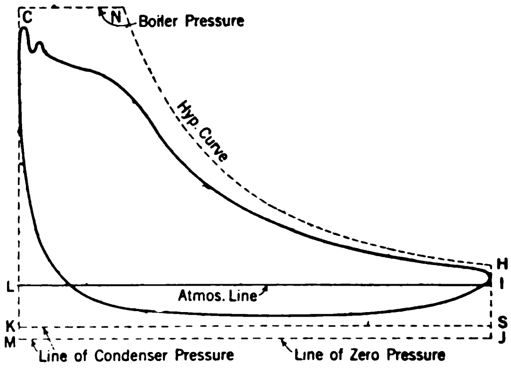


Fig. 26 Diagram Factor Condensing Engine

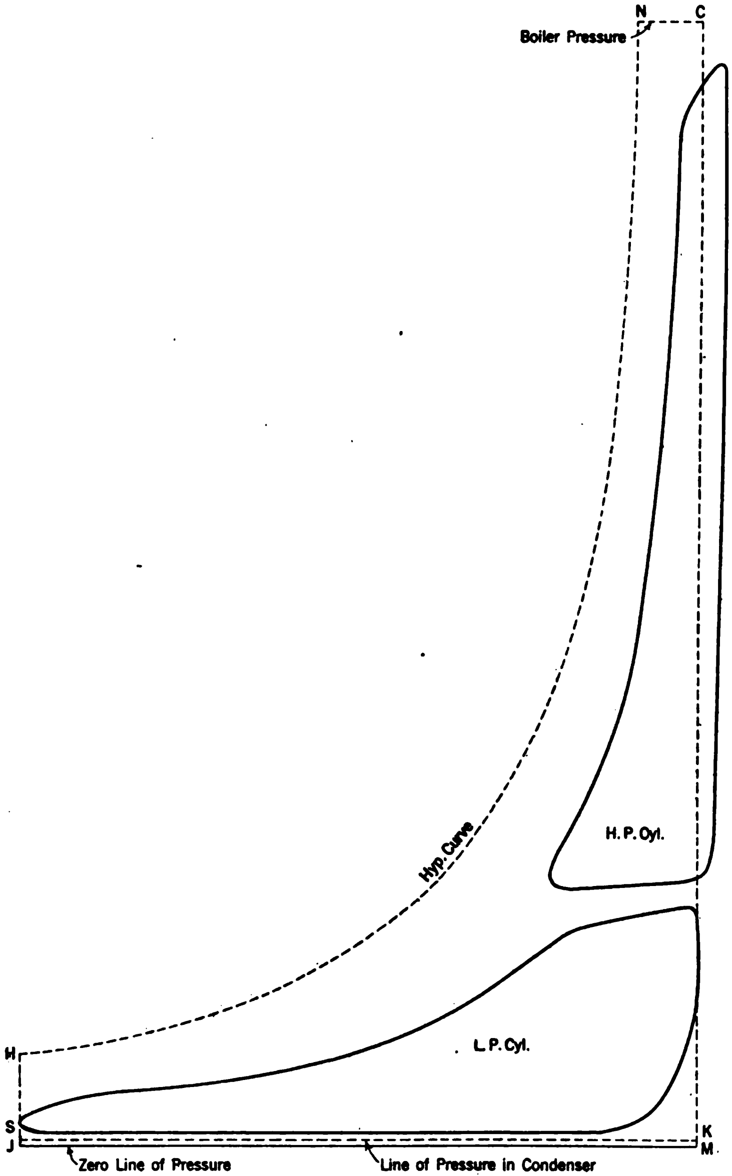


FIG. 27 DIAGRAM FACTOR COMPOUND ENGINE

ergy taken from the steam being converted into work. The heat consumption per lb. of steam in such an engine is the difference between the total heat per lb. of the entering steam and that of 1 lb. of water at the temperature of the exhaust steam. The heat utilized by such an engine per lb. of steam is the difference between the total heat per lb. of the entering steam and that of the exhaust steam (or mixed steam and water), both being taken with the same entropy. The calculation can be most readily made by reference to the "total heat-entropy diagram" devised by Mollier, given in the volume of "Steam Tables and Diagrams" by Marks & Davis, and in Meyer's "Steam Turbine."

344 In using the heat-entropy diagram for this purpose, find the heat units per lb. of steam at the observed initial pressure and temperature, and note the entropy of the steam. Find also the heat units per lb. of steam (or mixed steam and water) of the same entropy at the observed back pressure. The difference between the two is the heat utilized by the ideal engine per lb. of steam.

345 The same result may be found without the aid of the diagram by taking the difference in the heat units per lb. of steam at the higher pressure (saturated or superheated as the case may be) and per lb. of saturated steam at the lower pressure, and adding thereto the product of the absolute temperature at the lower pressure by the difference of entropy at the two pressures.

346 Divide 2546.5 by the B.t.u. utilized in the ideal engine per lb. of steam (¶ 343) and the quotient is the number of lb. of steam it consumes per h.p-hr. Multiply this quotient by the difference between the total heat per lb. of steam supplied and the total heat in a lb. of water at the temperature of the exhaust and the product is the total heat consumption of the ideal engine per h.p-hr.

347 The formula for the efficiency of the Rankine cycle is

$$\frac{H_s - H_2 + T_2(N_2 - N_1)}{H_s - h}$$

in which

H_s = total heat of 1 lb. steam at the throttle
H_2 = total heat of 1 lb. steam at the exhaust
T_2 = absolute temperature of the exhaust
N_1 = entropy of 1 lb. steam at the throttle
N_2 = entropy of 1 lb. steam at the exhaust
h = heat of 1 lb. of feed water at the temperature of the exhaust

APPENDIX NO. 29

LEAKAGE TEST OF INSIDE PLUNGER PUMP

348 The leakage of an inside plunger is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

349 Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the wooden head. The test should be made, if possible, with the plunger in various positions.

350 In a case where it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings provided for the inspection of the suction valves, the head remaining in place.

351 It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

352 If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

APPENDIX NO. 30

LOCOMOTIVE TESTING STATIONS

353 The laboratory test described in the Locomotive Code cannot readily be made unless a testing station is available where the locomotive may be sent, and where this work may be carried on. Such a station is too expensive to be constructed for the purpose of ascertaining the performance of an individual locomotive. How to arrange and install a laboratory testing plant seems therefore a matter for independent consideration.

354 A number of universities have these plants in use or in process of installation, a notable instance being the Purdue University, which was the first to inaugurate the laboratory test.

355 The most elaborate work that has recently been done in the laboratory testing line is that which was carried on in 1904 by the Pennsylvania Railroad Company at the Louisiana Purchase Exposition, St. Louis. A full description of the plant there installed, and of the tests made upon it, may be found in a volume published in 1905 by that company, entitled "Locomotive Tests and Exhibits." A study of this book is commended to any who require detailed information regarding the design, construction, and operation of laboratory testing apparatus. The plant and tests referred to were planned under the counsel of an advisory board, three of whom were members of the Society.

APPENDIX NO. 31

LOCATION OF APPARATUS AND INSTRUMENTS FOR LOCOMOTIVE TESTS

356 The location of apparatus and instruments in a laboratory test does not need to conform to restrictions of space such as pertain to road tests, and substantially the same considerations apply in that case as in the ordinary stationary steam plant. Reference may therefore be made to the Boiler and Engine Codes for any needed instructions in this matter.

357 As to the locations and requirements for road tests, the following directions should be observed:

(*a*) The water meter should be attached to the suction pipe of the injector at a point where it can be conveniently read when the train is in motion. A check valve should be provided to prevent hot water backing through it when starting and stopping the injector, as also a strainer.

(*b*) The indicator driving rig should be some form of pantograph motion, with a light tube transmitting the reduced motion to a point near the indicator. Lack of room and facility of operation make it desirable to use a single indicator for each cylinder, connected to the two ends by a three-way cock. It is best to carry the pipes to the side of the cylinder rather than the heads, and provide a branch leading to the steam chest, sharp bends being avoided, and the outside protected from radiation. Absolute rigidity of the indicator cock is essential, and it should be obtained by clamping it securely to the cylinder.

(*c*) A special steam gage should be employed which can be read by the observer stationed in the pilot box at the front end.

(*d*) A rod should be attached to the reverse lever and carried forward to the pilot box, where a scale is provided to show its position. A rod should likewise be connected to the throttle-valve lever for the same purpose.

(*e*) The draft gage may be a simple U-tube containing water, and it should be connected to both front and back of the diaphragm at the centre of the smoke box and to the ash pan.

(*f*) The flue thermometer or pyrometer should be inserted so that the bulb occupies a position midway between the table plate and bottom of smoke arch.

APPENDIX NO. 32

FURNACE EFFICIENCY

358 Attempts have been made to separate the combined efficiency of boiler, furnace, and grate into two parts, viz., efficiency due to boiler alone, and efficiency due to furnace (including grate), but there is no agreement as to the exact line of demarcation to be used in separating one from the other. The following paragraphs (359-365) show the impossibility of making an exact separation.

359 The heat losses chargeable to the furnace alone are clearly those designated *a, b, c,* and *d* in the following list:

(*a*) Loss due to unburned solid fuel dropping through the grates or withdrawn from the furnace, including the solid combustible matter in the cinders, sparks, flue dust, etc.

(*b*) Loss due to the production of CO instead of CO_2.

(*c*) Loss due to escape of unburned volatile hydrocarbons.

(*d*) Loss due to the combination of carbon and moisture and production of hydrogen (by the reaction $C + H_2O = CO + 2H$) when fresh moist coal is thrown on a bed of white hot coke.

360 The remaining heat losses, which are those due to heat carried away by the air and moisture in the escaping gases, loss from radiation, and losses unaccounted for, may be divided as given below in Items *e* to *j*.

(*e*) Moisture losses; embracing evaporation of moisture and heating of steam thus formed to T_P (T_P = temperature corresponding to boiler pressure).
- (1) Moisture in coal
- (2) Moisture in air
- (3) Moisture due to burning hydrogen in the fuel

(*f*) Moisture losses consisting in the further heating of steam of Item $360e_3$ from T_P to T_g (T_g = temperature of escaping gases).
- (1) Moisture in coal
- (2) Moisture in air
- (3) Moisture due to H

(*g*) Theoretical air supply losses
- (1) Heated to T_P
- (2) Heated from T_P to T_g

(*h*) Excess air supply losses
- (1) Heated to T_P
- (2) Heated from T_P to T_g

(*i*) Radiation
- (1) Due to furnace
- (2) Due to boiler

(*j*) Unaccounted for losses
- (1) Due to furnace
- (2) Due to boiler

361 It has been suggested that these losses be grouped and apportioned as follows:

U = unavoidable losses $= e_1 + e_2 + e_3 + g_1$
F = furnace losses $= a + b + c + d + i_1 + j_1$

B = boiler losses $= f_1 + f_2 + f_3 + g_2 + h_1 + h_2 + i_2 + j_2$

The efficiencies based on this apportionment of the losses may be expressed as follows:

(1) Maximum theoretical efficiency based on utilizing all the heat excepting unavoidable losses $= \dfrac{00-U}{100}$

(2) Furnace and grate efficiency based on heat available after deducting unavoidable losses $= \dfrac{100-(U+F)}{100-U}$

(3) Boiler efficiency based on heat available after deducting unavoidable losses and those due to furnace and grate $= \dfrac{100-(U+F+B)}{100-(U+F)}$

(4) Efficiency of boiler, furnace and grate, based on total heat of combustion of fuel [Product of (1), (2) and (3)] $= \dfrac{100-(U+F+B)}{100}$

362 These formulae do not, however, furnish a method of determining the true individual efficiencies desired, because it is impossible to determine Item *d*, and impracticable to obtain Item *c* with the gas-testing appliances ordinarily available. It is impossible also to separate the losses i_1 and j_1 attributed to the furnace, from the boiler losses alone due to radiation and those unaccounted for.

363 Another suggestion is to transfer the excess air loss h_1 to the group of furnace losses F; but this makes the matter even worse, inasmuch as the furnace efficiency is then dependent on the steam pressure in the boiler, which is a matter foreign to any furnace condition. It further assumes that the flue gases cannot be cooled below the temperature due to the pressure, which although true for many types of boiler, is not true in cases where the contraflow principle is used.

364 A third method suggested is to include among the boiler losses all those which have been classed above as unavoidable. By this method the furnace efficiency is

$$\frac{100-F}{100}$$

and the boiler efficiency

$$\frac{100-(U+B+F)}{100-F}$$

365 If it is desired to divide the combined efficiency between boiler and furnace in some such manner as those suggested, the method of division employed should be clearly stated.

366 In the case of stoker-fired boilers requiring efficiency guarantees, such guarantees may be made conditional on the stoker burning the necessary amount of a specified coal and maintaining not less than a specified percentage of carbon dioxide in the gases leaving the furnace, and not more than specified percentages of carbon monoxide in the gases and combustible in the ashes. For example, the specified percentages might be an average of 13 per cent CO_2 and 0.2 per cent CO in the gases, and 25 per cent combustible in the ash.

APPENDIX NO. 33

CALCULATION OF HEAT BALANCE FOR BOILER TEST

367 DATA: Semi-bituminous coal, 2 per cent moisture, 8 per cent ash, 90 per cent combustible, 82 per cent C, 4 per cent H, 3 per cent O, 1 per cent N.

B.t.u. per lb. combustible 15,800; per lb. coal as fired, 14,220.

Ash and refuse by boiler test, 13 per cent, referred to coal as fired. The 13 per cent ash and refuse is assumed to contain the 8 per cent of ash shown by the analysis and 5 per cent of combustible.

Efficiency of boiler, furnace, and grate, based on coal as fired, 70 per cent.

The gas analysis shows that 20 lb. of air is supplied per lb. of C burned; and that 0.05 lb. of the carbon burned was burned to CO.

The air is supplied at 92 deg. fahr., and contains 0.02 lb. of water vapor per lb. of dry air (60 per cent relative humidity). Flue gas temperature 592 deg. fahr.

Water from and at 212 deg. per lb. coal as fired 10.258; per lb. of dry coal, 10.467; per lb. of combustible, 12.068.

Referred to	Coal			Combustible	
	B.t.u. per lb. of coal as fired	B.t.u. per lb. dry coal	Per cent	B.t.u. per lb. combustible	Per cent
(a) Heat absorbed by the boiler (Item 39, 40 or 41 ×970.4).....	9,954	10,157	70	11,711	74.1
(b) Loss due to evaporation of moisture in coal, 0.02×(212—92) +970+0.47 (592—212).......	25	26	0.2	29	0.2
(c) Loss due to heat carried away by steam formed by the burning of hydrogen, 0.04×9×[(120+970+(0.47×380)]	457	466	3.2	538	3.4
(d) Loss due to heat carried away in the dry flue gases, 21 lb. per lb. C, 21×0.77×500×0.24.......	1940	1979	13.7	2,282	14.4
(e) Loss due to carbon monoxide 0.05×0.77×10,150...........	391	399	2.7	460	2.9
(f) Loss due to combustible in ash and refuse, 0.05×14,600.......	730	745	5.2	....	....
(g) Loss due to heating moisture in air, 0.02×20×0.77×500×0.47	72	74	0.5	86	0.5
(h) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for.	651	664	4.5	694	4.5
(i) Total calorific value of 1 lb. of coal as fired, dry coal, or combustible Items 42 and 43 and footnote)....................	14,220	14,510	100.0	15,800	100.0

The items are designated by the same numbers as in Table 7, Boiler Code.

368 If the fuel lost in ash and refuse is not the combustible of the original coal, but coke or carbon of a heating value of 14,600 B.t.u. per lb., then the heat loss due to it is $0.05 \times 14{,}600 = 730$ instead of 790 B.t.u. The heating value per lb. of combustible burned would then be $(14{,}220 - 730) \div 0.85 = 15{,}870$ instead of 15,800. The percentage figures in the last column would be changed accordingly, and the efficiency of the boiler and furnace would be $(11{,}711 \div 15{,}870) = 73.8+$ per cent instead of 74.1 per cent.

369 In this table the calculations expressed in the test (excepting Item *a*) refer to the quantities given in the first column, which are based on coal as fired. The quantities in the second column, which are based on dry coal, are obtained from those in the first column by dividing each one by

$$\frac{100-2}{100} = 0.98$$

APPENDIX NO. 34

EXAMPLE FOR CALCULATING STEAM ACCOUNTED FOR BY INDICATOR IN TRIPLE EXPANSION ENGINE

Calculations Made for Cut-off Points

370 DATA: Cylinder Ratios 1 to 2.94 to 6.5.
Proportion of Clearance each cylinder (E) 0.025

	Which cylinder	H.P.	Int. P.	L. P.
(1)	Pressure at point near cut-off above zero ....lb. per sq. in.	145.2	38.7	16.0
(2)	Weight of one cu. ft. of saturated steam at cut-off pressure noted (Wc).......lb.	0.3217	0.0924	0.0404
(3)	Pressure at point near compression above zero ....lb.	46.8	20.7	2.3
(4)	Weight of one cu. ft. of saturated steam at compression pressure noted (Wh)....lb.	0.1105	0.0514	0.0066
(5)	Proportion of direct stroke completed at cut-off point (C)...................	0.346	0.406	0.357
(6)	Proportion of return stroke uncompleted (H)	0.006	0.008	.0
(7)	M.e.p. ....lb. per sq. in.	60.56	13.22	10.16

371 The combined m.e.p. referred to the H.P. cylinder is $60.56 + (2.94 \times 13.22) + (6.5 \times 10.16) = 165.47$. That referred to the intermediate cylinder is

$$13.22 + \frac{60.56}{2.94} + 10.16 \times \frac{6.5}{2.94} = 56.28$$

That referred to the L.P. cylinder is

$$10.16 + 13.22 \times \frac{2.94}{6.5} = \frac{60.56}{6.5} = 25.46$$

372 Substituting in the formula given in ¶ 77g we have for the steam accounted for at cut-off per i.h.p-hr. in the various cylinders the following:

$$\text{H.P. Cyl.} = \frac{13{,}750}{65.47}[(0.346+0.025)\times 0.3217-(0.006+0.025)\times 0.1105] = 9.63$$

$$\text{Intermediate Cyl.} = \frac{13{,}750}{56.28}[(0.406+0.025)\times 0.0924-(0.008+0.025)\times 0.0514] = 9.32$$

$$\text{L.P. Cyl.} = \frac{13{,}750}{25.46}[(0.357+0.025)\times 0.0404-(0.025\times 0.0066)] = 8.24$$

APPENDIX NO. 35

EXAMPLE OF HEAT BALANCE FOR STEAM POWER PLANT TEST

373 DATA:

Boiler pressure	120	lb. per sq. in.
Water per lb. coal as fired	9.95	lb.
Water per i.h.p-hr	13.0	lb.
Coal as fired per i.h.p-hr	1.306	lb.
Temperature of water entering economizer	165	deg.
Temperature of water leaving economizer	220	deg.
Temperature of gases entering economizer	592	deg.
Temperature of gases leaving economizer	392	deg.
Efficiency of boiler, furnace, grate and economizer	74.9	per ct.
Other data, same as Appendix No. 33		

	Per lb. coal as fired	Per cent
(49) Heat units in coal	14,220	100.0
(50) Boiler losses		
(a) Loss due to evaporation of moisture in coal 0.02 × (212 — 92) + 970 + 0.47 (392 — 212)	28	0.2
(b) Loss due to heat carried away by steam formed by the burning of hydrogen 0.04 × 9 × (120 + 970 + 0.47 × 180)	423	3.0
(c) Loss due to heat carried away in the dry flue gases (21 × 0.82 × 300 × 0.24)	1240	8.7
(d) Loss due to carbon monoxide (same as App. 33)	416	2.9
(e) Loss due to combustible in ash and refuse (same as App. 33)	790	5.5
(f) Loss due to heating moisture in air (0.02 × 20 × 0.82 × 300 × 0.47)	46	0.3

	Per lb. coal as fired 14,220	Per cent 100.0
(*g*) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for 14,220 × 100 — 74.9 ÷ 100 — (*a* + *b* + *c* + *d* + *e* + *f*)	631	4.5
(*h*) Heat supplied steam-driven appliances for operating boilers less that recovered by heating feedwater, by test	145	1.0
(*i*) Total boiler losses	3714	26.1
(51) Engine consumption		
(*a*) Radiation from steam pipe (by test or calculation)	125	0.9
(*b*) Radiation from engine or turbine (by calculation)	130	0.9
(*c*) Heat rejected to condenser	8026	56.5
(*d*) Heat withdrawn from engine receivers or turbine stages or for other use than heating feedwater	0	0
(*e*) Heat lost by leakage of steam piping (by test)	125	0.9
(*f*) Heat converted into work (2546.5 ÷ 1.306)	1950	13.7
(52) Heat in steam supplied for purposes foreign to engine or turbine (by test)	150	1.0
Total (same as Item 49)	14,220	100.0

374 The boiler output, 10,506 (Item 49 — Item 50*i*) may be divided into

(50*j*) Heat units absorbed by water in boiler (10.258 × 970.4)	9954	70.4
(50*k*) Heat units absorbed by water in economizer (9,954 × 0.055)	552	3.9

375 The quantity representing the sum of Item 51*b*, *c*, and *f* (10,106 = 71.1 per cent) may be divided according to the steam distribution into

(51*g*) Heat consumed by engine cylinders or turbine alone (including reheaters or jackets, if any) i.e., total heat supplied to engine or turbine alone less heat recovered therefrom by heating feedwater	9793	68.9
(51*h*) Heat consumed by steam-driven auxiliaries, i.e., total heat supplied to auxiliaries less heat recovered therefrom by heating feedwater	313	2.2

376 The same quantity may be divided according to the distribution of work done by engine or turbine into

(51*i*) Heat consumed in supplying power lost in friction of engine or turbine	711	5.0
(51*j*) Heat consumed in supplying frictional, electrical, or other losses of power delivered by engine or turbine shaft	1422	10.0
(51*k*) Heat consumed in supplying useful power delivered by engine or turbine, whether mechanical, electrical or otherwise	7973	56.1

APPENDIX NO. 36

METHOD OF DETERMINING MEAN EFFECTIVE PRESSURE (M.E.P.) FROM INDICATOR DIAGRAMS

377 The simplest and usually the most accurate method of finding the mean effective pressure from an indicator diagram is to employ a planimeter and determine the enclosed area, divide the area in square inches by the length in inches, thus finding the average height in inches, and multiply the quotient by the scale of the spring. The manner of using a planimeter is described in trade catalogues of the instrument.

378 Another method is to divide the diagram lengthwise into ten or more equal parts, draw pencil lines at each point of division perpendicular to the atmospheric line and crossing the lines of the diagram, marking each point of intersection. Then lay the edge of a long strip of white paper, or the edge of an indicator card, on each line one after the other, marking off the enclosed differences one by one until the last one is added. The total length is then measured using the scale of the spring. Divide the total by the number of lines and the result is the approximate m.e.p. expressed in pounds per square inch.

379 To obtain a correct result by the method above described, the first and last distances should be laid off at the beginning of the operation and their sum divided by two. Then the remaining ones should be added to the half thus found and the final total divided by the total number of spaces (one less than the number of lines). In case the diagram is of such form that straight lines joining the various points of division do not form an area which is equal as near as can be judged to the area of the diagram, as may happen at the ends of the diagram or at the cut-off, the points of intersection should be varied sufficiently so that the desired equality will be obtained.

APPENDIX NO. 37

EXAMPLE FOR CALCULATING RESULTS OF AIR COMPRESSOR TEST

380 OBSERVED DATA:

Item 9	Pressure in steam pipe near throttle by gage	135.3	lb. per sq. in.
	H = total heat above 32 deg. (150 lb. absolute pressure)	1193.4	B.t.u.
Item 14	Temperature of steam in exhaust pipe near engine or turbine.............	141.5	deg.
	Absolute temperature corresponding to Item 14, $141.5 + 460 =$	601.5	deg. abs.
	H_1 = total heat in 1 lb. steam at exhaust pressure	1121.6	B.t.u.
	h = total heat in 1 lb. feedwater at temp. 141.5 deg.	109.4	B.t.u.

	$H - h$ = total heat in 1 lb. steam above feedwater temperature	1084.0	B.t.u.
Item 15	Temperature of air in delivery main t = 200 deg. fahr. + 460 deg. fahr...	660	deg. abs.
	(*a*) Temperature of air in engine or turbine room or air supplied to machine 68 deg. fahr. + 460 deg. fahr.	528	deg. abs.
Item 16	Pressure in delivery main by gage (impact pressure) 93.9 + 14.7 (P_1).....	108.6	lb. per sq. in.
	Pressure of atmosphere (P)	14.7	lb. per sq. in.
	$P_1 \div P$	7.39	
	Hyp. log.	2.0	
	Area of delivery main at point where P and t are taken	0.25	sq. ft.
Item 25	Dry steam consumed by engine or turbine per hour	5872	lb.
Item 30	Indicated horsepower of steam end.....	275	h.p.
Item 31	Gross air horsepower as indicated in air cylinders	250	h.p.

381 COMPUTED RESULTS:

Item 32	(*a*) Net air horsepower as computed from Item 35*a* $$\frac{200 \times 108.6 \times 144 \times 2}{33{,}000} =$$	189.5	h.p.
Item 35	Compressed air delivered per min. as measured	250	cu. ft.
	(*a*) Compressed air delivered per min. reduced to atmospheric temperature (250 × 528 ÷ 660).............	200	cu. ft.
	(*c*) Compressed air delivered per min. reduced to atmospheric temperature and pressure (200 × 7.39), (free air)	1478	cu. ft

ECONOMY

Item 36	Heat units consumed per i.h.p-hr. Item 38 ($H - h$) = 21.35 × 1084........	23,147	B.t.u.
Item 37	Heat units consumed per net air h.p-hr. Item 35*c* ($H - h$) = 30.977 × 1084.	33,579	B.t.u.
Item 38	Dry steam consumed per i.h.p-hr. Item 25 ÷ Item 30 (5,872 ÷ 275)......	21.35	lb.
Item 39	Dry steam consumed per net air h.p-hr. Item 25 ÷ Item 32 (5,872 ÷ 189.56).	30.98	lb.

EFFICIENCY

Item 40	Thermal efficiency referred to i.h.p. 2546.5 ÷ Item 36 × 100)........	11.0	per cent
Item 41	Thermal efficiency referred to net air h.p. (2546.5 ÷ Item 37 × 100)....	7.6	per cent
Item 42	Efficiency of compression (Item 32 ÷ Item 31 × 100 = 189.5 ÷ 250 × 100)	75.8	per cent
	(a) Mechanical efficiency of machine (Item 31 ÷ Item 30 × 100)....	91.0	per cent

WORK DONE PER HEAT UNIT

Item 43	Net work per B.t.u. (1,980,000 ÷ Item 37)	58.96	ft. lb.

APPENDIX 38

COAL SAMPLING

382 When great accuracy is desired the methods of sampling used by the U. S. Bureau of Mines (described in B. of M. Technical Paper No. 8, 1913) or of the joint committee of the American Chemical Society and the American Society for Testing Materials (Jour. Indust. and Engrg. Chem., June 1913, Proc. A.S.T.M., 1914) should be followed. The Bureau of Mines has found that the results of determinations of the calorific value of coal depend on the weight of the gross sample; for example, the differences in the results obtained from numerous samples were as follows:

Weight of Gross Sample	Differences B.t.u. per lb. Average	Maximum
100 lb.	251	620
500	75	244
1000	37	94
1300	25	56

383 The Bureau recommends that when prepared by hand the pieces of coal or impurities should be crushed to the following approximate sizes before each reduction.

Weight of Sample to be divided	Size to which coal and impurities should be broken before each division
1000 lb. or more	1 in.
500 lb.	¾ in.
250 lb.	½ in.
125 lb.	⅜ in.
60 lb.	¼ in.

384 A sub-committee of the joint committee of the A. C. S. and A. S. T. M. recommends the following minimum weights of sample to be transmitted to the laboratory:

Size of Largest Impurities	Minimum Weight of Sample
½ in.	75 lb.
⅜ in.	30 lb.
¼ in.	9 lb.
3/16 to 1/5 (4-mesh)	5 lb.
⅛ in.	3 to 5 lb.

385 The sub-committee states that unless special crushing and sampling apparatus is available much moisture is lost during the reduction of the gross sample to the smaller sizes given above. Coal when in a pulverized condition is exceedingly susceptible to change in moisture. The general tendency is loss of moisture on dividing to finer sizes. This may amount to several per cent in coal that has not been previously air dried. The equilibrium point varies with the temperature and humidity.

The use of an air tight grinding apparatus such as the ball mill is essential for the final pulverization.

386 Mr. W. F. Hillebrand, of the sub-committee, states that it is needless to strive at present in ordinary work for a very high degree of refinement in the determination of moisture. So sensitive are coals to humidity changes of the air that it is only by chance that two or more analysts will reach the same results for moisture in a given coal. Allowable variations are as follows:

	Same Analyst	Different Analysts
Moisture under 5 per cent.......	0.2%	0.3%
Moisture over 5 per cent........	0.3%	0.5%

No. 1527

TRIBUTES TO FREDERICK W. TAYLOR

The American Society of Mechanical Engineers, as a tribute to Mr. Frederick W. Taylor, Past-President, Am. Soc. M. E., at the Annual Meeting in December 1915 suspended its regular business at eleven o'clock for a period of one hour, during which a Memorial Meeting was held.

It was the desire of the Society, after receiving the distressing news of Mr. Taylor's untimely death on March 21, 1915, to show honor to his memory in some fitting way, and in May the Council appointed a Committee, for the purpose of preparing a memorial, consisting of four Past-Presidents, Henry R. Towne, Chairman, John R. Freeman, Frederick R. Hutton, and Oberlin Smith.

This committee learned later of the service in memory of Mr. Taylor to be held in Philadelphia in October by the Society to Promote the Science of Management. An account of this remarkable service, which was conducted on Friday, October 22, in Houston Hall, University of Pennsylvania, at which many leaders in management and personal friends of Mr. Taylor were present, was given in the November 1915 issue of The Journal.

At the Annual Meeting, Mr. Towne said it was the intention of his committee to include in its report an account of the proceedings of the Philadelphia service, and that it would ask leave from the Society to print the full report of the committee in Transactions as a lasting tribute to Mr. Taylor's memory.

Later it was decided that, as the addresses at the Philadelphia meeting were to be printed in full in a Memorial Volume by the Society to Promote the Science of Management, and as certain speakers at both meetings were the same, to avoid unnecessary duplication they should be abstracted and combined with the addresses at the Memorial Meeting of this Society.

Mr. Towne gave a brief but comprehensive resumé of Mr. Taylor's career, and was followed by Henry L. Gantt, Past Vice-

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Collated and Edited by H. K. Hathaway, Mem. Am. Soc. M. E.

President, Am. Soc. M. E., and Rear Admiral Caspar F. Goodrich, quotations from whose addresses are made further on. Mr. Towne was one of those first to understand and appreciate Mr. Taylor's work and ideals, to which for years he lent his sympathetic support.

The following is quoted from Mr. Towne's address:

"Science is defined as 'knowledge gained by systematic observation, experiment and reasoning.' That is the Baconian system, and this was followed by Mr. Taylor throughout his whole professional career. He not only aimed to acquire knowledge, but did acquire it, and out of the knowledge so acquired he created a new science, and, emphatically, he is entitled to be classed as a scientist as well as an engineer. Mr. Taylor's achievements were not the result of sudden impulse or inspiration,—they were the fruition of years of patient investigation, study, toil, experimentation and intense application of his admirable reasoning faculties. One of the most notable of his traits to all who have followed his career and to those of us who knew him was the endless pertinacity with which he pursued an object to which he had once turned his serious attention, and the work which he did was ever pursued in the true spirit of research.

'Facts, not opinions' is a motto which peculiarly applies to the life and methods of Mr. Taylor. He was ever seeking to get facts, to understand those facts correctly, to analyze them, and to apply them to useful ends. That is true throughout the whole of his wide range of work, but above all to his study of facts concerning human nature and their application to industrial work and management.

True to the spirit of the scientist, Mr. Taylor never rushed into print. Like Charles Darwin, before giving the world the results of his studies and discoveries he waited long years to assure himself of the facts and that his deductions from them were sound. He began his career at Midvale in 1878. His first public utterance on the subject of industrial management was in 1886, and then only as a discussion of a paper presented by Metcalf. His first paper on the piece-work system was not presented until 1895, after fifteen years of investigation and study, and his classic paper on The Art of Cutting Metals, presented in 1906, came out only after twenty years of research. He spoke only when he had a message, and then always with absolute modesty. One could always feel that the man was not presenting himself, but his topic, and that that possessed him, that he was trying to present what he believed to be a new and interesting truth for the world to consider.

Mr. Taylor's major publications constitute a remarkable list. His first in 1886 on The Relative Value of Water Gas and Producer Gas for Open Hearth Furnaces, was a short publication of only eight pages. In 1893 came his Notes On Belting, the most comprehensive address on the subject the world had ever had up to that date, which was accepted as the basis for all further studies and investigations, and is practically the authority today on all the phases of the subject to which it relates. In 1895 was issued his Piece-work System, the first paper on the subject. In 1899 he and Maunsel White wrote a short paper of great interest and importance on Colors of Heated Steel. In

1903, he brought out his first elaborate paper on the subject of Shop Management, and for the first time presented to the world what we now call commonly the "Taylor System". In 1905 was issued his book on Concrete of which a third edition is in press today, and in 1906 came his paper entitled On the Art of Cutting Metals, 246 pages, which was epochal in its significance and importance, and finally in 1911 his complete gathering together of his Facts and Arguments on the Subject of the Principles of Scientific Management, a book of 144 pages. What a record for twenty-five years! How wide the range of subjects, how long and thorough the study, and how complete his summing up of them!

To whatever he turned his thoughts and his hands he gave concentrated attention, and in all cases he improved where he did not wholly originate. His range of work covered many fields and is an interesting chapter of human experience.

As is quite generally known, Taylor was, up to the time of his death, a great enthusiast on the game of golf, and ranked as a player above the average. His interest in this sport, coupled with his passion for research, led to his making extensive experiments in the art of growing grass for putting greens, and this in turn led him to develop some rather revolutionary theories and practices in the cultivation of plants. Ordinarily, to develop a putting green requires a great many years of careful cultivation, and to keep it in good condition under all the varying weather conditions is under ordinary circumstances almost impossible. The method developed by Taylor permitted a green to be in shape for playing equal to the condition of the best greens several years old in about six months time, and almost eliminated the difficulties of keeping it in first class shape. This was entirely due to a scientific method of constructing the foundation and the soil, and finally of planting the seeds in exactly the right number to a given area which would permit it to grow and thrive in a manner entirely impossible under the ordinary conditions of seeding.

The records of this achievement, which made Mr. Taylor rank as the authority on the subject, were published in *American Golfer* and *Country Life in America.*

In 1901 he withdrew from salaried positions of all kinds. He had won a competence and was satisfied with it. His ambition lay not in that direction but in the carrying forward and upward of the work in which he had been so deeply interested and with which he was so closely identified, and he devoted himself from that time to the end of his life, without compensation, indeed even at the expenditure of much of his private means, to spreading the gospel of better industrial management.

Mr. Taylor's was an abounding life. From youth to manhood and maturity it was brimming over with mental activity, an activity that flowed into many channels, but in each of them concentrated sooner or later upon some definite, specific object. He followed each object until a solution of the problem it presented had been worked out, and worked out so completely as to satisfy him, to make him willing to present the solution to the world.

As to the "Taylor System," the achievement with which his name is chiefly identified, there are many misconceptions. Most truly it is a science, and a new science. It is perfect in its simplicity and completeness. It is not piece-work, it is not differential rates, nor improved tools, nor time studies, nor routing plans, nor functional management, but it is all of these coördinated and combined, and above all, it aims to effect the uplift of the worker. Mr. Taylor aimed to show the workingman how to apply his time and efforts to better advantage and to accomplish more work than before in shorter time with less effort.

Sooner or later this fundamental fact in Mr. Taylor's work will be appreciated, and then the leaders of organized labor who, unwittingly, I like to think, thus far are opposing the introduction of the "Taylor System", not only into government plants but into private plants as well, will see their error and will lend their influence to promoting the introduction of a system which will do more than anything else that has been proposed in our day and generation for the uplift of labor.

Mr. Taylor saw the inequity of the old system of piece-work compensation, under which increased efficiency and output, instead of being rewarded, were punished by a reduction of the piece-rate. I partly grasped that fact, and proposed in 1889 a method which would supply a partial remedy. Mr. Taylor grasped the full fact and furnished a remedy which completely solved the problem. His best monument is the science which he created, and that will endure forever."

While the engineering profession is chiefly interested in Taylor's career after he reached maturity, one cannot help wanting to know something of the youth and early manhood of such a man.

What may there have been in the influences of his early surroundings which contributed to the development of such a striking character?

Some rather interesting side lights are thrown on this period of Taylor's life by Birge Harrison the artist, a friend from boyhood, who says:

"With our brothers and our boyhood friends we led a healthy, normal life in the woods and fields about Germantown, playing cricket and rounders, foot-ball and mumble-the-peg, and scouring the country side for thirty miles about in search of minerals and 'specimens' of various kinds—for under Fred's leadership we were all more or less 'scientific,' even founding the 'Germantown Scientific Society' whose oldest charter member, if I remember rightly, had not reached the age of fifteen.

Fred was always a bit of a crank in the opinion of our boyhood band, and we were inclined to rebel sometimes from the strict rules and exact formulas to which he insisted that all of our games must be subjected. To the future artist, for example, it did not seem absolutely necessary that the rectangle of our rounders court should be scientifically accurate, and that the whole of a fine sunny morning should be wasted in measuring it off by feet and inches.

It seemed to some of us also that Fred was a trifle over severe in his insistence upon the strictest possible observance of all the rules of the game—whatever it might be that we happened to be playing. But once this observance of the law was conceded and agreed to he was most generous to his opponent, allowing him every possible chance to win and conceding every doubtful point. And indeed it always seemed to me that the combination of these two qualities, which were so remarkably apparent in the boy—generosity, and a strict and uncompromising demand for and adherence to the truth and the law, had much to do with the success of the *man* in the development of his system of scientific management—in which for the first time justice and kindness and scientific accuracy were made the basis of the ever troubled and uneasy relations of the employer to the employed.

Even a game of croquet was a source of study and careful analysis with Fred, who worked out carefully the angles of the various strokes, the force of the impact and the advantages and disadvantages of the understroke, the overstroke, etc.

Both of Fred's parents were, as is well known, ante-bellum abolitionists—giving freely of their time, their money and their energy to the cause which was at that time of such overwhelming importance to the nation. Fred's love of justice and humanity, his uncompromising sense of the essential equality and brotherhood of man was therefore a direct inheritance from two unusual parents, and was bred in the bone.

During the later years of the war when the appearance of the Confederate troops at Gettysburg made the invasion and perhaps the fall of Philadelphia seem quite within the range of possibility, thousands of the older men who had not been drafted into the active service enlisted as 'Home Guards' and drilled actively morning and evening and on holidays in order to be of some use in case the threatened invasion should actually take place. At this time I remember that all the boys of our neighborhood and acquaintance banded themselves together and formed a company of juvenile home guards, to which was eligible in the capacity of a private soldier any able-bodied citizen in good standing who was over the age of six years. I cannot remember that Fred displayed any enthusiasm in regard to this patriotic movement, however. Indeed I strongly suspect that even at this early age his convictions as to the fellowship and essential equality of all men, white or black, rich or poor, were so strong that he was capable of holding that even the universally hated and despised 'Rebels' had certain rights which were theirs and inalienable simply because they were human beings like ourselves—a proposition which was of course inconceivable to any of the rest of our company. In later years I have heard him maintain that the so-called virtue of patriotism was in reality a selfish, narrow and ignoble quality, leading to international strife and injustice, and blinding nations to the rights of other nations—and that it would one day be universally so regarded.''

In 1869 at the age of thirteen he went to Europe and spent two years in school in France and Germany. Following this he had two years of travel, visiting with his family all of the principal cities of Europe. His diaries kept at this time are interesting, one entry being an account of his struggles with a certain difficult problem in geometry and of how he mastered it.

It was after his return from this sojourn in Europe that Taylor started for Exeter to prepare for entrance in Harvard University, where it was his intention to study law, but to his great disappointment he was obliged to abandon this plan owing to a serious eye trouble. This appeared to him as a veritable tragedy. It was this unforeseen event that led to his entering the engineering profession.

Taylor's apprenticeship to the machinists' and patternmakers' trade was served in the shop of the Enterprise Hydraulic Works (later Ferrel and Mucklé) where he started to work in 1875, leaving in 1878 to take a job as a workman in the Midvale Steel Works. Owing to business depression he was unable to secure work as a mechanic, and so took a job as a laborer; but soon after, the position of shop clerk became vacant and Taylor, by reason of his education, was given the job.

A little later, when Mr. Brinley, who at that time was manager at Midvale, had conceived the then novel idea of establishing a tool room, Taylor was put to work as a helper to the man in charge of this undertaking—whom he ultimately replaced—carrying on the development of the tool room and taking the first steps toward the standardization of cutting tools. Those who were associated with him at Midvale during this time express the opinion that Mr. Brinley's influence and guidance had much to do with starting Taylor on the road which led to his developing what is today known as Scientific Management.

Later on he was given a job as a machinist running one of the lathes, and as he turned out more work than was being done on other similar lathes, he was, after several months, made gang boss over the lathes. His efforts here to increase production and to conscientiously discharge the duties of his position met with bitter opposition from the workmen, but nothing daunted he kept steadily on and after a few years on the sudden death of the foreman was made, quite unexpectedly, foreman of the shop.

This constant warfare between the workmen and the management led to his decision to make an effort to bring about a change in the system of management that would harmonize the interests of the workmen and the management, and it was here that his first work leading to elementary time study and his experiments in the art of cutting metals commenced.

These two important parts of Taylor's work, as in fact all of the essential features of his system, were the outcome of a desire to set piece-rates based on facts rather than opinion, and thus to eliminate the injustice to the worker inherent in the old methods while at the same time insuring to the employer a proper return for the wages paid and from his investment, and enabling the payment of a higher wage with a lowered total cost. Each step in this work emphasized the need for standardized conditions, and the elimination of those variables beyond the control of the workman. One of the most valuable of by-products from this work was the experiments in the use of leather belting described in a paper presented in 1893, which has been referred to by Mr. Towne.

Following the period as foreman he was promoted in 1885 to the position of chief engineer.

A rather remarkable indication of Mr. Taylor's character is evidenced in his having felt the need for more thorough technical engineering education, taking up in 1883 a course of studies at Stevens Institute of Technology which he carried through successfully, receiving his M. E. degree and at the same time doing his work at Midvale. In commenting on this Wilfred Lewis, Past Vice-President, Am. Soc. M. E., who was in close touch with Taylor during this period, says: "At Midvale his hours were long and his duties strenuous, and this meant hard work day and night for three years, but he never faltered, and many times after burning the midnight oil over his studies, I have known him to dispel the nervous tension and put himself in trim for much needed sleep, during the few hours that remained before he was due again at Midvale, by running through the streets of Germantown for half an hour."

One of his associates at Midvale relates that often Taylor would get up at two o'clock in the morning and apply himself to his studies until four—after which he would take a bath and dress in his working clothes and lie down to rest until time to get his breakfast and go to work.

While at Midvale, in addition to laying the foundation for his system, he distinguished himself as a mechanical engineer and made many notable developments in machine tools, taking out a number of patents, among these being his lathes with hydraulic feed used in boring ingots and large gun tubes, and various time saving devices for chucking driving and handling work.

Mr. Henry L. Gantt, who started to work with Mr. Taylor in 1887, and who was closely associated with him in the development and application of the Taylor System throughout Taylor's career, speaking of his attainments as an engineer says:

"The least heard of, but, to my mind, his most daring feat, was the design of the great hammer of the Midvale Steel Company, which kept its alignment by the elasticity of its parts, which yielded to the force of a foul blow and returned exactly to their former position. Dependence upon the principle of elasticity enabled him to build a hammer which, for its weight, had far greater power than any other hammer that had ever been built. All previous hammers of this class had been designed to keep their alignment by great mass and stiffness, and it took a bold man to throw precedent aside when the stake was such a large one. I do not know of any more daring or successful piece of engineering construction.

The work by which he is best known, however, is not what was then regarded as strictly engineering. Strange as it may seem, although much knowledge and thought had been devoted to the design of machinery and apparatus, but little study had been given to the possibilities of the men who were to operate that machinery. To Mr. Taylor perfection in design was worthless without efficiency in operation, and at an early date he turned his attention to the efficient utilization of human effort.

In this work he used the same methods that had already brought him success, namely, to disregard opinions, from whatever source, unless substantiated by facts. Where facts were not available, and they seldom were, he used the scientific method for their determination.

When I went to the Midvale Steel Works in 1887, he had already made considerable progress in this work, and had fully developed the methods of detail analysis and study, which later became the origin of scientific management.

He recognized as an economic as well as an ethical fact that the employer should always consider the interests of the employe.

If I were asked to point out his most prominent characteristic I should say that it was his ability to prosecute the task he had set himself, regardless of the lack of sympathy of his friends and the criticism of his enemies.

Having determined on a course of action he pursued it regardless of consequences; and inasmuch as such courses were planned by a clear head and followed with an iron will, he often accomplished results far in excess of what even his most earnest supporters thought possible.

He was not the steam roller that some people like to represent him, but he did believe that a strenuous life was the life worth while, and that it not only brought more financial compensation, but that it added to the usefulness and happiness of men."

Concerning the study at Midvale of the art of cutting metals, Mr. Gantt said:

"Balked at the outset of his career as foreman of the machine shop of the

Midvale Steel Company by the lack of knowledge of cutting steel, which then existed, he set himself the task of supplying that lack. The first three years were spent in finding out how to study the problem; and, although the work was not completed for over twenty years, it is a fact that when I entered his employ in 1887 the fundamental laws had already been approximately determined.

At Bethlehem he became so interested in determining these laws exactly, that it is doubtful if he ever realized how wonderfully accurate his earlier results really were. Much more ground was covered in the subsequent work, but as an investigation into the laws of cutting metals, his work as a young man at the Midvale Steel Works stands out, to my mind, as far the more remarkable achievement. In his subsequent work he followed strictly the methods he had previously perfected.

One of the by-products of this investigation was the discovery of the Taylor-White process of treating high speed steel, the far reaching effect of which has not yet been realized.''

While at Midvale Mr. Taylor developed a very complete system for the maintenance of plant equipment, providing for periodical inspection of all machinery with complete detailed instruction as to how such inspection should be made and the tools to be used. These instructions covered not only the larger and seemingly more important things such as inspection and cleaning of boilers, but such things as oiling of machine tools. A ''tickler'' was maintained to bring forth the order for inspection and repair of each piece of equipment at the proper time. So far as the writer knows this was one of the earliest, if not the first, attempts to establish a system for the prevention of break-downs or what is sometimes called preventative repairs.

Another novel installation during Mr. Taylor's incumbency as chief engineer at Midvale was a system for supplying soda water as a cooling agent to the lathes and boring mills—providing a copious flow to each machine from a central overhead tank, the water draining into a setting tank from which it was pumped back to the overhead supply tank.

Admiral Caspar F. Goodrich, in speaking of his friendship and association with Mr. Taylor said: ''I first met Mr. Taylor in 1885, and took a great fancy to him. I recommended him to the then Secretary of the Navy, Wm. C. Whitney, for the supervision of our great Gun Works at Washington, D. C. I do not think Mr. Taylor felt himself free at that time to accept the position, but at all events this recommendation brought him into relation with Mr. Whitney.''

The outcome of this recommendation was that Mr. Whitney, who was interested in the results of Taylor's work at Midvale, got Taylor to come to Washington for a conference and offered him a considerable increase in salary to assume the role of general manager of the Manufacturing Investment Company. This proposition was accepted, and Taylor left Midvale in June, 1890.

The Manufacturing Investment Company was organized for the purpose of putting up and operating plants for the conversion of forest products into fibre suitable for the making of paper, having in mind principally the utilization of the slabs, edgings, sawdust and bark produced in the manufacture of lumber, which were then regarded as waste and burned. This company purchased the patent rights to a new process invented by a German chemist, Prof. Andrew Mitcherlich. Mr. Taylor's first task was to supervise the construction, which had already been started, and organization of two mills then building, one at Madison, Me., and one at Appleton, Wis. In this work Mr. Taylor had with him Wm. A. Fannon, Mem. Am. Soc. M. E., one of his aides at Midvale, who continued with this company—now the Interlake Pulp and Paper Company, of which he is vice-president and general manager. It was also during this period that Sanford E. Thompson, Mem. Am. Soc. M. E., became associated with Mr. Taylor.

The Madison Mill was under the charge of Admiral Goodrich, and the Appleton Mill under the late Admiral Robley D. Evans, both of whom were on leave from the Navy Department for this purpose.

Mr. Thompson states that in the construction of these mills Mr. Taylor introduced large quantities of special machinery which he designed himself, and that in one of the mills he applied piece-work to all of the complicated operations of manufacture by his method of elementary time study, thus doubling the output. He also introduced in these mills, according to Mr. Fannon, certain other features of what is now known as the "Taylor System".

Mr. Fannon states that Mr. Taylor had hardly taken hold and got the organization started before it became evident that the enterprise was to be very largely a disappointment, due to the patents not affording protection commensurate with the high price paid for them, to the fact that in their haste to get mills built and running, sufficient time and care was not taken in the selection of the locations, and finally to the panic of 1893. "So great was this disappointment to Mr. Taylor," says Mr. Fannon, "that I think it affected his health."

Even to those who knew him best the scope and thoroughness of Taylor's knowledge was a source of constant surprise. Comparatively few people knew of his work in accounting, and yet he was unquestionably an expert in this line.

As a result of difficulties encountered in his dealings with accountants, he determined, while with the Manufacturing Investment Company, to master this subject, and as a result developed a system of accounting that coördinated with his system of management, which as a result of a logical and just method of distributing overhead or indirect expenses gave accurate costs. Taylor's system of accounting, which has served as a basis for all modern industrial accounting, provided for an analytical and comparative monthly statement of expenses, a perpetual running inventory, and for monthly closing of the books giving a general ledger balance sheet and profit and loss statement. In this work Mr. Taylor had the assistance of an accountant named Wm. D. Basley to whom he gave much of the credit for the system developed.

In June 1893 he left the Manufacturing Investment Company and for a number of years he devoted his time to the introduction of bookkeeping and costkeeping systems and to the application of his methods of shop management in various plants in the East and Middle West.

At this time Taylor's revolutionary ideas of management were regarded with mistrust, and he could not for a long while find clients possessed of the requisite understanding, courage and confidence to let him undertake the complete application of his methods. Where today manufacturers in installing the "Taylor System" are content to go to great expense and trouble expecting no return for perhaps two or three years, at that time immediate results were constantly demanded. This resulted in greatly retarding the working out and installing of a complete coördinated system and the development of an organization upon which permanent and complete success depend.

In spite of the discouragement and opposition encountered, Taylor's faith never wavered. By compromising, by modifying to meet existing prejudices, he might have made more friends, made his own path smoother, and have met with greater temporary success. But he was not a trimmer. Truth was his guiding star, and the attainment of his ideals more important than his own welfare.

At Midvale, while all of the essential features, broadly speaking, that are today recognized as making up the "Taylor System" were practiced, the mechanism through which they were applied was crude and loosely hung together, and it was during the years from 1893 to 1903 with the constantly broadening knowledge resulting from work in such diversified fields that Mr. Taylor evolved the methods for planning and control of work so necessary to the proper and successful application of the principles of scientific management.

Among the clients for whom he worked during this rather discouraging period were: Simonds Rolling Machine Company, Fitchburg, Mass.; Cramp's Ship Yard, Philadelphia, Pa.; Northern Electric Company, Madison, Wis.; Johnson Company, Johnstown, Pa.; Wm. Sellers and Company, Philadelphia, Pa. Of these, perhaps the Simonds Rolling Machine Company was the most important, and in his paper on Shop Management, presented before the Society in 1903, interesting reference is made to his work for this company.

While undoubtedly even in his earlier work Taylor appreciated the importance of the worker's welfare in securing the greatest efficiency, it is most forcibly brought out by the following quoted from Shop Management.

> "The hours of work were shortened from 10½ per day, first to 9½, and later to 8½; a Saturday half-holiday being given them even with the shorter hours. Two recesses of ten minutes each were given them, in the middle of the morning and afternoon, during which they were expected to leave their seats, and were allowed to talk. The shorter hours and improved conditions made it possible for the girls to really work steadily, instead of pretending to do so."

In addition to his other activities during this period, Mr. Taylor, in collaboration with Sanford E. Thompson, made exhaustive researches in construction work and the building trades. The following statement by Mr. Thompson gives an interesting view of Mr. Taylor's personality and work at that time.

> "Coming from Midvale, we recognize the competent, hustling, able, inventive engineer. In his notable paper, A Piece Rate System read before the Mechanical Engineers in 1895, we find the first presentation of what he then termed 'elementary rate fixing,' that is, the determination of the proper time for doing a piece of work by unit time study. But we find in this paper

scarcely a reference to the broader subject of management or scientific standardization.

In his paper, Notes on Belting, however, presented two years earlier, in 1893, the principles of standardization and of scientific research are clearly brought out in the development of definite laws and of a definite system for handling the complex problem of belting—the adoption of the scientific method, the method which eliminates from a test all variables but one, the method which develops a problem step by step until the attainment of definite laws.

The principle of unit times, which is now recognized as forming the basis for the accurate analysis of labor operations, was completely developed while at Midvale. During that same period also were made the belting tests and the beginning of various other researches aiming toward standardization of methods.

Not, however, until the publication of Shop Management in 1903, is the development of the complete system, based not on theory, not on opinion, but as a result of this broad experience in operation gained by his contact with manufacturing plants all over the country.

In other words, he discovered as a result of his work—a fact probably not yet fully appreciated even by some managers—that, in order to carry on these fundamental principles of elementary rate fixing, of unit times, there must be embraced a comprehensive plan of organization, a plan which includes the establishment of functional management, with its planning, its routing, its inspecting, and its training of employes; and above all, with its scientific analysis of labor and machine operations for the purpose of standardization of materials and methods.

As I said, at the beginning of this period we have the able engineer, at the close of this period we find the scientist, the man who has worked out, years in advance of his time, the application of science to the cutting of metals and the application of science to industrial management.

In 1894 while he was engaged in this introduction of management methods Mr. Taylor proposed that I take up with him an analysis of work in the building trades with a view to publishing unit costs of various kinds of construction work. This has resulted in the publication of the books Concrete, Plain and Reinforced and Concrete Costs, and material for other works on earthwork, carpentry, etc. is nearly ready for publication.

In this writing of books we find the same fidelity to standards. He makes up his mind as a result of examination of facts that a thing should be done in a certain way, and in that way it must be done. While Mr. Taylor did comparatively little in the direct preparation of these books, their success is due to Taylor principles. Adopt standards—present simple, clear-cut conclusions—give conclusions at the beginning of every discussion. It is interesting to learn that these principles are being accepted at the present time in engineering reports and technical writing as a result of a precedent thus established.

At the time of beginning this work I made my first visit to Mr. Taylor's early home—a quiet mansion located on Ross Street, Germantown. I had the great privilege of meeting his father and mother, an accomplished gentleman and a gentlewoman of the type rarely met with in the younger generations—in a home where the refinement of the family life was marked. It had been the desire of these parents to give the son in his young days a broad education. He spent three years, from his thirteenth to his sixteenth year in Europe, traveling and studying music, art and language. It is suggested that his acquaintance with the beauties of the Alpine passes developed a love of nature which found expression in his design and layout of the Boxly Estate.

It was during this association with him that I came to understand his real character. Before this I was a little in doubt what was the real Taylor—whether he was essentially the task master that he sometimes appeared, that he seemed to be when he would require the attainment of the apparently insurmountable—when he hauled us over the coals as man never did before. But I soon learned to distinguish the man himself from certain qualities that were not really traits but were simply acquired by him in his usual thorough and scientific manner because he saw that at certain times and under specific conditions a special plan of action, a special policy, a special manner of speech was necessary in order to train his subordinates or in order to accomplish his purpose. Always underneath was the generosity, courtesy, tenderness, loyalty to friends and subordinates, readiness to appreciate and commend absolute fairness. He went into everything he undertook a little farther—often immeasurably farther—than anyone else had gone before. As one of his Midvale associates said to me 'Taylor is all right except that he is a generation ahead of his times.' That remark was made more than twenty years ago and the industrial world is gradually growing up to the level then already attained by him.

Throughout my association with Mr. Taylor that which stands out most clearly is the definite accomplishment of purpose, not by brute force, not by the temporary and physical means of sheer weight or numbers, not of the type of ability which built the pyramids, but of the type which produced the accurate mechanism of the watch—the adherence to the scientific method, the appreciation of the establishment of standards.''

In 1898 Mr. Taylor went to the Bethlehem Steel Company.

The results of his labors here are commented on at considerable length in his paper Shop Management. Here he undertook the application of his system of management in a thorough manner and on a large scale. When one considers the short period he was at Bethlehem the work accomplished seems almost miraculous.

Alone, the development of high speed steel, the discovery of the Taylor-White process for its heat treatment, the extension of the experiments in the art of cutting metals, the studies in the handling of pig iron and other materials and setting for these of scientifically determined tasks would have been a sufficiently great accomplishment to be remarkable and bring fame to Taylor.

At Bethlehem was associated with him Henry L. Gantt, who during this period developed what is well known as the Gantt Bonus System most widely used in connection with the setting of tasks based on elementary time study replacing in a large measure but involving the essential principles of Mr. Taylor's differential piece-work system.

In the work of standardization of equipment, revising and completing the work of determining the laws of cutting metals, and the development of slide rules for working out the practical problems encountered in putting the results of this work into use Carl G. Barth, Mem. Am. Soc. M. E., took a prominent part.

Mr. Barth's comments on his association with Mr. Taylor are as follows:

"While I made Mr. Taylor's personal acquaintance at least as far back as the year 1884, while working for Wm. Sellers and Company of this city as a draftsman, and as such occasionally had to do with the working up of some of his many ideas for the improvement of machine tools, it was not until the summer of the year 1899 that I became associated with him at the works of the Bethlehem Steel Company, an association which continued uninterruptedly until his death, and thus during nearly sixteen years.

What I can say will be of a fragmentary nature only, and principally intended to give a little insight into Mr. Taylor's great character, as I learned to view it in my associations with him.

When Mr. Taylor began the original work that finally culminated in a complete system of scientific management for industrial establishments, he had no idea of what he was steering towards. The difficulties that first beset him in his career as a leader of men, led him to believe that most of them would disappear if he could find some scientific way of predetermining the time it should take to do a given piece of work in a machine tool, such as a lathe or planer. However, it took him and his several associates a period of some eighteen years before this problem was even theoretically solved to his satisfaction.

When I joined Mr. Taylor at Bethlehem, it was for the express purpose of assisting him in the solution of this, his original pet problem; and I shall never forget the intense delight evinced by Mr. Taylor one morning when I was able to hand him an empirical mathematical formula representing the results obtained by a set of experiments made in metal cutting with high speed tools of his and Mr. White's renowned make, which was at once recognized as the beginning of a better way of attacking the problem than anything previously brought to light.

The fact that the work was not his own, did not in the least detract from his satisfaction. The great soul that he was, it did not matter to him whence the solution came,—his efforts for so many years seemed finally likely of being crowned with success, and that was all he cared for.

It was a few months later that the final solution of this problem enabled Mr. Taylor's task system, in conjunction with Mr. Gantt's bonus, as a substitute for Mr. Taylor's differential piece-rates, to be instituted at Bethlehem. Inside of a comparatively short time this led to that most astonishing increase in production, which at that time was the wonder of all visitors to the works, and which was partly due to the high speed tools, and partly to the scientific methods employed in their use on machines that had been rebuilt and respeeded to meet the new conditions, in connection with the reward to the workmen who properly coöperated in the whole matter.

My first visit with Mr. Taylor at Bethlehem for the purpose of discussing a possible engagement, happened to be on a day that certain yard laborers, who were to be put on piece-rates, threatened to strike. Words to that effect reached Mr. Taylor during our interview, but while it is impossible to believe that it did not inwardly affect him, he did not betray the slightest perturbance and completed his interview with me as if nothing had happened. I will add that I learned later that the strike did not take place.

Perhaps the greatest lesson taught some of us by Mr. Taylor, is the value of confidence in general principles, and general experiences. I have thus never forgotten the absolute confidence with which he some twelve years ago assured a certain prominent manufacturer that the recent favorable reports the latter had received about greatly improved conditions in a plant in another city in which he was interested, could not represent the facts; as it would, he said, in the very nature of the conditions that were known to have existed there a few months earlier, take almost as many years to bring about the alleged improvements. This subsequently proved to be the case.

His faith in scientific methods and the immutableness of natural laws and general principles, he only shared, of course, with numberless scientists of his day, but as a practical engineer and manager he had had experiences that do not come within the range of the professional scientist.

Another of Mr. Taylor's most striking characteristics was his great appreciation of those of his superiors of former times that had taught him valuable lessons. For some of those he did not entertain a high general regard, but with a fine discrimination he would laud the good he had seen in them, and draw his lesson from it. And as regards seeing the good in other people, the development of his character as I had the rare occasion to notice it, resembled what I once heard a lecturer say about Abraham Lincoln. 'His heart grew more and more tender as the years went by, until just before his death, he was ever ready to see excuses for the behavior of even those of his disciples who were not as loyal as they might be to the great ideals for which he had worked so faithfully and disinterestedly.'

Great was his work viewed from only the material side, greater, by far, were the ideals that prompted it, and which he left to sustain us, as they did him, through the numerous difficulties, large and small, which the practical continuation of his work carries with it.''

Mr. Maunsel White, an able metallurgist, collaborated with Mr. Taylor in the development of high speed steel and the process for its treatment.

The value to the world and to the engineering profession of the accomplishments of these two years cannot be calculated.

In his great work On the Art of Cutting Metals, presented as his presidential address before this Society in 1906, Mr. Taylor gave to the world the results of twenty years of labor in this field, revolutionizing machine shop practice and the construction of machine tools.

The ownership of the Bethlehem Steel Company changed hands in 1901, and as a consequence the work of Mr. Taylor and his associates there was brought to an end when it was still far from completion. With the termination of his work at Bethlehem he withdrew from salaried positions of all kinds.

At this time he said to one of his intimate friends "I cannot afford any longer to work for money." That statement was typical of the man. Money was not an end with him but only a means, and having found the means to satisfy his reasonable desires his whole heart went into the furtherance of the work with which he had been identified throughout all his career.

During these last years, after having withdrawn from gainful occupation, he continued to work tirelessly in the interest of scientific management. He was constantly called upon for lectures and addresses by institutions of learning and various organizations of economists, manufacturers, and engineers. His correspondence with people interested in his activities equaled that of many business houses and extended to all parts of the world.

One of his greatest desires was to train men to carry on the work of developing and extending his system, and he gave much of his time and money to this end. He was ever accessible to those who sought his advice or help, and many times did he sustain the courage and faith of his disciples through the most trying and discouraging periods encountered in their work. His personality was ever a source of inspiration, and an unusually strong spirit of loyalty—partly to the man but above all to his ideals—inevitably grew in all his associates.

Taylor's influence was felt and he was looked up to as a master in all civilized lands. His writings were translated into French, Danish, German, Dutch, Russian, Italian, Lettish, Chinese and Japanese. Men came from all of the foreign countries to which his fame had penetrated as well as from all parts of the United States to study his system of management in plants where out of respect for Taylor the owners granted this courtesy.

At least once a week at his home Mr. Taylor set aside a day to receive those interested in his system, and after explaining at length its basic principles, which he illustrated from his own experience, he conducted his visitors through one of the plants in Philadelphia to see the system in operation and to explain the mechanism through which the principles are applied.

Mr. Wilfred Lewis, a close friend, in a paper dealing with the versatile genius of Taylor, says:

"He was a good sport and very fond of tennis, at which he soon became an adept by utilizing his Sundays and holidays without encroaching upon any of his regular duties. His play, in fact, was so nicely fitted in with his work that one helped the other; and very early in the history of the Newport tennis tournaments, when the double game was played almost exclusively, the team of Taylor and Clark will be found among the winners. At the same time he was quite a social favorite and did not wholly renounce the demands and pleasures of society while working under heavy pressure.

He had a keen wit, sometimes tinged with biting sarcasm and was very quick at repartee. Every phase of life in which he moved seemed to have its attractions and he always took a leading part in entertainments of various kinds. For a time, as I remember, he was especially devoted to private theatricals, and on one occasion he was the whole show at the Academy of Music in a very amusing monologue which brought down the house, and, by the way of encore, he gave another selection equally effective. I never understood how he found time to prepare for these diversions while so busily engaged in more serious matters, and can find an explanation only in the wonderful alertness of mind and body with which he was endowed.

His shop experience, coming as it did in the midst of his technical education, gave him a keen appreciation of industrial problems from the workman's standpoint and made him a profound student of human nature. This experience also made him value more highly the education acquired by his unusual energy and ambition and led him, no doubt, in later years to advocate a year of shop experience for all young men before completing a college course.

He was impressed very early in his career with the inequity of the piecework system, which encouraged an effort to increase wages by greater production and then rewarded that effort by a cut in the rate of pay. His sense of justice rebelled at this and he lived to establish a better way, known as the Taylor System of Scientific Management, which is really a constructive philosophy of industrial life. His sense of justice was one of his strongest characteristics and with this he coupled loyalty to friends as one of the cardinal virtues to be forever kept in mind.

He was a reformer of the first magnitude who took life as he found it, and standing firmly on his feet as a product of evolution, in which he firmly believed, he sought to improve the social fabric of which he formed a part. He was not a blatant demagog bent upon destruction for the sake of an imaginary reconstruction of society. He saw clearly the good and bad elements in its composition and held fast to the good while helping on the process of evolution to something better.

As an engineer, he took nothing for granted and insisted upon the final test of practical experience as the only verdict worth having. In the early part of his career he was very resourceful, bold and ingenious and full of enthusiasm for new and useful devices in the bud, as well as in the bloom. The 20-ton Midvale hammer has been mentioned as a monument to his genius as an engineer, and his boring lathes with hydraulic feed, at the same plant might also be mentioned as successful innovations of hardly less importance.

In later years, when he realized the cost of improvements and their effect in checking the wheels of progress, he began to question their value, and he was loth to give up any well established practice for something that appeared on paper to be better. His disposition was always toward simplicity and directness of means to an end, and he had a horror for what he called 'damned improvements' which resulted in never building two machines alike.

When in Germany. shortly before the war, one of my customers was pleased to show me his plant operating on the lines laid down by Taylor, and he was proud to acknowledge the source of his inspiration and the good results obtained. Other concerns were doing the same thing and it cannot be doubted that the Taylor System of Scientific Management will ultimately become an integral part of the German 'Kultur,' which stands for efficiency and the survival of the fittest. Taylor, himself, had no fear of the final results, although he knew the opposition it was bound to meet at home and abroad through ignorance and political intrigue.

While engaged in reorganizing the Simonds Rolling Machine Company he incidentally developed his ingenuity as a detective and succeeded in trapping a number of concerns suspected of infringing the Simonds patents. Although I have forgotten the details of these exploits, the impression made by their recital some fifteen years ago was one of admiration and amusement not excelled by any of the Sherlock Holmes stories I have since read, and I have no doubt that the license found among Taylor's effects as a stationary steam engineer in Massachusetts from 1897 to 1900 bears directly upon his detective work. As I remember, this was one of his schemes to obtain access to the plant of a suspected competitor where Taylor applied for a job to run the engine.

After leaving Bethlehem, he was offered fabulous sums for the reorganization of other large concerns, but he realized that he was nearing the limit to his powers of endurance and wisely considered his health of more importance to himself and to the cause for which he stood than any monetary compensation.

Believing that he could no longer afford to work for money, for the last fifteen years of his life he devoted himself to the welfare of humanity by spreading his new gospel of management throughout the world at the sacrifice of his fortune, personal comfort and health; and it was while so engaged that he contracted the cold which brought his long and useful career to an untimely end.

He was honored by institutions of learning and was an honor to everything with which he was connected. No one ever had a better friend than Taylor nor a more implacable enemy. Few men, I think, have been so cordially hated and beloved. When he fought he meant to win and never hesitated to hit hard, but he always fought for justice and never for peace or good-will based upon weak-kneed concessions of any kind. So, while his enemies fade away, his friends and followers increase in constantly widening circles.

He was a man of might because he was a man of industry and genius devoted to the truth, and the versatility of his genius has endeared his memory in many ways to a host of friends in whom the fire that he kindled will burn and glow to the glory of his name in future generations.

Mr. Taylor was married May 3, 1884 to Miss Louise M. Spooner, the daughter of Dr. and Mrs. Edward A. Spooner of Philadelphia, and they lived in Germantown until the termination of his connection with the Midvale Steel Company. It is interesting to recall the wedding ceremony as one of the last, if not the very last, performed by the venerable Dr. Wm. H. Furness in the old Unitarian Church at 10th and Locust Streets, and it is certainly a distinction to have been married by one so universally admired and beloved.

After leaving Midvale, Mr. Taylor was obliged to travel more or less between the paper pulp mills of which he had charge. This entailed a good deal of domestic inconvenience which Mrs. Taylor always preferred to share. I remember visiting their home in Madison, Me., on the banks of the Kennebeck River, which furnished power for the mill. The scenery was beautiful and the climate exhilarating, but Madison was a small town, far from the attractions of city life, and it seemed more like a romantic wilderness than a home for Mrs. Taylor. But here I found Taylor rejoicing in a new frame house built to Mrs. Taylor's very attractive original plans, clearly indicating their disposition to enjoy life wherever duty called them.

From Madison, they soon moved to Appleton, Wis., then back to New England towns, and at last to Bethlehem, Pa., where some of Taylor's greatest achievements were realized. Later they settled in Germantown where he built his beautiful home ''Boxley'' at Chestnut Hill. The hospitality of this mansion will be remembered by thousands who have been entertained in its spacious halls, and it will be a gratification for all of them to know that Mrs. Taylor continues to make it her home and is devoting herself untiringly to the promotion of her husband's ambition and the perpetuation of his memory. Here Mr. Taylor's office is kept open and the correspondence with people interested in the scientific management movement is carried on by a group of Taylor's associates under the name of the Frederick W. Taylor Coöperators.

During the last two or three years of his life, Taylor exhibited a spirit of devotion which few who had known him, chiefly in a professional way, suspected him of possessing. Mrs. Taylor's health failing, he placed his duty toward her above his work in the field in which he had labored for years, at a time when more than ever before enduring success was at stake, and all of his engagements were made conditional upon his wife's health permitting him to keep them.

In private, as in his profession, Taylor's life stands out as a model and a source of inspiration.

The sod that covers his grave like a blanket of rich velvet marks the last resting place of no common mortal. It bears its silent testimony to the genius of him whose form it covers, and among those who gather from time to time at this, the brightest green spot on the banks of the Schuylkill, will always be found the rank and file of labor, the bone and sinew of industrial progress, in loving remembrance of a great benefactor, a pioneer and leader of men."

While from the time he left Bethlehem, Taylor had retired, at least in theory, from active professional work, he was at all times in close touch with the work of his followers, and was ever ready with helpful counsel. Not until after his death did they realize how much they had leaned upon him.

During the latter years of his life in some measure he was compensated for the hard fight he had made against criticism and opposition born of prejudice and ignorance. The engineering profession honored him by his election as President of this Society in 1906. In this same year the University of Pennsylvania conferred upon Mr. Taylor the honorary degree of Sc. D., and Hobart College that of LL. D. It is typical of Taylor's democracy of spirit that after receiving these degrees, while being deeply appreciative of the honor, he rather resented the distinction of being addressed or referred to as Doctor Taylor. To him his work was of far greater importance than the matter of personal glory.

The more progressive element in the government service became interested in the Taylor System about 1905, and efforts were made with some success to apply Taylor's teachings in the arsenals and navy yards. This work was undertaken in earnest by General Crozier, Chief of the Bureau of Ordnance of the Army, who upon the recommendation of Mr. Taylor secured the assistance of Carl G. Barth and the system was completely installed in the Watertown Arsenal and extended to other arsenals.

In 1906, Admiral Caspar F. Goodrich brought Mr. Taylor and Truman H. Newberry, then Secretary of the Navy, together and a plan was evolved for carrying out in a more comprehensive way the work begun independently in the Navy Yards by certain officers in charge. Speaking of this, Admiral Goodrich, at the memorial services of this Society, said:

"I am indebted to him (Taylor) for a great many very valuable services. In 1907 I was sent as Commandant to the New York Navy Yard, where I found five separate industrial establishments within the yard, each one, in so far as it could be, complete in itself. Each one of these plants had been erected by one of the bureaus of the Navy Department,—there were buildings devoted to construction and repairs, ordnance, equipment, yards and docks, etc. This multiplication of plants was not conducive to economy, and I immediately saw the need for reform.

Mr. Truman H. Newberry, who was then Assistant Secretary of the Navy, and who later became Secretary, had charge of the navy yards at that time, and I went to him with the facts and I said: 'Mr. Secretary, I will present to you no scheme, no proposition, that I have not already threshed out with the ablest engineer in this country. I do not pretend myself to be a mechanical engineer, but I do, fortunately, possess the friendship and confidence of Mr. Frederick W. Taylor, and I will go over all these propositions with him and I will bring nothing before you that does not meet with his approval.' Mr. Newbury, of course, knew all about Mr. Taylor and he liked to feel that any matter in the way of reform would be based upon the broadest and the wisest of grounds. He promised me the backing of the Navy Department. I consulted Mr. Taylor and we drew up a scheme and carried it out, and suffice it to say that with Mr. Newberry's authority and Mr. Taylor's advice we succeeded in concentrating the work of the Yard into fewer shops and in enormously decreasing the appropriation necessary.

I will only give one other illustration of what he did. There came over to my desk on one occasion a statement or schedule, as it is called, of the tool steel required at the different navy yards in only one of the departments. I was amazed at the vast number of varieties of tool steel that were demanded by name, by brand, and in every case there was a certificate that none other would serve. I consulted Mr. Taylor who advised me to buy steel according to specifications and he very quickly offered me all the assistance I required. So we prepared the specifications, which were later adopted by the Tool Steel Board appointed by Mr. Newberry to take up the question of the varieties of tool steel required, and their composition.

The first high speed steel that was bought under Mr. Taylor's specifications, as adopted and promulgated by the Navy Department, was for the gun works at Washington, and Admiral Leutze, who was then in command, told me he had been previously paying \$1.25 a pound for the steel required and he was now getting tool steel for from 32 cents to 36 cents a pound, and the new steel was doing about 33⅓ per cent more work.

I thank you for this opportunity of claiming what I consider to be one of the greatest privileges of a long life, that of speaking of my former intimate and dear friend, Frederick Winslow Taylor.''

The plan referred to by Admiral Goodrich contemplated not only the standardization and purchase of tool steel, but also standardization of all tools and their manufacture and treatment in one central plant from which they would be supplied to all navy yards.

With the change in administration came a new Secretary of the Navy, George Von L. Meyer, who undertook the extension of the "Taylor System", appointing a board consisting of naval officers and civilians who under Mr. Taylor's guidance worked out a very comprehensive plan, which was never carried out.

Unfortunately, economy and efficiency in government work are not essential from a political point of view, and yielding under the pressure of opposition from within and without the navy, the Secretary abandoned the plan. Nevertheless Taylor's influence at this time so strongly affected many of the officers in charge of navy yard work that while the Taylor System could not be installed as such, its principles were applied in a considerable degree.

In 1911, Louis D. Brandeis, now one of the Judges of the Supreme Court, as counsel for the Eastern shippers opposing before the Interstate Commerce Commission the railroads' demand for an increase in freight rates, brought Taylor's work to the attention of the entire country. Previous to this time the Taylor System was known chiefly in engineering circles and among managers of iron and steel working industries. Mr. Brandeis aroused the country by making the statement that the railroads, by the application of scientific management, might effect economies which would make the granting of the increase in rates unnecessary. Accounts of these hearings, published broadcast in the daily press, naturally created a tremendous interest in Taylor's work and marked its general extension to industries outside of the strictly engineering lines.

In 1912, a society was formed by a small group of manufacturers and engineers interested in the Taylor System which was known as the Society to Promote the Science of Management (since changed to the Taylor Society). After Mr. Taylor's death this society held a Memorial Meeting October 1915, at which Mr. Brandeis was among the speakers. All his life Brandeis had been a champion of the working people, and it was with their welfare in mind that he considered Taylor's work. Perhaps in this respect no one better understood the motives which inspired Taylor. In view of this fact it seems desirable to repeat here Mr. Brandeis' address.

"It was Taylor's purpose to make the laborer worthy of his hire; to make the hire worthy of the laborer; to make the standard of living and the conditions of working worthy to be called American. The American standard of living implies a wage adequate for proper housing and food and clothing, for proper education and recreation and for insurance against those contingencies of sickness, accident, unemployment, premature death or superannuation, which fall so heavily upon the working classes. That standard implies hours of labor sufficiently short to permit those who work to perform also their duties as citizens and to share in the enjoyment of life. That standard implies postponement of the working period to an age which enables the child to develop into a rounded man or woman. That standard implies working conditions which are not only consistent with the demands of health and safety, but are also such as may make work for others what it was for Taylor—the greatest of life's joys.

Taylor recognized that in order to make such a standard of living and of working attainable, the productivity of man must be greatly increased; that waste must be eliminated, and particularly the waste of effort which bears so heavily upon the worker. And yet the man who sought to so develop industry as to enable labor to reach these higher standards of working and of living, met, throughout his life, wide-spread opposition from those whom he sought particularly to help. Let all who are undertaking to carry forward his work recognize this hostility as a fact of fundamental importance; for it presents the main problem which confronts scientific management.

The causes of this hostility are twofold:

First: Only a part of the necessary industrial truths have been as yet developed.

Second: The necessary assent to the application of these truths has not been obtained.

Taylor was a great scientist. He established certain truths, fundamental in their nature. But he obviously covered only a part of the field of inquiry. The truths he discovered must be further developed and they must be supplemented by and adjusted to other truths. The greater productivity of labor must be not only attainable, but attainable under conditions consistent with the conservation of health, the enjoyment of work, and the development of the individual. The facts in this regard have not been adequately established. In the task of ascertaining whether proposed conditions of work do conform to these requirements, the laborer himself should take part. He is indeed a necessary witness. Likewise in the task of determining whether in the distribution of the gain in productivity, justice is being done to the worker, the participation of representatives of labor is indispensable for the inquiry which involves essentially the exercise of judgment.

Furthermore, those who undertake to apply the truths which Taylor disclosed must remember, that in a democracy it is not sufficient to have discovered an industrial truth, or even the whole truth. Such truth can rule only when accompanied by the consent of men.

We who have had occasion to consider the hostility of labor leaders to the introduction of scientific management know that the hostility has in large measure been due to misunderstanding. Much of all the waste which Taylor undertook to eliminate, has no direct relation to the specific functions of the workingman. It deals with waste in machinery, in supplies, in planning, in adjustment of production and distribution—matters in which changes cannot possibly affect the workman injuriously. And yet we found in many leaders of labor undiscriminating opposition to the whole of the so-called Taylor System. But even if we succeed through education in eliminating the general hostility to the introduction of scientific management in departments of the business which do not directly affect labor, there will remain a wide field where the proposed changes do directly affect labor in which there is determined opposition. This opposition can be overcome only through securing the affirmative coöperation of the labor organizations. In a democratic community men who are to be affected by a proposed change of conditions should be consulted, and the innovators must carry the burden of convincing others at each stage in the process of change that what is being done is right. Labor must have throughout an opportunity of testing whether that which is being recorded as a truth, is really a truth, and whether it is the whole truth. Labor must not only be convinced of the industrial truths—which scientific management is disclosing—but must also be convinced that those truths are consistent with what may be termed human truths. Is the greater productivity attained clearly consistent with the health of the body, the mind and the soul of the worker? Is it consistent with industrial freedom? Is it consistent with greater joy in work, and generally in living? These are questions which must be answered in the affirmative, and to the satisfaction, not of a few, merely, but of the majority of those to be affected.

To do honor to Mr. Taylor and to worthily carry forward his work, those who are his disciples, and those who may become such, should recognize that they have in the solution of these questions a call upon them for patient effort no less exacting and severe than that to which Taylor subjected himself when pursuing the law of cutting steel. Every step in the installation and the working out of scientific management calls for such coöperation by representatives of labor. The obstacles to securing it are great. Twenty-five years may be required to remove them fully. But whatever the time required to fully convince organized labor, it must be given, if our work is to be well done. The consent and the coöperation of the worker so represented must be secured. In no other way can we attain in full measure the increase of productivity upon which our well-being so largely depends. In no other way can we secure that joy in work without which increase of productivity will not bring greater happiness. In no other way can we attain that freedom and development of the worker without which even his greater happiness would not promote the general welfare. Let us work unremittingly in the spirit of Taylor to solve the problem he left unsolved. In the solution of that problem—which in a true sense is the labor problem—the greatest honor will be done to his memory and the greatest service to mankind."

Some idea of the esteem in which Taylor and his work were held abroad is given in the addresses at the Memorial Meeting of the Society to Promote the Science of Management, already referred to.

The President of the French Republic, through Ambassador Jusserand, sent a representative to the meeting in the person of Colonel Vignal of the Engineer Corps, Military Attaché to the French Embassy at Washington. In a letter M. Jusserand said: "It is very difficult for me to leave my desk in these days; I would else have craved, myself, the honor of taking part in the celebration and rendering homage to one whose tuition has made him one of the benefactors of mankind."

Colonel Vignal presented a tribute from the great French scientist, Henri le Chatelier, between whom and Mr. Taylor had grown a strong friendship and mutual appreciation. M. le Chatelier's paper was entitled "How have I known Frederick W. Taylor; why have I endeavored to popularize Scientific Management?" The following is a partial quotation:

"Frederick W. Taylor is a mechanician and I am a chemist: he is an engineer, and I am a professor. What has brought us in touch with each other? How have I been led to undertake the popularizing of industrial methods, which is quite outside of my province? Some will say it is chance,—the veriest accident. But, in the Taylor System there is no room for chance; all facts are necessarily related to each other. The very object of this system is to disentangle the inevitable relations of phenomena. Chance has to do only with those relations of which we are still ignorant. The questions which I raise here give very clear proof of the correctness of this definition of chance. If the bringing together, across the Atlantic Ocean, of two scholars entirely unknown to each other seems at first sight inexplicable, the following statement will demonstrate on the contrary that it was inevitable and that chance had nothing to do with it.

I have devoted my life to the study of science, and in pursuit of this study, I have allowed myself to be guided by a few leading principles borrowed from the philosophical works of Taine. To my mind the end of science is simply the study of the relations existing between phenomena, that is to say, the study of natural laws. Moreover, a sound method for the study of these laws consists in at once directing all one's efforts toward the analysis of the most important factors, that is, of those which play a preponderant part in the determination of a given result.

Being moreover a professor in a polytechnical school, I naturally had to interest myself from the very first in the elements of industrial progress; in my opinion science is the dominating factor therein. In order to develop the influence of science in French industry, and to make our engineers understand the beneficial role of scientific methods of work, I established La Revue de Metallurgie about fifteen years ago. In this publication I proposed to give a leading place to the studies of industrial science, while giving ample space to purely technical information, which was necessary to insure the reading of my review by those manufacturers who are often but partially convinced of the practical value of science.

Faithful to these principles, in editing this review I was obliged systematically to give a conspicuous place to the dominating facts; to allot the number of pages devoted to each industrial process, according to its real importance. At the time of the Paris Exposition, in 1900, struck by the evident importance of high speed tool steel, I reviewed systematically all the articles bearing on this discovery, in order to give extracts from them in La Revue de Metallurgie. I published, among other things, an extract from a lecture of a Sheffield engineer, Mr. Gledhill, attributing the discovery of high speed tool steel to a lucky chance. A careless workman had overheated one of his tools and, far from damaging it, he had considerably improved it. This incident coming to the knowledge of two industrial engineers, Messrs. Taylor and White, had given birth to high speed tool steel.

Not believing in chance, I had followed up this article with some personal remarks, saying that it had certainly required a high order of scientific observation and investigation on the part of the engineers in question to have been able to draw such an important discovery from the carelessness of a workman. This article fell under the eye of Frederick Taylor. Some months afterwards when he decided to publish the history of his discovery in his celebrated presidential address to the Mechanical Engineers, On the Art of Cutting Metals, he sent me a copy of the final proofs, thanking me for my words of appreciation. It might interest me, he said, to know that chance, as I had foreseen, had had absolutely nothing to do in the discovery of high speed tool steel.

I then asked Mr. Taylor to authorize my publishing a French translation of his paper, which he very obligingly granted. But, he added in his letter, he believed that he had done something much more important than his work on cutting metals, namely, his scientific management of shop work. He asked me to read attentively his paper called Shop Management and to give him my criticism of it. I knew the work in question very well by name, but I thought that it treated simply of a system of paying wages, the differential system, more or less similar to Halsey's system of premiums, which had not seemed to me sufficiently interesting to make me buy the book and read it. Once in possession of this volume, I studied it conscientiously and I was profoundly surprised to find in it a very remarkable application of the scientific method to industrial problems. In undertaking the publication of La Revue de Metallurgie, I had proposed to generalize the applications of science to industry, but I had not understood the full extent of the domain of science. I had hardly dreamed beyond the introduction of the laboratory and of its experimental methods in factories, but I had not foreseen the possibility of extending the domain of science over all the realm of industry, including questions of organization, commercial questions, labor questions, etc.

I was somewhat ashamed to find the science of a practical man infinitely more developed than my own. From that day on, I felt myself obliged, in order to remain faithful to the program which I had from the first mapped out for myself, to constitute myself an apostle of the Taylor System. From the beginning I was perfectly aware of the difficulties, and of the time which the spread of the new ideas would require. It had already been hard enough to induce manufacturers to make use of laboratories, even when the material results were tangible and paid immediately. It would be still more difficult to make them accept a more complex method of work, more costly to put in operation and, above all, giving only more remote results. Calling to mind then, this other principle of Taine, that to convince people it is not sufficient to give them good reasons, but that above all you must fire their imaginations by a series of individual facts which they can easily digest and which all lead to the same end, I made up my mind, either in La Revue de Metallurgie, or in other publications, to come back incessantly to the advantages of Taylor's Scientific Management. A nail is finally driven home by the constant repetition of little blows. It was in this way that an active correspondence with Frederick Taylor was brought about, and the beginning of those sentiments of friendship arose which made his premature death particularly painful to me. We shall endeavor at least to make his ideas live, and to awaken the feeling of gratitude to which he is entitled, because of the beneficent work he left behind him.

After all, the bond which inevitably drew us together was the community of our scientific interests, directed alike toward industrial progress. We have, independently and without any acquaintance, come upon each other from different routes which led to the same end: We had to meet sooner or later. There was indeed no accident in the origin of our collaboration.

The French translation of the Principles of Scientific Management, has been printed in two editions, to the number of 8000 copies, of which 3000 have been gratuitously distributed, and 4000 sold—representing then, at least 5000 readers. Today Frederick Taylor's ideas are familiar to the majority of French engineers: Whether they will or no, they necessarily exercise an influence on every one of their decisions.

The advice of consulting engineers, which is considered so natural in the United States does not obtain with us. To reorganize a factory you appeal to one of Taylor's disciples—Barth, Gantt, Thompson, etc., or to one of their imitators. With us, on the contrary, a factory insists on reorganizing itself only with the help of its own staff, and on invoking no name except that of the firm. I had warned Frederick Taylor that in France his system would take the name of the engineers or the firms which would put his ideas into practice. 'I desire nothing more,' said he, 'so that my ideas spread, it matters little the dress under which they circulate.'

In France, the most earnest opponents of the Taylor System are perhaps the economists. That may seem surprising, but on second thought one understands why scholars discussing industrial questions which are altogether out of their province without ever having set foot in a machine shop, must in the nature of things conform their ideas in their criticisms to previous opinions and to systems established by long tradition. They do not dare to launch out into new fields, whose foundations they are not able personally to appreciate. Be that as it may, Frederick Taylor's ideas are making their way little by little. Machinery was forced upon industry in spite of the attacks of which it was the object; it will be the same with the scientific principles of management of work. From certain points of view, their success would be even easier, because ideas have a far greater force of penetration and of diffusion than material objects. One can break up machinery, burn down shops, but there is no way of coercing ideas."

Another noted French engineer and manufacturer, Charles de Freminville, said in part:

"The associates of Frederick Winslow Taylor, gathered together to perpetuate his memory have, in asking me to present a few remarks on the work of their master, conferred on me a great honor. I wish to point out, from the point of view of the engineer, how Frederick W. Taylor's ideas were presented in France; with what sympathetic interest they were received, and how great a service they are called upon to render to the French.

When Frederick Taylor's works, On the Art of Cutting Metals and Shop Management appeared, they attracted the particular attention of a learned engineer whose name is universally recognized, Henri le Chatelier, by whom these important works were first published in the Revue de Metallurgie. The method used involved such determination and continuity of effort, such close coöperation for an unprecedented length of time, and laws carried out to such a fine point, that it was difficult to think it was not exaggerated.

When the works of Frederick Taylor were published in France, the name of the great engineer must have already been known there, for it was that of one of the inventors of the high speed tool steel which had made such a great sensation at the Paris Exposition in 1900.

On the Art of Cutting Metals had been published with Shop Management by M. le Chatelier. The volume passed from hand to hand, and after having commanded the attention of manufacturers, of the directors of railroad companies, etc., it reached the managers of the shops and the foremen, who were struck by the practical advice, based on a profound knowledge of the world of labor, which they met in every line.

From that moment Frederick Taylor acquired in France the right of citizenship, and the assimilation of Mr. Taylor's ideas and of his method was only a question of time.

'Method' has long been honored in France. It characterizes the spirit which the great technical schools endeavor to inculcate in their pupils and which has contributed not a little to their ability to occupy an extremely important place in industry.

The French workman himself was not the last to understand how Taylor obtained such astounding results in the working of metals. Long since accustomed to associate the names of scholars with great industrial discoveries, he willingly accords to them the admiration and respect due to extraordinary men, and frequently experiences a lively desire to contribute, however little it may be, to their work. 'Never mind,' he has said more than once to himself, after having applied himself very methodically to his task, 'I have done a little like Taylor.'

It is not surprising that the seekers after precedents have explored France with the greatest care, in the hope of finding the germ of Mr. Taylor's ideas there: But they have been obliged to admit that they found themselves in the presence of a new work. If they happened to discover that some of the finest geniuses of French mechanics, such as Belidor, Vanban, Condomb, had paused for a few moments to analyze the motions of the workman and had left a few notes on the subject, there was still no connection between these notes and the labors of Frederick Taylor. Not only were these labors not minimized by these great men, but they received brilliant tributes from them.

Frederick Taylor liked France and would have liked to be of use to her. Once during two brief hours when we were together, we happened upon a maritime laboratory, on the coast of Brittany, born of the most modest beginnings and sheltering for several years, the joint labors of fishermen and scientists. Discoveries of the greatest importance had been made here with the most rudimentary material, but they had brought very little honor to their authors. The most elementary of their valuable findings were hardly taken advantage of. For instance, when they investigated matters of public interest, such as the public health, they brought upon themselves endless trouble.

They consoled themselves with reading the accounts of the application made in distant countries, of their discoveries. and on such a colossal scale as to be almost incredible. Frederick Taylor must have made more than one reflection when visiting this laboratory. At any rate, he had the pleasure of being able to say to his hosts that there was no exaggeration in what they had read, and that their discoveries had been made the object of the most important application.

Frederick Taylor found there a graphic example of a fact which had been pointed out to him again and again. What he saw excited his enthusiasm to help France to desire more practical advantage from the discoveries she was able to make. He was full of hope on this subject, for he could see how completely his ideas had found an echo in France, and how serious were the efforts already made to put them into practice, and so during the last sojourn he made in our midst, in one of those addresses in which one would seek in vain for a word of flattery he did not hesitate to salute in France the country which offered the finest future for the application of his methods.

The movement then launched, which seemed so full of promise, has been arrested by the war, but already France is thinking of the future, and of the necessity, greater than ever before, of undertaking a systematic organization of her resources and of her work.

Frederick Taylor was an observer of exceptional penetration but his work is witness to the fact that he was indeed one of those men, rare in any country, who from the beginning of their career subordinated all their actions to a high and perfectly definite purpose. No one has ever done so with more energy or determination. In full command of the masterly trait of scientific observation, which he possessed in such high degree, he continually directed it towards a definite end. Never did he allow his imagination to bewilder his observations, or to alter their exactness. They all bear the trace of absolute sincerity.

Again we detect the same integrity when he is concerned with the application of his ideas to the organization of labor. He is not afraid of provoking contradiction and he did not welcome that approbation which is so dangerous and which drowns the idea by returning it to that void from which it had been delivered.

Taylor possessed a lofty mind, embracing a widely extended field of activity. If he gave himself particularly to the task of making people understand the efficacy of his method for the better utilization of material resources and of the every day activities; if it pleased him to show with rare ability, that this method could be applied as well to agriculture as to mechanics or to sports, he knew how to raise himself above his work itself in order to affirm that the use of the means which he recommended must cease with the material world. And it is no exaggeration to say, that in struggling against the waste of energy and time which constantly accompanies not only industrial labor but also those of every day life, he strives to make a larger place for the intellectual life.

Frederick Taylor regarded spiritual things with great respect. He was deeply touched when he learned that one of the most authoritative voices of the French pulpit in a daring comparison, had not been afraid to define 'the love of God' as 'the Taylor System of our inner life.'

The man of genius is not frightened by the greatness of the task which he undertakes; and troubles himself little about the profit which he ought to get out of it, for it is a small task indeed whose materialization does not exceed the life of an individual.

Inevitably Frederick Taylor could not have put in the complete development of the movement which he had begun, but he was able to see that the roots were already very deep, and that a brilliant future opened up before him."

Prof. J. J. Sederholm, of Helsingfors, Finland, who prepared a paper for this meeting, said:

"The most memorable event during my three months traveling in North America was undoubtedly my meeting with Mr. Taylor. It was so, both because I personally admired him so much, and because I think that his teachings are exactly what we want in my country. Finland.

I can add nothing to the characteristics of his personality, and should I speak of the kindness with which he offered me his help, I should have only to describe what all of his friends have experienced. Let me therefore restrict myself to considering what benefit my own country may be expected to reap from his influence.

As is well known, the struggle for existence between the nations of Europe is very keen both politically and economically. The small nations in particular are forced to strain all their efforts in order not to be overwhelmed or left behind.

It is therefore simply a necessity for us to learn from every nation, appropriating the best which they have to offer. Personally I think that we have especially much to learn from the United States. Of this I may mention an instance, rather trivial in itself, but significant. When our sportsmen formed the ambitious plan that Finland should beat at the last Olympian games, as many of the other countries as possible, they took their training methods from America. The result was, that little Finland became the fourth country of the world in this international sporting contest; next to the United States, Sweden and England, beating Germany, France and other large countries.

Now I think that in all kinds of human industry we also ought to learn from the Americans how to 'go ahead,' hoping a similar success, if we do so.

The Taylor system is not only to Europe 'an American lesson,' it is *the* American lesson. It is true that I have read in a foreign newspaper an article about it, full of misrepresentations, in which it was styled 'false Americanism.' To me, on the contrary, it is the *very essence of good Americanism.*

The Taylor System enables us to introduce American briskness also on every field of human industry in Europe. As is well known, the Taylor System is not at all a method of 'speeding up.' It is a system of working intelligently, with spared effort, and it offers to everybody taking part in the work advantages unheard of before. Every workshop where the Taylor System is used is a place where men work in harmony, in conjoint effort, to mutual benefit, and more intelligently than ever before.

This we regard to be the great discovery of Mr. Taylor, that he has found the necessity of using much more brains than before in managing industries and, in general, all human work. His scheme seems to be as simple and obvious as the famous egg of Columbus, but it is, however, on most fields an innovation.

Last winter when I was alternately lecturing on the Taylor System and the Evolution of the Animal World, I was struck by a curious analogy. In the history of the earth, it was only at a late date that nature discovered the usefulness of large brains. The monster reptiles of the Mesozoic era had only minute brains in their gigantic bodies. Still at the eve of the Tertiary era mammals with the size of our cattle had brains as small as a walnut, and it was not until the end of that era that the brains attained their present size. This development reached its climax in Man, *Homo Sapiens*, the animal with brains who, on account of his intellect, became the master of the earth.

Industry has still to learn the same lesson. It has not yet advanced beyond the Mesozoic stage, but the time will soon come, when people will regard shops without a planning department of sufficient size, shops where hundreds of laborers are managed by half a dozen of engineers and foremen, with the same wonder as is felt by us when we look at the skeleton of a *Diplodocus Carnegie* with its gigantic body and almost microscopical brain. And when that time arrives, then everybody will also recognize the greatness of Frederick Winslow Taylor, the discoverer of the simple truth that large brains are necessary in industries and, in general, for managing all kinds of human labor."

That Germany was not slow to appreciate the importance of Taylor's work is made evident by the address of Prof. A. O. Wallichs, Mem. Am. Soc. M. E., of The Royal Polytechnic School, Aix-la-Chapelle.

"The character and significance of the Taylor doctrine have been accepted by the German people only in part, both because the time for making Taylor's meaning clear has been too short, and because we here are just at the beginning of its practical application. The essence of the doctrine, namely, 'To better the conditions of the laboring classes and to increase the general pleasure in work' must certainly find in the German people fertile ground for development. This is especially true, because in Germany the demand for a means to settle social inequalities is becoming steadily stronger and the earnest purpose to get together in the pursuit of a common goal is beginning to ripen among ever larger sections of both labor and capital.

Taylor did not grow tired of pleading with both sides: 'Your interests are, for the most part, not hostile, but identical.' And he did not content himself with words alone: he had *proven* the truth of his doctrine in practice before he proclaimed it. In that lies, as I see it, the immense value of his work.

The world has had enough of the profound opinions of learned economists and philosophers as to human happiness. Taylor, however, did not give his propositions publicity until after he had by hard work and incessant struggle, thoroughly tested the possibility of carrying them out. In his classic, Shop Management, he declares repeatedly that 'nothing is so convincing as bringing to pass actual results.'

The remarkable thoroughness in the execution of his work, his consideration of all the circumstances,—whether or not they had hitherto seemed insignificant —his perseverance in the pursuit of his aims, must and will find ultimate recognition on the part of the German people. Economists, German engineers and scholars, who understand the true essence of the Taylor doctrines have, almost without exception, become his disciples. Naturally the Taylor principles are most widespread in the ranks of engineers and industrial managers. Among these one finds unreserved appreciation, while the critics are found more among those who have only a superficial knowledge of the system. This fact is the best recognition of the correctness and practical value of Taylor's principles.

A small group of German scholars and manufacturers, immediately after the publication of Taylor's basic works, Shop Management and On the Art of Cutting Metals, recognized their far reaching significance. So it seemed to the writer a worthy task to make Taylor's books useful to a wider circle of German engineers by translating them into German.

After the favorable reception, which the German edition of the book, On the Art of Cutting Metals, had met in Germany, it was clear that Taylor's basic work entitled Shop Management could not long be withheld from German engineering circles. The writer, therefore, undertook at once this task, with all the greater zeal because after careful study he found it a treasure house of great truths in the difficult art of management and *especially in the treatment of the workmen.* Taylor recognized that the often asserted antagonism of the interests of employer and employe need not exist with proper management and treatment; that rather there exists a mutual interest in the success of the enterprise, and that this same interest can well be united with a higher wage and more humane treatment of the worker. With remarkable insight he devised ways and means to save unnecessary loss of time, yet without the necessity of overworking the operatives.

Taylor shed light in every nook and corner of the daily routine; and examined everything in the effort for well-planned use of time. *His ability to grasp things fully and with keen perseverance to draw from the knowledge attained its practical application, together with his wonderful knowledge of men and his true love for humanity, enabled him to win a success which has aroused the astonishment and admiration of the world.*

The reasons which hinder a rapid and extended application of Taylor's doctrine lie not in the limitations of the field but in the *lack of trained forces* to guide its introduction, and partly also in the weakness of human nature among the managers. The self-esteem of many of these gentlemen is hurt by the thought that processes discovered and developed by others should be better than their own kind of management worked out through decades of struggle and strife, in many cases with successful results. They oppose it, therefore, chiefly because they fail to recognize the superiority of new methods. Many assert that they have long since recognized and adopted to a large extent the principles developed by Taylor. They also consider that Taylor's control of the smallest elements, going beyond anything they undertake in their own organization, is superfluous hair-splitting. Where these conceptions have taken root, they undoubtedly work against the spread of scientific management. Nor can it be disputed that at least a part of these assertions are correct. We know of numerous organizations in most branches of our industries, which have achieved phenomenal success, because of their well regulated and fairly well executed planning of work; but *no one* had proven or even suspected that thorough scientific observation of all, even the minutest and seemingly most insignificant processes, would result in such significant reductions in the time and effort of work. No one else has shown Taylor's perseverance in *carrying through* a well-defined logical program. Still less have our managers perceived the importance of the coöperation of the workers for obtaining the greatest economy. The social aspects of his success are not sufficiently recognized. Taylor himself places them above the purely technical features. Just a short time ago he expressed himself clearly on this subject to a German visitor. His words follow: 'These plans for reducing the cost of production will be improved and surpassed by others, primarily through machine technique and also through better ideas of organization; of all these things which today we claim as the best obtainable, not one will remain. But one thing will and must remain, and that is the basic idea which guided me, from the very beginning, in all my work; namely, the fundamental recognition which alone carries us forward: THE EARNEST AND HONEST EFFORT FOR IMPROVED RELATIONS BETWEEN EMPLOYER AND EMPLOYE, THE STRIVING TO ABOLISH THE ANTAGONISM BETWEEN THESE TWO FACTIONS—TO THIS WE MUST STEADFASTLY HOLD.' Such words prove that Taylor sees, in social progress, his greatest success.

We have a sufficient number of men, who, free from an exaggerated idea of their own success, wish to make the most of the economic value of this new doctrine. But more difficult to meet is the above mentioned lack of organizers trained in Taylor's ideas. For the improvement of this condition, both public and private bodies should coöperate in the interests of industry and, through special courses of instruction and trips to the United States for study, take steps to build up a suitable force of teachers.

The systematic training of organizers is quite essential for the following reasons. The development of scientific management must be undertaken in the factories without disturbance and along with the regular routine work, by a special organizer, not by the manager of the plant. The manager has neither the time nor the thorough knowledge to carry out all the details of study in accordance with the prescribed regulations.

All the reports received from German industrial circles of initial experience during introduction are, almost without exception, favorable. As an example I append a report of a firm engaged in the wood industry in the Rhine District: 'The first test with Taylor's principles was made on a 'Fasson Lathe' which turned out large quantities of wooden pieces for cabinet makers. A worker on this machine worked at piece-work at the rate of three pfennings per meter, earning at his maximum capacity between 4 and 4.20 marks daily, accomplishing about 130 to 150 meters' work during the same working time. After time studies were taken with a stop-watch, it proved that the actual working time was only one-fourth of the total time and that the rest of the time was lost through the sharpening of tools, setting up, repairing of belts, bringing of the material, etc. We introduced, first of all, tools of *high speed steel* instead of the ordinary kind hitherto used; we replaced the existing bronze bearings with ball bearings, and the ordinary belts with best quality leather belts. The result was remarkable. Although the worker was transferred from piece-work to day work based on his average daily earnings, he easily produced 300 meters daily. After the introduction of a premium system, based on his daily wage, in a short time he ran up to 400 meters daily. Through further time studies it was then established that the forward and backward run of the machine which heretofore had been done by the workman, could be done automatically, and that during the backward run the next piece of material could be brought up by the workman. The result of these further improvements is the present daily production of at least 550 meters and average earning of about 5 marks daily for young workers from 17 to 18 years of age, while formerly adults, working to the limit of their endurance, could earn only 4.20 marks. As we did not simply pocket as profit the advantage in production so obtained, but reduced correspondingly the selling price, there resulted immediately an increased volume of sales, so that not only was no reduction in working force necessary, but on the contrary, the installation of a second and then a third Fasson lathe. To the above mentioned favorable result was added the circumstance that according to Taylor's doctrine, we established a functional foreman whose only duty was to provide the raw material and set up the machines. The above result shows how this unproductive labor paid for itself.'

This report is in many respects very instructive. It shows that in a plant excellently managed according to ordinary standards, studies made in a small auxiliary department under the inspiration of Taylor's works, and changes made on the basis of these studies, have resulted in surprisingly large economies for the business, and in essentially increased earnings for the workman; this, too, without taking any steps to change the entire organization through especially trained organizers. Naturally in this plant further studies and adjustments will follow until, in the course of years, it can be said that in all respects the standardization of the working processes has been completed. The report shows that where the determination for improvement exists, former shortcomings are quickly found.

Another report lies before me, from the repair shop of a chemical plant in the South of Germany. Here, to be sure, the work was done by one of the few German engineers who had been trained in one of the Taylor plants in the United States. The success here was in an altogether different field; while in the former instance the improvements were limited principally to an improvement in method for a work chiefly mechanical, in the chemical plant the success was achieved *exclusively by a strictly supervised preparation and division of the tasks to be done by the repair workers*, employed mainly on hand work. There was given to each workman each day, a task defined in writing and with only one possible interpretation as time, nature and extent. At the same time order in keeping of stores was established according to the Taylor methods. Even in the first year after the introduction, the yearly losses in stores was greatly decreased, and the amount of work accomplished was greatly increased.

In one department of a large Berlin machine shop, the men no longer object as they have heretofore, to the exact measurement by the stop-watch of the working time of the best workmen for the purpose of determining a just piece-rate. They have recognized this method as just and, since the introduction of this strictly controlled management, they have drawn essentially higher wages, averaging 90 pfennings per hour. Of course the stop-watch can be used only openly for the measuring of the working time, any underhanded methods being rightly subject to deep mistrust, and you must not call it the Taylor System.

It will be seen, from these reports, that Taylor's ideas have found fertile ground here in Germany. The great European war has been an obstacle in the development of this as in so many other relations. Nevertheless, I am fully convinced that after peace is proclaimed, which, let us hope, will be soon, Taylor's stimulation will be felt again in Germany with redoubled force—to bless a favorable economic development and to better the lot of the workman."

It was peculiarly fitting that the Memorial Meeting should have been held in a hall of the University of Pennsylvania which a few years before had recognized his genius by bestowing upon him an honorary degree. Provost Edgar F. Smith as host of the evening expressed the satisfaction the University felt in having recognized Mr. Taylor and honored him with a doctor's degree. He quoted at length from Mr. Taylor's address at the time, in which Mr. Taylor showed his grasp of one of the most acute problems in the life of the undergraduate student, i.e., the laxness of discipline and lack of training for thoroughgoing, responsible work.

Following the Provost's address, Mayor Rudolph Blankenburg paid the following brief but eloquent tribute to Mr. Taylor.

"The greatest tribute I can perhaps pay to the memory of Mr. Taylor is to advise you that soon after my election as Mayor of Philadelphia, four years ago, I requested him to call upon me. He did so and at my house we discussed all phases of city government and what would best serve the City of Philadelphia during the new administration.

After fully discussing this important question, I asked him to make a great sacrifice for the public by accepting the position of Director of the Department of Public Works. He seemed pleased but hesitated, stating that he did not see how he could do so. When I saw him again, a day or two later, he said, 'It would be a real pleasure for me to accept your offer so as to help you in the great work of regenerating Philadelphia, but it is impossible for me to do so on account of my health. I have really more to do now than should be asked of any man and it is a physical impossibility for me to add to my work.'

But Mr. Taylor helped me after all. When I looked further for a man to fill the important position of Director of Public Works, Mr. Taylor helped me in the selection and recommended to me one of his disciples. I appointed that disciple as director and he has made good and is an honor to the City of Philadelphia.

Mr. Taylor was to me a paradox. On one hand we find his rugged intellect blasting its way up through layer after layer of conventions formed by generations of prejudice, tradition and ignorance until he became recognized as perhaps the world's foremost industrial leader. When truth was the stake, he was resourceful, robust and tireless. The problem once even dimly visioned he pursued with the zest of a hunter until he conquered.

On the other hand, those whose contacts with him were, like my own, only casual and who went to him as converts, rather than to be converted, could hardly sense his power. He was borne and bred to a gentle manner. His sweet smile and courtly bearing were only the surface indications of an innate and broad spreading sympathy and kindliness. He knew he had much to give and he gave it with a generosity which knew no limits. Yet few men of this or any other time had sensed so clearly how much there is to be known and what a short way we have gone on the journey.

We in Philadelphia who saw Mr. Taylor come and go among us as our friend and neighbor only dimly comprehend—if at all—that the world has been listening to his teachings for years as to one of the master minds of his time. The Japanese, the French and those of Scandinavian lands were among the peoples who have read his books in their own tongues for years. The industrial scientists of Germany, Italy and Russia have crossed the sea with the beautiful home of Boxly as the end of their pilgrimage.

Today his fellow-townsmen are alive to the significance of his mission, and an eagerness to acquaint ourselves with his methods and principles is springing up in all our hearts.

This war-torn world of ours has indeed lost a great leader at a time when it needs him most. It would seem that when the moment comes to bind up humanity's wounds, the creed which Mr. Taylor lived and died to establish may prove one rock on which we may build a more lasting peace.

The City of Philadelphia is indeed proud of his genius and even more proud of the great service he rendered to mankind.

While we may some day erect monuments in marble or bronze to his memory, Frederick Winslow Taylor has erected for himself, in the city of his birth, an imperishable memorial in the great work which he has woven into the fabric of our institutions."

The late James M. Dodge, Past-President of both this Society and the Society to Promote the Science of Management, a staunch supporter and friend of Mr. Taylor, made the closing address with an inspiring characterization of Taylor, the man, ending with the words "we shall not see his like again."

Perhaps no other man contributed so largely and so disinterestedly to the promotion of Mr. Taylor's work as did Mr. Dodge, and it seems proper that acknowledgement should be made here.

No more fitting closing for this brief sketch of the life and work of Frederick Winslow Taylor may be found than in the remarks of James M. Dodge at Mr. Taylor's funeral.

"Frederick Winslow Taylor was a prophet, with honor, in his own country and, at the same time, honored and respected in every civilized country of the globe. He was a remarkable student, a devoted husband, faithful friend, an inventor of the first rank, an engineer of resource, knowledge and keen perception, indefatigable in his work, unswerving in his devotion to truth, modest and considerate, and with this remarkable combination of temperament and learning became the bearer of a message that is destined to make him recognized the world over as the emancipator of the worker and the employer, delivering the worker from the oppressive burdens of the old order and granting him freedom to do his best for himself, his family, and his employer from the necessity of being only the task-master and granting him freedom and opportunity to be the friend and co-worker of those associated with him.

Through his scientific investigations of the relations between employer and employe he was able to formulate a system which made it possible for both parties to realize that their interests instead of being in irreconcilable conflict were identical and that they were interdependent, and that all questions between them could be settled by kindness, forbearance and patient investigation without resort to mistrust, suspicion, or antagonism. He was the bearer of the only flag of truce that was ever carried upon the battlefield of industrial strife. Ignorance and prejudice have fired upon this flag, but it was never lowered, and now that the hand that carried it must relinquish its noble office, thousands of others will sustain it in its exalted position, and I predict that it will never be lowered and that the employer and the employe will both prosper under it as they have never prospered before, and with increasing respect, regard and solicitude for each other's welfare.

Many others have prayed for an industrial social millennium, expecting it to come from spiritual grace through lapse of time, but Dr. Taylor not only saw the possibilities of the future but he did more, he told in detail exactly how this long-hoped for condition might be actually accomplished at once. The seed he has sown is springing up in thousands of places; the message he gave us is making hundreds, yes, even thousands of converts; the work he so ably started being based upon eternal truth will partake of the lasting characteristics of its foundation."

A letter to the public press written, on the day of Taylor's funeral, by his friend Henry R. Towne, closes with the words. "One of the world's discoverers and creative leaders has closed his career. The world is greatly enriched by what his genius accomplished. The world is grateful that he lived and for what he did."

No. 1528

NECROLOGY

HAROLD BENTLEY ANDERSON

Harold Bentley Anderson was born in 1878. He received his technical training at the Case School of Applied Science. While in college he constructed a motorcycle and also a sewing machine driven by a small gasoline engine, and by the time he left the School he had built an automobile. In 1901, he secured patents on two automobile inventions.

He entered the employ of the American Bicycle Company in Toledo in 1901, where he brought out a system of double acting brakes and also the steering gear used on the copy of the "Lifu" truck, made in Toledo.

In 1902, he became associated with The Winton Company as personal engineer for Mr. Winton and in 1904, he was made chief engineer of this company, which position he held at the time of his death.

Mr. Anderson became a member of the Society in 1903, and was also a member of the Society of Automobile Engineers, American Institute of Metals, American Society for Testing Materials, International Association for Testing Materials, Cleveland Engineering Society, Beta Theta Pi, Theta Nu Epsilon, Cleveland Yacht Club, Cleveland Automobile Club and The Aeronautical Society. He was also an officer of the aero squadron of the First Aviation Corps, New York City. He died on July 13, 1915.

JACOB ROBINSON ANDREWS

Jacob Robinson Andrews was born on September 6, 1861, at Bridgewater, Mass., and was educated at Bridgewater High School and Bridgewater Academy. In 1879, he obtained employment as apprentice in the machine shop of the Hyde Foundry at Bath, Me. He was rapidly advanced to the position of foreman, and a few years later was made vice-president and general manager of the Hyde Windlass Company. When the latter firm separated from the United States Shipbuilding Company in 1905, he became its president.

Mr. Andrews worked untiringly to advance the interests of American shipping, and was one of the best known figures in shipping circles. He was a member of the Society of Naval Architects and Marine Engineers, the Engineers' Club of New York and a member of the Society since 1906. He died in New York City on March 25, 1915.

WALTER SEAVER BALL

Walter Seaver Ball, a member of the Society since 1900, was born March 17, 1867, at Upton, Mass. He was educated in the Upton schools and graduated from Worcester Polytechnic Institute with the degree of S. B. in 1889. He was associated for a short time with the Deane Steam Pump Company at Holyoke, and in 1894 he became connected with the McKay Metallic Fastening Association as assistant to the superintendent. He then became assistant superintendent with the United Shoe Machinery Company, and later moved with this company from Winchester to Beverly. For twenty-three years Mr. Ball held a position of great responsibility with the allied companies making this great industry. He died on September 11, 1915, at the Beverly Hospital, Beverly, Mass.

JOHN BIRKINBINE

John Birkinbine, whose death occurred in Cynwyd, Pa., on May 14, 1915, was born in Reading, Pa., on November 16, 1844. His education was received at public schools and the Friends High School in Philadelphia, the Hill School at Pottstown, Pa., and the Polytechnic College of Pennsylvania. His studies were interrupted by military service in 1863-1864 on scout duty with the Union Army under two enlistments. Later, he devoted two years to practical work in a machine shop and subsequently he was associated with the late P. L. Weimer under the firm name of Weimer and Birkinbine, which operated the Weimer Machine Works at Lebanon, Pa.

Much of his work was in mining, metallurgy and blast furnace construction. As manager for the South Mountain Mining & Iron Company, he carried on experiments with various fuels for iron ore smelting while maintaining the furnace in constant operation. The carefully recorded results obtained were widely published and are referred to in metallurgical text books as the most complete tests made.

From his Philadelphia office, he was sent to nearly every state and to Canada and Mexico for examinations, reports, constructions of or improvements to iron ore mines, blast furnaces, iron works, water supplies, hydraulic developments, irrigation projects, etc., and his engineering knowledge was requisitioned by several European corporations.

Mr.. Birkinbine was probably the first to suggest an iron industry at the head of the Great Lakes, using coke made from Pennsylvania coal, and his report was an important factor in establishing the iron industry at the head of Lake Superior. The blast furnace at West Duluth, Minn., too, was built under his supervision. He was engaged by the State of Texas to investigate the practicability of iron manufacture in that State. As an engineer, he cooperated with E. S. Cook of Pottstown, Pa., who did much to advance the iron industry. He was for some years consulting engineer for the Philadelphia and Reading Coal and Iron Company, and held a similar position with Thomas A. Edison during the latter's early experiments on magnetic concentration of iron ore, and with Witherbee, Sherman and Company in beneficiation tests; also for the Colorado Fuel and Iron Company for the enlargement and improvement of their works and the construction of an augmented water supply system.

He also acted as an expert for financial interests and for a number of the greatest corporations and several large railroads in this country. He was chief engineer, vice-president and chairman of the Committee of Awards of the National Export Exposition and served on Juries of Awards at the Centennial, World's Columbian, Pan American and Cotton States General and other expositions.

He was chairman of the Water Supply Commission of Pennsylvania from the time of its inception in 1905 and was active in forming the Pennsylvania Forestry Association. He was also active in the formation of and served as secretary to the United States Association of Charcoal Iron Workers, and for nine years edited its journal. For many years, he was special agent for the United States Geological Survey, preparing the reports on iron ores for the 11th and 12th Censuses and that on manganese ores for the 12th Census. He was appointed by the Secretary of the Interior as expert metallurgical engineer for the Bureau of Mines.

During his career, Mr. Birkinbine maintained his specialty in hydraulic engineering, acting as engineer on water supplies for various muncipalities. In 1888, he prepared a comprehensive report on the development of the great water power of the St. Louis River in Minnesota.

For ten years, he served as president of The Franklin Institute in Philadelphia. He was also a member of the Engineers' Club of New York, the American Society for Testing Materials, the Engineers' Club of Philadelphia, of which he was president in 1893, the Manufacturers Club of Philadelphia, the Pennsylvania Foundrymen's Association, the George C. Meade Post 1, G. A. R. of Philadelphia, an honorary member of the Canadian Mining Institute and a member of the Institute of Mining Engineers, of which he served as manager, vice-president and president. He was elected to membership in the Society in 1888.

AUSTIN LORD BOWMAN

Austin Lord Bowman was born in Manchester, N. H., in 1861. He studied engineering at Yale University, and was graduated with the degree of B. A. in 1883. For four years he specialized in construction and bridge work for western railroads, and in 1887, he came East and took up similar work with the Norfolk & Western Railroad. From 1890-1895 he was engineer and superintendent of construction for the American Bridge & Iron Company, Roanoke, Va. In 1897, he established himself in New York City as a consulting engineer on heavy railroad work. For six years, beginning 1901, he was consulting bridge engineer for the Central Railroad of New Jersey, reconstructing most of the important bridges on that road. In December 1907 he became consulting engineer of the Department of Bridges of New York City, and was made chief engineer of the Department in 1914, a position which he retained until his death.

Mr. Bowman was a member of the American Society for Testing Materials, American Railway Engineering and Maintenance of Way Association, and the New York Railroad Club. He was a member of the American Society of Civil Engineers and a director from 1905-1907. He was elected to membership in the Society in 1899. He died on June 3, 1915.

LUCIEN MAXWELL BRIGHAM

Lucien Maxwell Brigham was born in Brooklyn, N. Y., on June 6, 1874, and received his education in the schools of that city. His entire business career was spent in the service of the firm of Maxwell, Manning and Moore, with which he first became connected in 1894. He acted in the later years of his life in the capacity of sales manager of the brass goods department, and during the last five years he was a member of the board of business associates in the carrying on of his department.

Mr. Brigham was a member of the Engineers' Club, the Railroad Club, and several social clubs. He became a member of this Society in 1906. He died at his home in Orange, N. J., on December 11, 1915.

HOLSTEIN DE HAVEN BRIGHT

Holstein De Haven Bright was born in Philadelphia, Pa., on June 30, 1880. He was educated at the William Penn Charter School in Philadelphia and then entered The Baldwin Locomotive Works as an apprentice. He rose rapidly and in a few years was placed in charge of the upkeep of the works. He was then transferred to be assistant secretary of the Standard Steel Works Company, a subsidiary of The Baldwin Locomotive Works. He also organized the sales department of the subsidiary company.

In 1912, he resigned to accept the presidency of the Southwark Foundry and Machine Co., Philadelphia, which position he held until 1914, when he retired because of ill health. He died in Philadelphia, Pa., on November 2, 1915.

Mr. Bright was a member of the Union League and Meridian Clubs in Philadelphia and the Merion Cricket Club. He became a member of this Society in 1905.

WILLIAM ALEXANDER CHERRY, JR.

William Alexander Cherry, Jr., elected to membership in the Society in 1912, was born in Denver, Colorado, in October, 1888, and was graduated from Columbia University in 1911. During the summer of 1909 he served a machine shop apprenticeship with the Acme Machine Company of New York and a power plant apprenticeship with the New Rochelle Light and Power Company of New Rochelle during the summer of 1910. After his graduation from

Columbia he became associated with Viele, Blackwell and Buck, consulting engineers of New York, and worked on several designs of hydraulic power stations and gas producers. He was also employed in the laboratory in testing out various materials to be used in the manufacture of producer gas.

At the time of his death, Mr. Cherry was connected with the Florida Abstract and Title Insurance Company of Jacksonville, Florida in whose employ he had been for two years. He died in Atlanta, Ga., on March 6, 1915.

JOHN HENRY CLARK

John Henry Clark was born in Cornwall, England, on April 7, 1859. He served an apprenticeship as a machinist in the shops of Cooke, Rymes and Company at Charlestown, Mass., and also with the Whittier Machine Company in Boston, Mass., afterwards becoming superintendent of the works of the latter company. He was also associated with Hon. Oliver Ames, formerly governor of Massachusetts, in the development of an oil engine.

In 1890, Mr. Clark took a position with the Thomson-Houston Motor Company, an affiliation with the Thomson-Houston Electric Company for exploiting the electric elevator business. This company afterwards became merged in the General Electric Company, organized in 1892. When its power and mining department was formed, Mr. Clark became connected with it, and continued in that department until his death. He was transferred from Boston to Schenectady in 1895.

Mr. Clark became a member of the Society in 1906 and was also a member of the Boston Engineers' Club and the Engineers' Club of New York. He died at Schenectady, N. Y., January 3, 1915.

RAYMOND EARL CRANSTON

Raymond Earl Cranston was born in Providence, R. I., on November 25, 1883. He was educated in the Providence schools, spent one year in Brown University, and graduated with honor in 1906 from Massachusetts Institute of Technology, with the degree of B. S. He then entered the employ of the Manufacturers' Mutual Fire Insurance Company of Boston, and was later sent to the Providence office and became associated with John R. Freeman in his private engineering work. In 1912 he went to California with

Mr. Freeman as assistant on the Hetch-Hetchy water supply for San Francisco. Soon after he became assistant engineer for the company.

Mr. Cranston was a member of the Providence Society of Mechanical Engineers and was elected to membership in this Society in 1907. He died at his home in Providence, R. I., on June 25, 1915.

GEORGE HARWOOD CUSHING

George Harwood Cushing, superintendent of the North and South plants of the H. B. Smith Company of Westfield, Mass., was born in Worcester, Mass., on October 13, 1860. He graduated from the Worcester Polytechnic Institute in 1884. For a number of years he was assistant superintendent of the H. B. Smith plants, was superintendent of a pump plant at Seneca Falls, N. Y., for thirteen years, and was in charge of a foundry at Montreal, Canada, for two years. In 1906 he returned to the H. B. Smith Company at Westfield.

Mr. Cushing became a member of the Society in 1891. He died at his home in Westfield, Mass., on December 30, 1915.

WILLIAM H. DOANE

William H. Doane was born near Norwich, Conn., on February 3, 1832. He received his early education at Woodstock Academy in Connecticut, and later went to Cincinnati, where he took an active part in business and religious affairs. He was president of J. A. Fay & Co., wood machinery makers, for many years, and when the firm was consolidated with the Egan Company he relinquished his connection with the enterprise. At the Paris Exposition in 1889, Dr. Doane was one of the three Americans to receive an award of honor. His exhibit of woodworking machinery was one of the features of the display in the French capitol.

His chief interest was music, and he was granted the degree of Doctor of Music and wrote many compositions. His great interest in harmony caused him to take an extensive trip to many lands and in his world journey he collected marvelous groups of instruments. The collection aroused considerable interest among music lovers and Dr. Doane presented it to the Cincinnati Art Museum, where it is preserved.

He was elected to membership in the Society in 1885. He was

President Emeritus of the American Baptist Publishing Company of Philadelphia, Trustee of Denison University, Granville, Ohio, and was affiliated with a number of religious societies.

He died at the home of his daughter in South Orange, N. J., on December 23, 1915.

JAMES MAPES DODGE

James Mapes Dodge, Past-President of the Society, died in Philadelphia, Pa., December 4, 1915. He was born at Waverly, N. J., June 30, 1850. He was educated at Cornell University and at Rutgers, and had mechanical training in the shops of the Morgan Iron Works, New York City, and those of John Roach, the shipbuilder, at Chester, Pa., where he advanced to positions of foreman and superintendent of erection.

Shortly after the Centennial Exposition at Philadelphia, in 1876, he left the shipyard and later went to Chicago and joined with William D. Ewart, the inventor of the Ewart link belting, and his associates in the development of the chain business. At that time the application of chains to power transmission was exceedingly limited and their use in elevating and conveying machinery was practically unknown. Mr. Dodge was alive to the opportunity and new types of chain, new methods of manufacture and new conveying and elevating appliances were brought out in rapid succession. After this period of development Mr. Dodge entered into partnership at Philadelphia with Edward H. Burr, to represent the Ewart Manufacturing Company of Indianapolis, and out of this partnership grew in 1888 the Link-Belt Engineering Company in which Mr. Dodge carried out his idea of development along strictly engineering lines, with a highly-specialized engineering staff. Among other developments was the Dodge system of storage for anthracite coal, by which the coal is stored in large conical piles and handled entirely by machinery, both in and out of storage. The Dodge Coal Storage Company was formed, and in 1892 Mr. Dodge was made president of this company, and of the Link-Belt Engineering Company. In 1906, when these and allied companies were merged and known as the Link-Belt Company, he was elected Chairman of the Board of Directors.

For many years Mr. Dodge was a close friend of Frederick W. Taylor and was the first to adopt in its entirety, at the Philadelphia plant of the Link-Belt Company, the Taylor system of scientific management. He made many contributions to the literature of scientific management and in these as well as in his spoken words and in the practical application of the system in his own plants, he held to the necessity for the mutual advancement of employe and employer. He showed always the greater concern for the welfare, happiness and success of the employe. His relations with his employes were both cordial and intimate, and his influence was to inspire self help, initiative and ambition, as well as to aid them when needed in every possible way.

Mr. Dodge was deeply interested in civic affairs and was president of the Public Service Committee of One Hundred and a member of the Committee of Seventy in Philadelphia, and in other ways was closely identified with the best interests of the city.

He became a member of The American Society of Mechanical Engineers in 1884 and served the Society in many ways thereafter. He was Manager from 1891 to 1894; Vice-President, 1900 to 1902, and President in 1903. He had been Chairman of the Public Relations Committee and of the Sub-Committee on Administration and in 1908 was Chairman of the Nominating Committee. His interest and belief in young men were shown by his presidential address before the Society on Money Value of Technical Training. At the meeting in Germany, one of the two papers presented by this Society was by Mr. Dodge on Industrial Management. Two other papers by him contributed to other meetings were: The History of the Introduction of a System of Shop Management, and New Method of Stocking and Reloading Coal.

AUGUSTUS JAY DU BOIS

Augustus Jay DuBois, a member of the Society since 1881, was born April 25, 1849. He was graduated from the Sheffield Scientific School in 1869, and was awarded the degree of Civil Engineer in 1870 and the degree of Doctor of Philosophy in 1873. Later he studied mechanics for two years at the Mining Academy in Freiburg, Saxony, and from 1875 to 1877 he was professor of civil engineering and mechanical engineering at Lehigh University. In 1877 Professor DuBois was appointed professor of mechanical engineering in the Sheffield Scientific School and in 1884 was transferred to the professorship of civil engineering which position he occupied until his death.

Professor DuBois was the author of some of the best-known treatises on mechanics and stresses in the English language. His book on Graphic Statics, published in 1876, was largely instrumental in introducing to American engineers the graphic method of determining stresses in framed structures now so widely used. This was followed by his translations of Röntgen's Thermo-dynamics, Weyrauch's The Calculation of the Strength and Dimensions of Iron and Steel Construction, and Hydraulics and Hydraulic Motors and Heat, Steam and the Steam Engine from Röntgen's Mechanics. In 1883, his elaborate and original book on Strains in Framed Structures took its place as one of the most important contributions to engineering literature, being perhaps the first comprehensive treatment of the subject. A series of books on mechanics culminated in his Mechanics of Engineering published in 1901.

Professor DuBois was a member of the American Society of Civil Engineers, the American Institute of Mining Engineers, the Connecticut Society of Civil Engineers and the Society for the Promotion of Engineering Education. He died on October 19, 1915.

RAFAEL ESTRADA

Rafael Estrada was born in Havana, Cuba, on October 19, 1840. He received his early education in Cuba and later went to Lowell, Mass., where he was given private instruction. During 1855 and 1856 he served an apprenticeship in the machine shop of William Sellers & Company, of Philadelphia, and then resumed his studies in Cuba. In 1860 he entered the Southwark Foundry & Machine Company as an apprentice and rapidly rose to the position of erecting engineer, having charge of the erection and starting of sugar mills on several of the large plantations in Cuba and the Facala Refinery in Peru, S. A. In 1870 he became manager of the Grocers Sugar House in Philadelphia, and remained there until the house was closed in 1891. In 1892 he took charge of the sugar business of Bea Bellido & Company in Cuba, for whom he designed, completed and managed a molasses sugar house. From 1897 to 1904 he owned and operated zinc and lead mines in Joplin, Mo., but in 1904 he resumed his activities in the Cuban sugar industry. During this time he was active in the reconstruction and management of sugar mills on many large plantations and was retained as consulting engineer on several others.

Mr. Estrada became a member of this Society in 1904. He died at his home in Cuba on December 26, 1915.

HERBERT NICHOLAS FENNER

Herbert Nicholas Fenner was born in Providence, R. I. on March 13, 1843. He obtained his early education in that city, and after a few years experience in business he succeeded his father in the New England Butt Company. He served as treasurer of that company for many years and at the time of his death was president. He was also a director in the Industrial Trust Company and the Joslin Manufacturing Company.

Mr. Fenner took a great interest in public affairs, and was prominent in club life. He was a director of the Puritan Life Insurance Company. He became a member of the Society in 1891.

He died in Providence, R. I., on January 5, 1915.

WILLIAM H. GERRISH

William H. Gerrish was born in Lowell, Mass., in 1865, and was educated at the Lowell High School and the Massachusetts Institute of Technology. He served his apprenticeship at the Lancaster Slate Company, Lancaster, Mass., from 1888-1890. For several years he held a position in the drawing room of the Massachusetts Cotton Mills, Lawrence, Mass., and at the Fulton Bag Mills, Atlanta, Ga.

During the Spanish-American War he was associated with the Ordnance Department of the War Office, Washington, D. C.

In 1899, he became superintendent of the Barber Flax Spinning Company, Paterson, N. J., from which he went to the Dolphin Jute Mills, Paterson, N. J. Four years later, he came to New York as superintendent of the Commercial Twine Company. In 1910, he was appointed smoke inspector for the State of Massachusetts, which office he retained to the date of his death.

He was elected to membership in the Society in 1901. He was a member of the Order of Masons and the Royal Arcanum. He died on July 15, 1915, in Malden, Mass.

JOHN CHARLES WILLIAM GRETH

John Charles William Greth was born in Buffalo, N. Y., in 1874. He attended the public schools of that city and was graduated from the Buffalo Central High School in 1893. In the same year he entered Cornell University and was graduated in 1897 with the degree of Mechanical Engineer. After graduation, he began his engineering work by installing and operating for a few months a power plant at a summer resort on Lake Erie. From 1898 until 1902 he installed pumping machinery, designed special machinery, operated a power plant and installed and operated refrigerating and ice making plants.

In 1902, he entered the service of Wm. B. Scaife & Sons Co., Pittsburgh, Pa., as manager of the water softening and purification department, and from that year until his death his time was devoted to the development of apparatus and methods for the softening and purification of water for all purposes.

Mr. Greth took out sixteen patents on improvements in water purifying apparatus, several of which embodied radical features. He occupied a leading position in the field of water purification, and his forceful presentation contributed very materially to the advancement of the science and application of water purification for industrial use.

He was the author of a number of articles on water purification published in the engineering press, and he read several papers on the same subject before various engineering societies.

He was a member of the American Society of Civil Engineers, the American Institute of Chemical Engineers, Engineers Society of Western Pennsylvania, and of the American Chemical Society.

He was elected a member of the Society in 1907. He died at Gibsonia, Pa., on August 7, 1915, after a short illness.

JAMES TAGGART HALSEY

James Taggart Halsey was born in Philadelphia, Pa., in 1854, and was educated at the Episcopal Academy in that city. He was apprenticed in the Pennsylvania Railroad shops at Altoona, following which the railroad placed him in charge of its signals, and he made a number of basic inventions of types of this railroad auxiliary. He resigned from service with the Company to take up a position with the Talbot Works, Richmond, Va., in which he continued for seven years. He later specialized in portable machine tools, maintaining a shop in Philadelphia. He was the inventor of the Halsey motor truck for trolley roads; this was a pioneer invention in this field, and one in which several prominent engineers displayed interest.

Mr. Halsey was elected to membership in the Society in 1885. He died in Philadelphia, Pa., on April 27, 1915.

JOHN BROWN HERRESHOFF

John Brown Herreshoff was born in Bristol, R. I., in 1841. An attack of infantile glaucoma destroyed his sight, but did not prevent him from obtaining an education in the schools of his native town. His bent for mechanism revealed itself at an early age, and the handicap of total blindness seemed to serve as a spur to his tireless energy. He built and rebuilt miniature craft, which laid the foundation of the skill and knowledge which he later turned to good account.

At the age of twenty, he began the construction of larger craft, and in 1863 he embarked on naval construction as a business, which he carried on for more than fifty years.

In 1872, he entered into partnership with his brother, and together they designed and constructed for the U. S. Government the first torpedo boat, The Lightning. In 1879, he and his brother incorporated the Herreshoff Manufacturing Company, of which he was president and treasurer from its formation to the time of his death. In 1892, the Herreshoff Manufacturing Company took up the designing and construction of yachts, with special reference to racing craft for the New York Yacht Club, and built the Vigilant, Defender, Columbia, Reliance and Resolute, which testify to what height of perfection his firm has carried the science of naval designing and construction.

Mr. Herreshoff was a member of the Institute of Naval Architects of London and a member of the Society since 1884. He died on July 20, 1915.

EBENEZER HILL

Ebenezer Hill, elected a member of the Society in 1892, was born in South Norwalk, Conn., on October 5, 1849. He graduated from Wesleyan University with the degree of B.A. in 1870, and with the degree of M.A. in 1891. From 1870 to 1880 he participated in the design, construction and installation of steam pumping machinery and steam engines of various types. In 1880, he became the responsible executive head of the Norwalk Iron Works Company, and shaped the business to the manufacture of air compressors and allied machines. He held this position up to the time of his death, which occurred on February 26, 1915.

JOSEPH AUSTIN HOLMES

Joseph Austin Holmes was born at Laurens, S. C., in 1859. He was educated at Cornell University, from which he was graduated Bachelor of Science in 1881. He became professor of geology and natural history at the University of North Carolina in the same year, and continued as such until 1891. He was state geologist from 1891 to 1903. In 1904 he was appointed by President Roosevelt as chief of the U. S. Geological Survey Laboratories for the testing of fuels and structural materials, rendering noteworthy services. President Taft appointed him in 1910 head of the newly-created U. S. Bureau of Mines, and under his management great progress was made in perfecting methods of saving life in mines. The chief work of the Bureau under his direction has been the investigation of the cause of coal mine explosions, and one of his most important discoveries was that the dust from bituminous coal was more dangerous to miners than firedamp.

He received the degree of Doctor of Science from the University of Pittsburgh, and also of Doctor of Letters from the University of North Carolina, both being conferred upon him in recognition of effort in the mining industry.

He was a member of the American Institute of Mining Engineers, American Society for Testing Materials, the National Conservation Commission, and other organizations. He was elected to membership in this Society in 1908. He died on July 13, 1915.

BENJAMIN FRANKLIN ISHERWOOD

Benjamin Franklin Isherwood was born in New York City on October 6, 1822. He was educated at Albany Academy and afterward served under David Matthews, master mechanic of the Utica and Schenectady Railroad. He was promoted to the civil engineer's office and, on the completion of the road, he went to work on the Croton aqueduct. After this was completed, he worked on the construction of the Erie Railroad under Charles B. Stuart, division engineer, who later became engineer in chief in the navy, and it was through his influence that Mr. Isherwood entered the navy in 1844. Later he was assigned by the U. S. Treasury Department to work on the construction of lighthouses, and was sent to France to superintend the construction of lighthouse lenses there from his own designs. At the outbreak of the war with Mexico, he served on board the Princeton, the first American screw steam vessel built by Ericsson for the government as an experiment. He was promoted to be chief engineer of the Spitfire, and he took part in every action in which the American fleet was engaged during the war.

His experiments in the expansion of steam on board the U. S. S. Michigan in 1859 almost revolutionized the methods of using steam. He designed the engine of the U. S. cruiser Wampanoag, which was built in 1868 and was the fastest steamship in the world at that time, having a speed of 17¾ knots.

Mr. Isherwood was chief engineer in the navy from 1861 to 1869, covering the entire period of the Civil War, when more than six hundred steam vessels and three thousand engineers were in the service.

During the years 1870 and 1871, he was stationed at the Mare Island Navy Yard, California. His experiments there with screw propellers are regarded as among the greatest contributions to engineering. He was retired as chief engineer with the rank of rear admiral on June 6, 1884. He was the author of many engineering works, some of which have been used widely as text books in technical schools.

Mr. Isherwood was elected an honorary member of the Society in 1894. He died at his home in New York City on June 19, 1915.

JOHN LOYD

John Loyd was born in Newton, Mass., on May 1, 1837. He received his education in the public schools, and learned the machinist and engineering trades.

At the outbreak of the Civil War he entered the United States Navy. He was commissioned as acting third assistant engineer and was later promoted to first assistant engineer. On May 7, 1867, he was ordered to the Portsmouth Navy Yard as assistant engineer, and was granted an honorable discharge December 27 of the same year.

Shortly afterward he started into business in New York City in the manufacture of machinery, knives and dies under the firm of McLoughlin, Grover and Loyd. On the death of his partners he continued the business under the name of the John Loyd Company.

Mr. Loyd joined the Society in 1899 and was also a member of the Military Order of the Loyal Legion, the Navy League, the Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers and the Metropolitan Museum of Art. He died at his home in Brooklyn, N. Y., on October 5, 1915.

GUSTAVUS TYLER LUCKETT

Gustavus Tyler Luckett was born at Owensboro, Ky., on February 26, 1870. He received his early education in the schools of Owensboro and at Trinity College, near Louisville, Ky. From 1888 to 1896, he held the positions of machinist and pattern maker, and molder superintendent, with the Novelty Foundry and Machine Company at Owensboro, Ky. From 1898 to 1900 he was assistant superintendent in charge of construction for the New York Edison Company. He then served the Best Manufacturing Company as its New York representative. From 1901 to the time of his death, he was with the M. W. Kellogg Company. He had active charge of designing steam and hydraulic piping, machines for the manufacture of Van Stone joints and improvements, welding flanges, forge welding shells, piping, piping outlets and construction and erection of piping systems. Mr. Luckett died at his home in New York City on July 16, 1915.

DWIGHT E. LYMAN

Dwight E. Lyman was born in Marshall, Oneida County, N. Y., on October 12, 1845. He was educated in the Deansville schools, and served an apprenticeship at the Willowvale Machine Works of Utica. Later he was employed for four years as superintendent and draftsman by Keeney Brothers of Manchester, Conn. At the age of twenty, he removed to Hartford, and soon entered the employ of Asa S. Cook, serving as superintendent and mechanical engineer from 1875 to 1882. After this he held for one year a similar position with the Syracuse Screw Company, returning in 1883 to the Asa S. Cook Company as superintendent, which position he held at the time of his death.

Mr. Lyman was elected a member of the Society in 1901. He died at his home in Hartford, Conn., on July 8, 1915.

ROBERT MC ARTHUR

Robert McArthur, for twenty-three years head of the Pepperell Manufacturing Company of Biddeford, Me., died on December 23, 1914.

Mr. McArthur was born in Ashton, England, May 18, 1838 and came to America in 1842. He began his mill career in 1853 with the Globe Manufacturing Company of Woonsocket, R. I. and it was here that he learned the rudiments of the cotton mill business. He remained with this company for three years, and from 1856 to 1870 was employed in different capacities in various mills of southern New England.

In 1870 he was appointed superintendent of the Millville Manufacturing Company's mill at Millville, N. J. In 1873 he accepted the position of agent for the Manchaug Mills at Sutton, Mass., where he remained for ten years. He then became agent of the Grosvenordale Mills at Grosvenordale and North Grosvenordale, Conn., which position he held for three years. In 1887 he became agent for both the Pepperell and Laconia corporations of Biddeford, Me., continuing in this capacity after the consolidation of the two companies; he resigned this position in 1910.

Mr. McArthur was president of the New England Textile Club for many years. He was a Mason, a member of Sheridan Post G. A. R., and was connected, during the Civil War, with the engineering corps of the army. He was elected to membership in the Society in 1894.

JAMES MC BRIDE

James McBride, a member of the Society since 1886, was born on April 27, 1836. He learned the trade of pattern maker, and from 1858 to 1865, he was employed by the Duquesne Engine Works at Pittsburgh at pattern making and the erection of machinery on steamboats. During this period he assisted in the erection of the machinery on thirty-seven boats. At this time he obtained a license for engineer of the Second Class on the western rivers.

In the four years that followed, he established in Pittsburgh a pattern shop of his own in which he made patterns and mechanical drawings and designed machinery. He discontinued this, however, to take a position as draftsman for the Root Steam Engine Company of New York, but later returned to the Duquesne Engine Works as draftsman and designer. In 1876, he entered the employ of the

New York Dye Wood Extract and Chemical Company of Brooklyn, as chief engineer. Three years later, he became superintendent of the Stamford Manufacturing Company in Stamford, Conn.

Mr. McBride died at his home in Stamford, Conn., on April 14, 1915.

JAMES FINNEY MC ELROY

James Finney McElroy was born in Greenfield, Ohio, on November 25, 1852. He attended the Salem Academy at South Salem, Ohio, in 1869 and the Bloomingburg Academy at Bloomingburg, Ohio, from 1870 to 1872. He was graduated from Dartmouth College in 1876 with the degree of A.B. and received the degree of A.M. from the college in 1879.

From 1876 to 1880, he was principal teacher of the Indiana Institute for the Blind at Indianapolis and from 1880-1887 he was superintendent of the Michigan Institution for the Blind, at Lansing, Mich. For the latter institution, he designed and constructed the heating and power plant.

In 1887, he organized at Buffalo, the McElroy Car Heating Company, which operated under its own patents. In 1889, this concern was combined with the Sewell Car Heating Company, forming the Consolidated Car Heating Company in Albany, of which Mr. McElroy was consulting engineer and acting president to the time of his death. Mr. McElroy became a member of the Society in 1900. He died on February 10, 1915.

WILLIAM MC INTOSH

William McIntosh was born August 20, 1849, and received a common school education. From 1867 to 1870 he served an apprenticeship with the Chicago, Milwaukee and St. Paul Railway. He was employed by the Chicago and North Western Railway for 27 years, in the capacities of foreman at Waseca, Minn., and Huron, S. Dak., and of master mechanic at Winona, Minn. He left this company to become superintendent of motive power with the Central Railroad of New Jersey. When in 1909 he resigned on account of ill-health, he had given forty years of active railroad service. He died on March 16, 1915.

Mr. McIntosh joined the Society in 1902 and was also a member of the New York Railroad Club, the Canadian Society and the Engineers' Club.

ROBERT A. MC KEE

Robert A. McKee was born on September 12, 1873, in Towanda, Pa. He graduated from Lehigh University with the degree of M. E., in 1895. For a short time after graduation, he was employed by the Brooks Locomotive Works of Dunkirk, N. Y., as draftsman and designer. He then entered Cornell University in the course in marine engineering, receiving the degree of M. M. E. in 1897. Returning to the Brooks Locomotive Works, he worked in the capacity of designer and in 1899 became connected with the Baldwin Locomotive Works, remaining there until 1900, when he took a position with the Holly Manufacturing Company of Lockport, N. Y. Here he commenced as a tracer and was rapidly advanced to the position of leading designer. After fifteen months he left this position to enter the employ of the Westinghouse Machine Company at East Pittsburgh, Pa.

In 1904 he was engaged by the Allis-Chalmers Company to take charge of the development and construction of steam turbines, and it was due to his efforts more than those of any other individual that the steam turbine developed along lines which have proved to be based on correct fundamentals.

His advice on steam problems was sought by leading engineers, and his reputation as a steam turbine engineer extended to foreign countries. Mr. McKee was elected to membership in the Society in 1908 and was also a member of the American Institute of Electrical Engineers. He died in New York City on September 5, 1915.

ARTHUR S. MANN

Arthur S. Mann was born in West Medway, Mass., on September 4, 1867. He received his early education in the schools of Medway, and graduated from the Massachusetts Institute of Technology with the degree of S.B. in 1888. The first two years out of college he spent with the George F. Blake Manufacturing Company. Following this he was with the West End Street Railway Company in Boston, Mass. From 1892 to 1894, he served the E. P. Allis Company of Milwaukee as mechanical engineer. In 1894, he started a machine shop with F. E. Lammert in Chicago, building special machinery and conducting a general engine repair trade. He was vice-president of this company up to the time of his death, although he had taken no active part in the business since 1897.

In 1897, he became engineer of construction of the Ninety-sixth Street power plant and operating chief engineer of the various plants for the Metropolitan Street Railway Company of New York City. In 1901, he accepted a position with the Sydney Street Railway Company in Australia as construction engineer of power plants. In 1903, he became associated with the General Electric Company in Schenectady, N. Y., where he was engineer in charge of construction of their power plants and of the steam, air and water distribution of their entire works. Also, he was in charge of the installation of the new powdered coal system and had charge of the design of their new furnaces and boilers.

Mr. Mann became a member of the Society in 1900. He died on October 3, 1915.

FREDRIK V. MATTON

Fredrik V. Matton was born in Stockholm, Sweden, on June 6, 1856. He received his technical training at Stats College, and served his apprenticeship as a machinist in his native country. He came to this country early in 1882, first securing employment at Altoona and later at the Harrison Sugar Refinery at Philadelphia. At the latter plant he had charge of the repairs and construction of general machinery.

In June, 1885, Mr. Matton became employed at the Camden Iron Works in Camden, N. J., as chief engineer, which position he held until his last illness.

Mr. Matton became a member of the Society in 1892. He died at Atlantic City, N. J., on December 3, 1915.

WALTER K. MITCHELL

Walter K. Mitchell was born October 13, 1866, in Glasgow, Scotland, and came to this country at an early age. He was educated in the schools of Pittsburgh, and served an apprenticeship as machinist with the Jones & Laughlin Steel Company. From 1889 to 1899 he was employed as sales engineer by Best Fox & Company of Pittsburgh, piping engineers and contractors, going to Philadelphia as their eastern representative in 1893. In 1899 he entered into the partnership of W. K. Mitchell & Company, specializing in high pressure piping and its accessories. When this partnership was changed to a corporation in March 1909, Mr. Mitchell was elected president and treasurer. He continued actively in these offices up to the day of his death, which occurred on November 6, 1915.

He was elected to membership in the Society in 1913.

EDWARD THOMAS MORRIS

Edward Thomas Morris was born in Lutterworth, England, in 1863. He was educated in public and private schools in England, after which he served his apprenticeship at Ruston & Proctor, engineers and boiler makers, Lincoln, England, from 1880 to 1884. From 1884 to 1885, he was under instruction at the firm of J. & J. Thompson, marine engineers and boiler makers, Glasgow, Scotland.

From 1887 to 1888, he was outside erector of machinery for the Union Iron Works. From that date until 1899 he was superintendent of the marine engineers department of the Union Iron Works, having charge of the installation of boilers and machinery on the U. S. S. Charleston, San Francisco, Monterey, Olympia and Oregon. From 1899 to 1900, he was assistant to the marine superintendent of the Pacific Mail Steamship Company, and from 1900 to 1902, general superintendent of the Fulton Engineering and Shipbuilding Works.

During the years 1902 to 1905, he was superintending engineer of the Oceanic Steamship Company, and in 1906 he became superintendent of construction with the Tracy Engineering Company. From 1908 to 1911, he was engineer of construction for the Associated Pipe Line Company. In 1911, he was appointed manager of the Pipe Line Department of the Associated Oil Company, in which position he was in charge of the construction and operation of pipe lines and pumping stations of both companies.

Mr. Morris was a member of the Society since 1912. He died in Oakland, Cal., on May 14, 1915.

HENRY G. MORRIS

Henry G. Morris was born in Philadelphia, Pa., May 25, 1839 and graduated from Haverford College. He engaged in active manufacturing business in early life, and when still a very young man was a member of the celebrated firm of Morris, Tasker & Co., who were the first manufacturers of wrought iron pipes and boiler flues in this country. A few years later he became the sole owner of the Southwark Foundry, where some of the largest blowing engines, pumps and other heavy machinery were constructed, and at that time he was justly regarded as one of Philadelphia's greatest captains of industry.

Mr. Morris was one of the first directors of the Pennsylvania Steel Company, and was one of the few men in this country, like Alexander Lyman Holley, to recognize the importance and value of the Bessemer process. He was a leader in the design and manufacture of the most diversified kinds of machinery used on sugar plantations and in refineries, and in gas plants and waterworks, and he was among the first in this country to recognize the value of compound engines in marine engineering.

He early became interested in electrical engineering, and will undoubtedly be best remembered in connection with Mr. Salom in the invention and development of the electric vehicle for which, twenty years ago, they received the gold medal of the "Times-Herald" Motorcycle contest in Chicago, 1895 (the birth of the automobile), and the John Scott Legacy Medal of The Franklin Institute, Philadelphia. He took out numerous patents in connection with this vehicle and for storage batteries.

He was a member of many national societies and associations, including this Society which he joined in 1882 and of which he was a Vice-President from 1887 to 1889; the Engineers' Club of Philadelphia, of which he was a past-president; the American Institute of Mining Engineers, the American Society of Civil Engineers, The Franklin Institute of Philadelphia, and the Union League of Philadelphia, of which he was the oldest living member in point of time, with a single exception, having been elected a member in 1863. Mr. Morris died on January 19, 1915.

TEILE HENRY MÜLLER

Teile Henry Müller, who became a member of the Society in 1888, was born January 18, 1841, at Grossensiel-Oldenburg, Germany, and was educated at the Polytechnic School of Hanover, completing the course in mechanical engineering in 1862. He then entered the service of the North German Lloyd Steamship Line as ship engineer. In 1865 he came to New York and was engaged first as machinist and later as draftsman by the Root Steam Engine Company. He left this firm in 1866 to accept a position as superintendent engineer with the Convex Weaving Company, designing and building their mill and machinery. He then took the position of superintendent with the Eagle Pencil Company, to redesign their machinery and revise the processes of manufacturing pens, pencils and other articles.

In 1877 he entered the employment of S. S. Hepworth & Company, sugar engineers, as superintendent. He designed the California Sugar Refinery in San Francisco for Claus Spreckels and the Belchers Refinery in St. Louis, and built a large part of the machinery for these factories. Later he designed and built the Spreckels Refinery in Philadelphia, the National Sugar Refinery in Yonkers, and designed and partly built the Camden Sugar Refinery. In 1900 he was engaged by the Federal Sugar Refining Company as constructing engineer and built their works in Yonkers in 1914. Mr. Müller continued in this position until the time of his death, which occurred on September 21, 1915.

EDGAR H. MUMFORD

Edgar H. Mumford was born at Groton, Mass., on September 20, 1862. He was graduated from Massachusetts Institute of Technology in 1886 in Mechanical Engineering.

In 1886, he entered the service of the Union Pacific Railroad at Omaha, Neb., in the motive power department, and left there after promotion to master mechanic of the Leavenworth Division, to become superintendent of the Russell Wheel & Foundry Company, Detroit, in 1889. After three years in this position, he became associated with Henry R. Worthington, Inc., Brooklyn, N. Y., and was shortly afterward placed in charge of the new foundry built by this company at Elizabeth, N. J. He left this position to become manager of the New York office of Bement, Miles & Company of Philadelphia, where he remained until 1895, when he started in business for himself.

With Harris Tabor and Angus Sinclair, he purchased the business of the Tabor Manufacturing Company from Manning, Maxwell and Moore, and remained with it as secretary and treasurer for ten years. In 1900, the Company's plant was moved to Philadelphia, and in 1905 he resigned from the firm to form the E. H. Mumford Company, builders of molding machines, of which he was president and treasurer. In 1909, he became vice-president and general manager of the Mumford Molding Machine Company, where he remained until shortly before his death, at which time he was arranging to again enter the molding machine business under his own name.

Mr. Mumford was for twenty years an inventor and builder of molding machines known as "Mumford" machines, and was a recognized authority on machine molding.

He was elected to membership in the Society in 1887 and was also a member of the Engineers' Club, the Machinery Club, Technology Club, and the Art Club of Philadelphia. He died at his home in Plainfield, N. J., on April 18, 1915.

FRANK RUSSELL PACKHAM

Frank Russell Packham was born at Hadley, Mich., on May 11, 1855. He received his early education in Canada, and his first business training with his father who was a miller. He also served an apprenticeship in a sewing machine factory, and learned the machine and pattern making trades.

When he was 18 years old, his parents moved to Springfield, Ohio, where he was employed as a machinist for the Wardell-Mitchell Company, now a part of the International Harvester Company. In 1878, he became superintendent and experimenter for the Baker Drill Company in Mechanicsburg, Ohio, and manager and designer of turner's tools for the Packham Crimper Company until 1886. In 1887, he returned to Springfield and identified himself with the Superior Drill Company devoting his time to designing and pattern making. Upon the formation of the American Seeding Machine Company, he was made a director and manager of the experimental department. This latter position he held until the time of his death.

As an inventor, he contributed as many as 150 improvements on various agricultural implements, most of which are now manufactured by the American Seeding Machine Company. Probably his most important work was the invention and development of the "single disc" drill.

In 1900, Mr. Packham was appointed as a representative of the United States Government to tour the world in the interests of the Foreign and Domestic Bureau of Commerce. In 1909, he was appointed mechanical guide to the Honorable Commercial Commission of Japan in its visit to this country.

Mr. Packham became a member of the Society in 1913. He died at Springfield, Ohio, on January 1, 1915.

JOHN PARKER

John Parker was born in Mansfield, England, on March 28, 1864, and came to this country in 1887. For four years, he was employed in the drafting department of the Corliss Steam Engine Company in Providence, R. I., and in 1891, he accepted a position with the Brown and Sharpe Manufacturing Company. In 1893, he took active charge of their miller designing, and in addition to this position also held that of assistant chief draftsman, from 1895 to 1902. In the latter capacity he developed executive ability in putting work through correctly and efficiently, and many patents were granted to him chiefly in connection with his work on millers. He was in charge of the miller designing for this company up to the time of his death, which occurred on July 23, 1915. He became a member of the Society in 1909.

FRED STARK PEARSON

Fred Stark Pearson, who lost his life on May 7, 1915, in the Lusitania disaster, was born July 3, 1861, in Lowell, Mass. He graduated from Tufts College in 1885 with the degree of E.E. and received the degree of M.E. in 1886.

In the early part of his career, after leaving college, he held the positions of chemist for the Boston Butter Company, superintendent of the Blue Ridge Gold Mining Company in Virginia, mining expert surveying copper mines in Texas and gold deposits in Brazil, manager of the Somerville Electric Light Company, consulting engineer for the Woburn Electric Light Company, and for the Chandler Electric Light Company in Halifax, consulting engineer for the American Aluminum Company and chief engineer in the steam and electric departments of the West End Street Railway Company in Boston.

His later enterprises, results of his organization, may be classed as electric light and power companies, electric tramway companies, telephone companies, gas companies, steam railroads, irrigation and land companies, lumber manufacturing companies and mining and chemical companies. These companies were centered principally around the following cities: Mexico City, Mexico; Rio de Janeiro and Sao Paulo, Brazil; Barcelona, Spain; Toronto, Winnipeg and Niagara Falls, Canada; and El Paso and Juarez in Texas and Northwestern Mexico.

In steam railroads, he was principally interested in the Mexico and Northwestern Railroad from Chihuahua to El Paso in the center of several million acres of timber and mining land which his companies own, and the Denver & Salt Lake Railroad, or Moffatt System. The details of his numerous projects, however, were handled by his New York staff, a corporation called the Pearson Engineering Corporation, Ltd., in conjunction with the local staffs in each of the cities where his public utilities were located.

Dr. Pearson was a member of a large number of technical societies, including the American Society of Civil Engineers, the American Institute of Electrical Engineers, the American Society of Naval Engineers, the Institution of Electrical Engineers (England); the American Institute of Mining Engineers, the American Electro Chemical Society, the New York Railroad Club and the American Forestry Association. He was elected to membership in this Society in 1899.

JAMES EDWIN QUIGLEY

James Edwin Quigley, whose death occurred in Pittsburgh, Pa., on April 11, 1915, was born in Center County, Pa., February 9, 1873. After graduating from Pennsylvania State College, he was associated for several years with the Buffalo Forge Company of Buffalo, N. Y., and then entered the employ of the B. F. Sturtevant Company, Philadelphia, Pa. In 1899 he was appointed district manager for the B. F. Sturtevant Company, with headquarters in Pittsburgh. He resigned this position in 1902 to become manager of the insulation department of the Armstrong Cork Company, Pittsburgh, Pa., of which he had charge since its formation.

Mr. Quigley was a member of the Board of Directors of the Armstrong Cork Company and the Armstrong Cork and Insulation Company. He was a member of the Duquesne Club, the University Club and the Edgewood Country Club of Pittsburgh, the Machinery Club of New York, the Phi Kappi Sigma fraternity and the Ross Mountain Park Association. He was also a member of the American Society of Refrigerating Engineers. He joined this Society in 1900.

THOMAS L. RANKIN

Thomas L. Rankin was born at Ripley, Ohio, on June 16, 1839. He received his education at Iberia College at Iberia, Ohio, from which he graduated in 1855. In 1869, he began to build isolated refrigerating cars in the West with Barney and Smith of Dayton, Ohio. In 1873, he built fifty of the refrigerator cars at the Wasson Works in Springfield, Mass. Previous to this date, he had laid iron floors for refrigerating breweries and packing houses. From 1873 to 1877, he was superintendent of the Arctic Ice Machine Company, with many plants in Texas, Arkansas and Louisiana. During this time and through 1891 he continued the construction of ice machines at the Phoenix Works, Houston, Tex., at the Atlas Works, Indianapolis and the Reading Iron Works, Reading, Pa. He later had charge of the ice machine department of the Pennsylvania Iron Works Company of Philadelphia, Pa.

Mr. Rankin became a member of the Society in 1892. He died at his home in Sacketts Harbor, N. Y., on November 12, 1915.

GEORGE T. REISS

George T. Reiss was born in Cincinnati, Ohio, on December 6, 1849. He was educated in the common schools and received private instruction in drawing, designing and mechanics.

In 1877, he became draftsman at the old Cope and Maxwell shops in Cincinnati, but in 1878 he was appointed master machinist in charge of the engineering department of the Niles Tool Works, which was moved from Cincinnati to Hamilton, Ohio, in 1871. He was chief mechanical engineer of the concern, later becoming superintendent of the drafting department and subsequently being elected to the board of directors, of which he was also secretary. He became vice-president of the directorate, which position he held until the time of his death. Mr. Reiss died at his home in Hamilton on May 5, 1915. He was elected a member of the Society in 1890.

PAUL G. ROESTI

Paul G. Roesti was born in Berne, Switzerland, in April, 1878 and received his early education in the public schools of his native city. After graduating in mechanical engineering from the technical school at Berne, Switzerland, in 1896, he was employed for two years in the engineering offices of Frickart in Munich, Bavaria, and one year with Sulzer Bros. in Winterthur, Switzerland, working on Corliss and poppet valve steam engines.

In 1899 he entered the Swiss Technical High School at Zurich, Switzerland, from which he graduated in 1903. He then came to this country and was employed by the Buffalo Forge Company as a designer of steam engines, centrifugal pumps and blowers. In 1904 he took a position with the Beckstrom Smith Steam-Turbine Company and the Filer and Stowell Company as mechanical engineer and designer of steam turbines and steam engines, remaining with them until 1907 when he became a designing engineer with the A. O. Smith Company, Milwaukee, Wis., manufacturers of automobiles. He was later made chief engineer of this company.

In 1911 he returned to Switzerland to accept a position with Sulzer Bros. as chief designer in the high-speed Diesel engine department, and he there designed and standardized several new types of engines. He gave up this position in 1915 on account of poor health. He died in California on December 23, 1915.

Mr. Roesti was a member of the German Engineers' Society, and the Swiss Engineers' and Architects' Society. He became a member of this Society in 1914.

WILLIAM F. SARGENT

William F. Sargent was born in New Haven, Conn., on June 27, 1882. He received his education in the public schools of New Haven and in the Boardman Training School, from which he graduated in 1901. From September, 1901, until May, 1902, he was in the employ of the Winchester Repeating Arms Co., where his work consisted of mechanical drawing, surveying for new buildings and roads, laying out machines in factory additions and general machine design. In June, 1902, he accepted a position with the Bigelow Company, with which company he was connected until the time of his death. For five years he was chief draftsman, and later was made sales engineer and estimator. Mr. Sargent became a member of this Society in 1914. He died at his home in New Haven, Conn., on December 7, 1915.

FRANCIS WINTHROP SCARBOROUGH

Francis Winthrop Scarborough was born in Cincinnati, Ohio, on September 6, 1865. He received his preparatory education in the public schools of Cincinnati and graduated from the Rensselaer Polytechnic Institute in 1888. Soon after this, he entered the service of the Chesapeake and Ohio Railway Company as engineer, from which service he resigned in 1908. While with the railway he became engineer of bridges and signals, and at the time he resigned was chief engineer of maintenance of way and structures. Immediately after his resignation he became actively engaged in the operation of coal mines in the New River region of the Chesapeake and Ohio Railway Company. Here he held the position of general superintendent in charge of the operation and development of eight coal companies controlling 30,000 acres of coal land and shipping at that time over one million tons annually.

He was a member of the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Society for Testing Materials, and the American Railway Association. He became a member of this Society in 1905.

Mr. Scarborough died in New York City on December 24, 1915.

JOHN M. SHERRERD

John M. Sherrerd was born at Scranton, Pa., on February 26, 1859. He graduated from Lafayette College in 1878 with high honors, and he took a course at the Columbia School of Mines during the two years following. He then became chemist for Ario Pardee at Secaucus, N. J., continuing for two years, when he became connected with the Troy Steel and Iron Company where he remained thirteen years. He was engaged first as chief chemist, then metallurgist, and later he ran the blast furnace.

In 1895, he became connected with the Taylor Iron and Steel Company as general sales agent. At the time of his decease, he held the position of assistant to the president of the Kennedy Stroh Corporation of Pittsburgh, Pa.

He was a member of the following societies: the American Society of Civil Engineers, the American Institute of Mining Engineers, the American Society for Testing Materials, the Lake Superior Mining Institute, the National Geographical Society and the Engineers' Club of New York. He became a member of this Society in 1905. He died at his home in Easton, Pa., on April 16, 1915.

GEORGE RIPLEY STETSON

George Ripley Stetson was born in Brooklyn, Conn., in 1837. His early education was received in Brooklyn, Conn., and in Florence, Mass., and his subsequent education he gained entirely through his own efforts and without attendance at school.

In 1855, he entered a machine shop in Northampton as an apprentice. After completing his apprenticeship, he engaged in the manufacture of Britannia ware. In 1864, he entered the employ of O. F. Winchester, manufacturer of rifles and ammunition, where he remained eight years. In 1877, he became superintendent of the Morse Twist Drill and Machine Company, New Bedford, Mass., and while in the service of this company and the O. F. Winchester Company he patented a number of devices which proved of value, receiving thirty-six patents for his inventions. In 1890, he became president and general manager of the New Bedford Gas & Edison Light Company.

He was a member of the Sutton Commandery, K. P., and a charter member and past patron of the New Bedford Chapter, No. 49, O. E. E. He was an active member of the National Electric Light Association and the Association of Edison Illuminating Companies, frequently representing these organizations as a delegate.

He was a charter member of the Society. He died on July 26, 1915.

R. PAUL STOUT

R. Paul Stout was born at Bethlehem, Pa., in 1869. He was educated at a private school at Audenried, at the Hill School of Pottstown, and later at Lehigh University, from which he was graduated in 1891 with the degree of M. E. During vacation periods, he served as a machinist in the shops of the Jeansville Iron Works.

His first position after graduation was that of mechanical engineer with the Lehigh & Wilkes-Barre Coal Company at Audenried. He entered the employ of the Bethlehem Steel Company as assistant superintendent of the armor plate department in 1894, when the armor for the United States battleships Brooklyn, Oregon, New York and Iowa, and the Russian ships Petropovlovsk, Admiral Siniavan, Admiral Oushakoff and Rostislav, and also the first large armor plate vault for the Philadelphia Savings Fund Society, was being manufactured by the company. During this time, too, face-hardened armor was first manufactured in this country and the best

Harveyized armor in the world up to that period was developed at the Bethlehem plant.

In August, 1897, on the formation of the Bethlehem Company's ordnance department, Mr. Stout was transferred to that department, and two years later he assumed charge of the development of the first piece of ordnance mechanism of its own design which the company sold. Under engineer-of-ordnance Lieutenant Meigs, he had charge of all the experimental and development work in this department until September, 1910, when he himself was appointed engineer of ordnance, which position he held up to the time of his death.

Within his period of service as engineer of ordnance, the Bethlehem Steel Company secured the largest naval ordnance order placed in this country, in the complete ordnance equipment and armor for the Argentine battleships Moreno and Rivadavia, and it was in the last two or three years that the company attained its present position in the manufacture of all kinds of ordnance and munitions.

Mr. Stout was elected to membership in the Society in 1906. He was also a member of The Franklin Institute. He was killed on August 25, 1915, as the result of the accidental detonation of a high explosive shell.

FREDERICK WINSLOW TAYLOR

Frederick Winslow Taylor, Past-President of the Society, was born in Germantown, Pa., in 1856. His primary education was received in this country and in France and Germany. He was being prepared for Harvard at Phillips Exeter Academy but his eyesight became impaired and he began an apprenticeship with William Sellers and Company. He finished his apprenticeship in 1878, having served four years as a patternmaker and machinist. He then entered the machine shop of the Midvale Steel Company, and was promoted to shop clerk, and later given charge of the toolroom, advanced to gang boss, assistant foreman and then foreman of the machine shop. Next he became master mechanic in charge of repairs and maintenance of the work, and in 1884, chief engineer.

While at Midvale, he secured, by evening study, the degree of M. E. from Stevens Institute of Technology. Later he received the degree of Sc. D. from the University of Pennsylvania.

He remained with the Midvale Steel Company until 1890, making many improvements in machinery and methods. He designed the great steam hammer of the Midvale Steel Company, the largest successful hammer ever built in the United States. In 1890 he became manager of The Manufacturing Investment Company, operating large paper mills in Maine, where he remained until his three-year contract expired.

He then began, as consulting engineer, to introduce his principles of organization and management into various establishments about the country. During this period and in connection with this work he made many valuable improvements and inventions. Mr. Taylor's success in producing increased shop efficiencies at other plants led the Bethlehem Steel Company to retain him to increase their machine shop capacity.

Experiments made in this connection in conjunction with Mr. Maunsel White led to the discovery of the Taylor-White process of heat treatment of tool steel, which increased the cutting capacity 200 to 300 per cent. This process and the tools treated by it are now used in almost every machine shop in this country and abroad.

It was during his connection with the Bethlehem Steel Company that Mr. Taylor's ideas regarding management took concrete form, and it is probable that during that time he first recognized the great possibilities of the broader application of the principles according to which he had been working, and realized the results that would be attained if these principles should become generally adopted throughout our industries. Having grasped the tremendous importance of this subject, Mr. Taylor decided, on leaving the Bethlehem Steel Company, to devote the remainder of his life to expounding these principles, which he now saw would create a new era in the industrial world.

He believed that he could do this to best advantage if he should make no charge for his work; and, having acquired a competency, he gave his services during the last fourteen years free to anybody who was sincerely desirous of carrying out his methods.

He served as Vice-President of the Society in 1904 and 1905, and President in 1906, when he delivered, as his presidential address, his exhaustive monograph, "On the Art of Cutting Metals," a treatise of 250 pages.

His written contributions were many; one of the earliest to come before the Society was his "Notes on Belting," presented in 1893.

This paper contained the results of a long series of practical tests, and settled many contentions regarding the use and care of belts.

In 1895, he presented his paper on "A Piece-Rate System," in which he expounded the principles on which his system of management was subsequently based.

In 1903, he presented his signal paper on "Shop Management," and in 1911, "The Principles of Scientific Management."

He was joint author with Sanford E. Thompson of two works on concrete: "A Treatise on Concrete, Plain and Reinforced," and "Concrete Costs."

He died in Philadelphia, Pa., on March 21, 1915.

As a tribute to Dr. Taylor's memory, the business of the Annual Meeting of the Society, December 1915, was suspended for one hour and memorial exercises held. An account of these was published in The Journal, January 1916, and the memorial appears elsewhere in this volume.

JAMES P. TOLMAN

James P. Tolman was born in Boston, Mass., on November 7, 1847. He received his early education in the public schools of Boston and entered the Massachusetts Institute of Technology, receiving the degree of Mining Engineer in 1868, which was the first class to be graduated from the Institute.

In 1870, he became superintendent of the Silver Lake Cordage Company of Newtonville, Mass. In 1884 he organized J. P. Tolman & Company, manufacturers of braided cord. In 1888, the Samson Cordage Company in Shirley, Mass., was organized as successors of J. P. Tolman & Company and Mr. Tolman became president of the company, which position he held up to the time of his death. This company is one of the largest in the world manufacturing braided cord.

Mr. Tolman became a member of the Society in 1894. He died at his home in West Newton, Mass., on July 28, 1915.

HERBERT GRAY TORREY

Herbert Gray Torrey was born in New York City in 1839. He was educated in the College of the City of New York, from which he was graduated in 1860. He became the assistant of his father, John Torrey, who was the first chief assayer in the U. S. Assay Office in New York City, and he succeeded his father at the latter's death in 1873. He served as chief assayer until 1910, when he went into private practice, becoming president of H. G. Torrey & Company, assayers and metallurgists; he retired in 1912.

Mr. Torrey was also a consulting chemist, a specialist on alloys, a government expert in textile fabrics and an examiner of mines. He maintained a private metal shop at Stirling, N. J., manufacturing magnolia metal. He invented Torrey metal, an anti-friction alloy.

He was a member of the Society of the Cincinnati, the American Institute of Mining Engineers and The Franklin Institute. He became a life member of this Society in 1890. He died at his home at Stirling, N. J., on August 29, 1915.

WILBER H. TRAVER

Wilber H. Traver was born at Mattawan, Mich., on May 3, 1863. He received a common school education, and as the mechanical profession and especially railroad work appealed to him very strongly, he served an apprenticeship with the Michigan Central Railroad as a machinist from 1880 to 1883. In 1889, he became master mechanic for the Atchison, Topeka and Santa Fe Railroad and, following this, he held the same position with the Kansas City, Pittsburg and Gulf Railroad. In 1895, he took a position with the Rand Drill Company in the sales department and later became manager of the Chicago office of this company.

In 1906, Mr. Traver took a position with the Pneumatic Tool Company as manager of the mining department, where he remained until his death, which occurred at Houghton, Mich., on April 15, 1915.

He became a member of the Society in 1904.

CHARLES WARD

Charles Ward was born in Leamington, England, on March 5, 1841 and came to America in 1871. He had been trained as a gas engineer in England and almost his first work in America was in Charleston, W. Va., where he installed the first gas works. He later became superintendent and general manager of the gas company.

Mr. Ward came into prominence when the late Admiral Melville invited the makers of water tube boilers to compete for supplying the boilers of the coast defense vessel Monterey. Mr. Ward's boiler was tested in 1890 and proved so noteworthy for the length of the test under severe conditions and attained such excellent efficiency that four boilers of this type were installed on the Monterey. He thus has the credit of the first installation of water tube boilers on a large war vessel. Even before this, smaller water tube boilers of a different design manufactured by Mr. Ward had been used in steam launches of the United States Navy, and they have given such satisfaction that the Navy Department has continued their purchase and use.

Mr. Ward was also a pioneer in the use of screw propellers on western river steamboats in the effort to reduce waste and increase efficiency over the time-honored stern wheel boats, which are almost exclusively in use. Some years ago, he repeated the famous test made in England during the first half of the last century of a tug-of-war between a screw-propelled boat and a stern wheel boat of the same power. In the later case as in the earlier one, the screw propeller showed the better results.

Mr. Ward was elected to membership in the Society in 1892. He died in Charleston, W. Va., on January 17, 1915.

JOHN E. WARREN

John E. Warren was born in Grafton, Mass., on October 7, 1840. When he was still an infant his parents traveled West as pioneers and settled on a farm in Wisconsin. The short-term country schools of the day supplied all there were of his opportunities for schooling, except that he supplemented them afterward for a time by teaching, and then by entering Ripen Academy shortly before the Civil War. At the outbreak of the war, he promptly enlisted in the Union Artillery, and served with it through to the end.

He entered the employ of S. D. Warren & Company, makers of book paper, and it was to this company at its plant at Cumberland Mills, Me., that Mr. Warren devoted his business and professional service to the close of his life. He began as a mechanic, worked upward through various capacities, until in 1883 he was appointed agent of the mills, thus becoming the head and leader of its organization, the position which he retained for 32 years, up to his death.

Mr. Warren had been a member of the Society since 1886. He died on August 13, 1915.

WILLIAM WATSON

William Watson was born in Nantucket, Mass., on January 19, 1834, and died at his home in Boston, Mass., September 30, 1915. He graduated from Harvard in 1857, taking the Boyden prize for mathematics. The same year he became an instructor at Harvard in differential and integral calculus. Later he took a course of special study at the École Nationale des Ponts et Chaussées in Paris. On returning to this country he became university lecturer at Harvard. While in Europe, from 1860 to 1863, Professor Watson collected information on technical education which was made the basis of a plan of organization in 1864 of the Massachusetts Institute of Technology, where from 1865 until 1873 he was professor of mechanical engineering and descriptive geometry.

Professor Watson was United States Commissioner in 1873 to the Vienna Exposition, and he served as a member of the International Jury of the Paris Exposition in 1878. He had been honorary president of the Paris Congress of Architects and vice-president of the engineering section of the French Association for the Advancement of Science, serving several terms, and vice-president of the International Congress of Construction in 1889.

He was a member of the French National Academy at Cherbourg, Société des Ingénieurs Civils de France and the American Society of Civil Engineers. He was a fellow of the American Academy of Arts and Sciences, member of the American Association for the Advancement of Science, the Colonial Society of Massachusetts and the Mathematical Club. He was elected a member of this Society in 1886.

He was the author of many notable works on technical education and science, engineering, architecture and other subjects.

THOMAS DYSON WEST

Thomas Dyson West was born in Manchester, England, in 1851, and was the son of a niece of Dr. Michael Faraday. He was brought to America in childhood. At the age of twelve, he began the practical study of engineering at the Portland Locomotive Works, Portland, Me. In 1887, he organized the Thomas D. West Foundry Company, now known as the Valley Mold & Iron Company, of Sharpsville, Pa., and ten years later he founded the West Steel Casting Company. He was vice-president and general manager of the former until 1909, and was chairman of the board of directors of the latter until the time of his death.

Mr. West was the author of many books and papers on practical foundry work, publications which were basic in foundry literature and included "American Foundry Practice," "Moulders' Text Book," etc. He established and used American Foundrymen's Standardized Drillings, which was taken over by the U. S. Bureau of Standardization in 1905. He was also the pioneer of the Safety First movement, and he organized the American Anti-Accident Society.

He was a member of the American Society for Testing Materials and was president of the American Foundrymen's Association in 1905-1906. He was elected to membership in this Society in 1884, and he died in Cleveland, Ohio, on June 18, 1915, from injuries received in an accident.

STEPHEN BETTS WHITING

Stephen Betts Whiting was born at Reading-Ridge, Conn., on January 22, 1834. After attending the public schools of his native town and of New Haven, Conn., he finished his schooling with a year at the New Haven Collegiate and Commercial Institute. When fifteen years of age, he was apprenticed to the machinist trade and, before he was sixteen, he designed and constructed a miniature steam engine.

At seventeen, he designed and built an air-pump. At this time, also, he went to work as a full-fledged journeyman.

In 1855, he was sent to Urbana, Ohio, to take charge of the shops of the Urbana Machine Company, and in 1857 he went to Alton, Ill., as superintendent of the Illinois Iron Works. In 1860, he returned East and took charge of the Kaighn's Point Iron Works at Camden, N. J. At this time he entered into partnership with his old friend and schoolmate, Charles G. Wilcox, bought the Kaighn's Point Iron Works and operated it under the firm name of Wilcox & Whiting.

This firm built the U. S. Monitor Koka, which was one of the light-draft monitors designed by John Ericsson. Some of the principal dimensions of this vessel were: Overall length, 225 ft.; beam, 45 ft.; draft, 6½ ft.; diameter of turret, 20 ft.; height of turret, 9 ft. The motive power consisted of two 22 by 30-in. single-cylinder engines direct connected to two propeller shafts, each of which was fitted with a 9-ft. propeller. The armament consisted of one 11-in. gun and one 150-lb. rifle. The firm also built and erected the superstructure of the Chestnut Street Bridge over the Schuylkill River in Philadelphia.

Early in 1865 the Kaighn's Point Iron Works were sold and on July 1, 1865, Mr. Whiting went to Pottsville, Pa., as superintendent of the Colliery Iron Works. In this position his inventive faculty often came into useful play in connection with the design and manufacture of machinery for the coal regions. He designed the so-called Whiting system of rope driving, hauling and hoisting machinery which was first installed at the Lehigh & Wilkes-Barre planes at Solomon's Gap, Wilkes-Barre, Pa. At later dates this system was adopted for the Mahoney planes of the Philadelphia & Reading Coal & Iron Company, for the Brooklyn Bridge, for the Red Jacket or Whiting shaft of the Calumet & Hecla Mining Company, and for a number of the diamond mines of South Africa.

On September 1, 1878, he entered the employ of the Philadelphia & Reading Coal & Iron Company as mechanical engineer. On April 1, 1880, he was promoted to chief engineer, and on March 1, 1883, to general manager, continuing as such for five years.

On May 1, 1888, he entered the employ of the Calumet & Hecla Mining Company as general manager, which position he held until April 30, 1901, when he retired from active life. While with this company he recommended and supervised the sinking of the Red Jacket or Whiting shaft, a vertical, six-compartment shaft which intersects the lode at a depth of about 3300 ft. and reaches the level of the lode at the property line at a depth of 5000 ft. The cross-section of the shaft measures 15½ by 25 ft.

Mr. Whiting was a charter member of this Society and was Manager from April 1880 to November 1882 and Vice-President from 1882 to 1883. He was also a member of the American Society of Naval Engineers, the American Institute of Mining Engineers, the Institution of Mechanical Engineers (England) and the North of England Institute of Mining and Mechanical Engineers.

He died at his residence in Cambridge, Mass., on December 23, 1915.

BAXTER D. WHITNEY

Baxter D. Whitney was born in Winchendon, Mass., in 1817.

His early education was received in Winchendon, Hancock, N. H., and Fitchburg, Mass., but his business life started in his father's woolen mill when he was very young. When he was ten years old, he constructed a saw mill, run by water, which, while merely a boy's effort, was prophetic of the line which his activities later followed. When he was thirteen years old, he went to Worcester to help build some looms for his father.

At sixteen years of age, he constructed an 18-in. by 6-ft. engine lathe, designing and making the patterns and machining the castings. It had a large wormwheel feed and a V-bed which he milled with a special fixture comprising wooden beams laid in the floor.

In 1836, he began the construction of machinery for making tubs and pails, and had twenty-eight men working for him. The next year, he built sixteen looms for cashmere and later he built two or three steam jigs. In 1845, he built for himself a foundry and machine shop and the next year the first Whitney wood-planing machine, which took six weeks to construct, was built. This is still in existence and is claimed to be the first cylinder planer ever made which was a practical success. In 1857, he built a scraping machine and also a shaper and the famous Whitney gage lathe. During the Civil War, he was busy making gun-stock machinery. He received awards for his machines at exhibitions in Paris (1867), Vienna (1873) and Philadelphia, (1876).

Mr. Whitney became a member of the Society in 1886. He died at his home in Winchendon, Mass., on October 17, 1915.

HENRY WICK

Henry Wick was born in Youngstown, Ohio, on May 13, 1846. He received his early education in the schools of Youngstown and later spent two years at the preparatory school of Western Reserve College at Hudson, Ohio. He returned to Youngstown, and after a very short time spent in his father's bank, he became manager of the Packard Coal Company, then opening a mine about two miles west of Youngstown.

In 1877 he became treasurer and active manager of the Youngstown Rolling Mill Company. He later took the lead in purchasing the rolling mill at Warren on the failure of the pipe works there and became the head of the company which undertook its operation. Up to this time, Youngstown had had no steel plant but in the summer of 1892, a determined effort was made by the leading manufacturers to establish one there. As a result of this, the Ohio Steel Company was formed and he was chosen as its head. In 1899, the Ohio Steel Company with several plants in other places joined in forming the National Steel Company of which he became the first vice-president, with headquarters in Chicago, Ill. A year later the offices were moved to New York City and early in 1901, Mr. Wick was elected president. He resigned this position shortly afterwards, however, and returned to Youngstown to take up some enterprises in which he was particularly interested.

Mr. Wick became a member of this Society in 1893. He died at his home in Youngstown, Ohio, after a short illness on December 22, 1915.

ALFRED WILLIAMSON

Alfred Williamson was born in New York City on July 12, 1880. His early education was received in the public schools of New York City, after which he took a course in mechanical engineering at Columbia University, graduating in 1902. He then entered the service of the New York Central and Hudson River Railroad Company as a special apprentice in their West Albany shops. In 1904, he was employed in the turbine construction department of the General Electric Company in Schenectady, N. Y., but in 1905 he returned to Columbia and took up post-graduate work. In the latter part of that year, he was employed on the work of the Metropolitan Street Railway Company, and in the following year he entered the employ of the Department of Water Supply of New York as a mechanical engineer. He was in charge of the pumping stations of Manhattan and the Bronx from 1910 to the time of his death.

Mr. Williamson became a member of the Society in 1902. He died at his home in Bronxville, N. Y., on December 26, 1915.

HORACE WYMAN

Horace Wyman, whose death occurred on May 8, 1915, at his country home in Princeton, was born in Woburn, Mass., on November 27, 1827. He was educated in the academies at Woburn and Francestown, N. H., and began his business career in 1846, when he became a machinist in the works of the Amoskeag Manufacturing Company at Manchester, N. H. He was later employed by the

Lowell Machine Company and by the Hinckley Locomotive Works at Boston, and served as draftsman with the Holyoke Water Power Company.

About 1860 Mr. Wyman became associated with George Crompton of Worcester. He was made superintendent and manager of the Crompton Loom Works, holding that position until the consolidation of the business under its present name of Crompton & Knowles Loom Works, when he was made vice-president and consulting mechanical engineer of the company, retaining that position until his death.

All through his many years of service with the Crompton and the Crompton & Knowles Loom Works he applied himself to inventions of looms and textile machinery, until he had practically perfected looms as they are now used. His inventions made it possible for woolen, gingham and silk fabrics to be woven in more than one color and in larger pieces than before. Through the processes developed by him, rugs and carpets can now be manufactured in large sizes. Textile mills all over the country are using every day machines invented by Mr. Wyman, and he is regarded as having done more for the loom industry than any other man in Worcester and possibly in the country. His inventions were the largest single factor in the success of the Crompton & Knowles Loom Works.

He was a member of the Society since 1892 and was also a member of the Worcester County Mechanics' Association and the Worcester Society of Antiquity.

JOHN PHILIP ZIPF

John Philip Zipf, was born at West Point, Cal., on March 19, 1888. He attended the California School of Mechanical Arts and the University of California, from which he graduated in May 1912. After graduation, he worked with R. F. Chevalier, a consulting engineer, in testing boilers for numerous gas and power plants in and about San Francisco. In December, 1912, he accepted a position with the Ramie Fibre Company of San Francisco as draftsman, and after the failure of this company, he accepted a position with the Sutter Basin Company of Sacramento as draftsman of plans for electric pumping machinery for their reclamation project. Having completed their work in May 1914, he entered the office of the California State Engineer at Sacramento, Cal.

Mr. Zipf died at his home in San Francisco, Cal., on May 24, 1915. He was elected to membership in the Society in 1913.

ALFONSO H. CARPENTER

Alfonso H. Carpenter was born at Ludlow, Vt., on October 22, 1850. He received his education in the ordinary home schools and at the age of 14 went to work for Helon M. Carpenter of Grafton, Vt., as a blacksmith, remaining there until he was twenty. The rest of his life was spent in the machinery manufacturing and foundry business as commercial traveler and manager. At the time of his death he was vice-president of the Acme Machinery Company of Cleveland, Ohio, which position he had held for many years.

Mr. Carpenter was elected to membership in the Society in 1895. He died December 24, 1915.

ACKNOWLEDGMENT

Attention has been called to the unfortunate omission from the report of the Committee on Standardization of Flanges as published in Volume 36 of the Transactions of the Society of the name of the National Association of Master Steam and Hot Water Fitters. Acknowledgment is hereby made by the Committee on Standardization of Flanges to the Association for its generous support and work in the preparation of this report, and the name of the Association will appear hereafter on any publication of the report by the Society.

It will be found that the dimensions published by the Master Steam and Hot Water Fitters Association under the title of 1915 U. S. Standard are identical with the dimensions in the American Standard published by the Society.

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No. 1529

INDEX TO VOLUME 37

NOTE

1 Names of authors and discussors, also deceased members preceding an obituary notice, are in caps and small caps. A discussor is distinguished from the author of a paper by (*D*), placed after the name of the paper.

2 Titles of papers, where placed after the name of the author, and appearing in their exact form, are in italic. Papers are indexed not under their title but under their subject matter.

3 The Society is not responsible as a body for the statement of facts or opinions in the papers and discussions.