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THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 38

NEW ORLEANS MEETING
NEW YORK MEETING
1916

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CONTENTS OF VOLUME XXXVIII[1]

NEW ORLEANS AND NEW YORK MEETINGS

[1] The Society is not responsible as a body for the statement of facts or opinions in the papers and discussions.

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Alaska	1	Nevada	2
Arizona	16	New Hampshire	18
Arkansas	7	New Jersey	428
California	205	New Mexico	8
Canal Zone	10	New York	1953
Colorado	40	North Carolina	21
Connecticut	397	North Dakota	2
Delaware	45	Ohio	529
District of Columbia	59	Oklahoma	21
Florida	13	Oregon	12
Georgia	40	Pennsylvania	913
Hawaii	12	Philippine Islands	12
Idaho	4	Porto Rico	9
Illinois	527	Rhode Island	93
Indiana	110	South Carolina	8
Iowa	24	South Dakota	4
Kansas	27	Tennessee	38
Kentucky	17	Texas	59
Louisiana	36	Utah	22
Maine	31	Vermont	20
Maryland	88	Virginia	51
Massachusetts	658	Washington	42
Michigan	265	West Virginia	17
Minnesota	86	Wisconsin	141
Mississippi	9	Wyoming	2
Missouri	141		
Montana	10	Total	7347

Foreign Countries

Africa	13	India	6
Australia	8	Italy	2
Austria	2	Japan	9
British West Indies	1	Mexico	4
Canada	132	Norway	3
Central America	1	Roumania	1
Channel Islands	1	Russia	8
China	5	Scotland	4
Cuba	22	South America	26
Denmark	3	Spain	1
Dutch East India	2	Sweden	4
England	60	Switzerland	3
Finland	2	Turkey	1
France	10	West Indies	1
Germany	19		
Holland	1	Total	355

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TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 38—1916

VOLUME 38 of Transactions contains the permanent record of the Society's activities for the year 1916. As in the previous volumes, the space is devoted mainly to the papers and discussion given at the Spring and Annual Meetings. These have been secured both through the efforts of the Committee on Meetings and its several sub-committees, and through the local Sections of the Society. In consequence, the papers cover a broad field and are representative of mechanical engineering development in widely separated parts of the country.

In the Council Report which follows on page 4 will be found a brief review of the affairs of the Society for the period covered. The reports of the Standing Committees outlined in the report of the Council were given in full in THE JOURNAL for December, 1916. A list of the meetings of the local Sections is included and it is noteworthy that over 100 were held, emphasizing the importance which this development is assuming and the increasing service which the Society is thus able to render to the membership. Accounts of these local meetings have appeared in THE JOURNAL throughout the year.

DAVID SCHENCK JACOBUS

David Schenck Jacobus, distinguished as one of the foremost American mechanical engineers and educators, was born in Ridgefield, N. J., in 1862. He was educated in a private school, then in Stevens High School, Hoboken, N. J., where he won by competitive examination a free scholarship in Stevens Institute of Technology. He was graduated from the Institute with the degree of Mechanical Engineer in 1884.

Upon his graduation he was appointed instructor, then assistant professor of experimental mechanics, being closely associated with Prof. J. E. Denton. In 1897 he was called to the chair of experimental mechanics and engineering physics, in which he continued until 1906. From 1900 to 1906 he was also in charge of the Carnegie Laboratory of Engineering, built and equipped with funds donated by Andrew Carnegie for carrying out a course of instruction to supplement class-room work by practical experiments made by the students.

At Stevens Institute Dr. Jacobus devised original apparatus for illustrating physical laws and for testing various mechanical devices, and developed a course of experimental mechanics especially fitted for a mechanical-engineering school.

While at the Institute Dr. Jacobus performed a large amount of expert work, involving investigations and reports upon mechanical devices and processes for the production of power, and efficiency tests on steam motors, turbines and other power-plant apparatus. He was early in the field of refrigeration, making tests of machines and writing articles on the subject. He also made early experiments on acetylene-gas generators and on fire-sprinkler systems.

Since 1906 he has been actively associated with the Babcock and Wilcox Company at the head of its engineering department in the position of advisory engineer. This made it necessary for him to give up practically all his work at Stevens Institute, although it was arranged that he should continue on the faculty under the title of special lecturer in experimental engineering and should deliver lectures on steam-engineering subjects. At the same time, as an evidence of esteem, the Institute conferred upon him the honorary degree of Doctor of Engineering and he was made a trustee.

Dr. Jacobus is an authority on steam-engineering, and he has written numerous scientific papers on steam-engineering subjects, many of which have been included in the transactions of the engineering societies and many published in engineering periodicals.

He was elected to membership in The American Society of Mechanical Engineers in 1889; he was a Manager of the Society from 1900 to 1903, and was Vice-President from 1903 to 1905. He has served on a number of committees of the Society, among which was the committee which framed the present constitution. He was a member of the committee appointed to standardize a system for testing steam engines which presented its final report in December, 1902. In 1903 he was appointed chairman of a committee to suggest a standard tonnage basis for refrigeration; this committee reported in 1904 and was reappointed. He has served on the Committee on Power Tests, which originally organized by electing him as chairman; owing to pressure of business duties he resigned the chairmanship, but he has nevertheless taken an active part in the preparation of the report. In 1914 he was appointed a member of an advisory committee which was formed to assist the Boiler Code Committee, and on the combining of the Boiler Code Committee and advisory committee into a single Boiler Code Committee he was made one of the members. He was elected President of the Society in December, 1915.

He is a member of the American Institute of Mining Engineers, the American Institute of Electrical Engineers, and the Society of Naval Architects and Marine Engineers, and a fellow of the American Association for the Advancement of Science. He was President of the American Society of Refrigerating Engineers in 1906–1907. He is a member of the American Mathematical Society, the Society for the Promotion of Engineering Education, the American Society for Testing Materials, and The Franklin Institute of the State of Pennsylvania.

ANNUAL REPORT OF THE COUNCIL

COVERING THE PERIOD FROM DECEMBER 7, 1915, THE BEGINNING OF THE THIRTY-SIXTH ANNUAL MEETING, TILL DECEMBER 5, 1916, THE BEGINNING OF THE THIRTY-SEVENTH ANNUAL MEETING

The Council has held eight monthly meetings and two special meetings, and the Executive Committee of the Council has met five times between meetings of the Council to act upon matters requiring immediate action. In the main the running account of the activities of the Society is reported under the heads of the respective committees. The reports of Standing Committees are published in THE JOURNAL, December, 1916.

The Council registers with deep regret the deaths of E. D. Leavitt and John E. Sweet, both Honorary Members and Past-Presidents of the Society. Special commemorative resolutions were passed by the Council.

Encouraged by Mr. Holley, Professor Thurston and Mr. Bailey, Professor Sweet corresponded with and visited several leading engineers with reference to the formation of a mechanical engineers' society. The response led Professor Sweet to plan the organization, and he called the first meeting on January 18, 1880. A memorial service to Professor Sweet will be held at the Annual Meeting, December, 1916.

Dr. Ambrose Swasey was this year elected an Honorary Member. His letter of acceptance is here recorded:

Cleveland, Ohio,
July 11, 1916.

PROFESSOR D. S. JACOBUS, President,
The American Society of Mechanical Engineers, New York.

Dear Professor Jacobus:

I am pleased to receive your favor of the 5th advising me of the action of the Council in electing me an honorary member of the Society. This is a great honor, and I appreciate it more than I have words to express. I regard it one of the gratifying experiences of my life to have been a member of the Society from the time of its organization, but to be placed on the honor roll is indeed a mark of great distinction.

Very truly yours,

(*Signed*) AMBROSE SWASEY.

While this year has been one of increased internal activities in the affairs of the Society proportionate to the growth in membership and increased inter-society coöperation in matters affecting the Engineering Profession as such, the most significant development of the year has been the Society's participation in movements for national service. In common with the other national engineering societies, our Society has been called upon by the Government to perform work for the public welfare, and, in turn, we as engineers have received recognition in legislation directly and permanently establishing our profession as an arm of the Government.

Before reviewing the conspicuous and important phases of the activities during 1916 of the membership, committees, sections and student branches, and enumerating the actions taken in joint society matters, this report first refers to the relations the Society has maintained in its service to the nation.

INDUSTRIAL CENSUS

Last year national recognition was accorded the Engineering Profession in the invitation extended to the engineering societies by the Secretary of the Navy of the United States to nominate the members of a Naval Consulting Board. This Society's representatives on the Board are Spencer Miller, Member of the Council, and W. L. R. Emmet.

In January, 1916, the engineers received a second recognition in the following letter from the President of the United States:

THE WHITE HOUSE
WASHINGTON

January 13, 1916.

DR. D. S. JACOBUS,
President of The American Society of Mechanical Engineers.

My dear Sir:

The work which The American Society of Mechanical Engineers has done through its members on the Naval Consulting Board is a patriotic service which is deeply appreciated. It has been so valuable that I am tempted to ask that you will request your Society to enlarge its usefulness to the Government still further by nominating for the approval of the Secretary of the Navy a representative from its membership for each state of the Union to act in conjunction with representatives from the American Society of Civil Engineers, the American Institute of Electrical Engineers, the American Chemical Society, and the American Institute of Mining Engineers, for the purpose of assisting the Naval Consulting Board in the work of collecting data for use in organizing

manufacturing resources of the country for the public service in case of emergency. I am sure that I may count upon your cordial coöperation. With sincere regard,

Cordially yours,

WOODROW WILSON.

In transmitting this letter to the membership of the Society, President Jacobus wrote:

> **The task to which we are committed is a vital one. We are to harness the resources of this country for a truly national service. We are to bring home to hundreds of thousands of our American people — business men and workmen alike — a better conception of the obligation and of the privileges of American citizenship. The burden laid upon each of us as an individual is not a great one. A few dollars of expense, a few hours or even a few days of time, are slight tribute to a cause which cannot but result in strengthening the reputation of American engineers and in furthering our national preparedness. The value of our completed work will be incalculable. Through the performance of this task, of which this inventory is the first essential step, we shall have served such notice of our industrial preparedness to all the nations of the world that we shall have written for this country, to which we all owe so much, the greatest possible insurance for peace. Although the Naval Consulting Board represents nominally a naval activity, the War Department is thoroughly in touch with this work which is of even more importance to that department than to the Navy. No other such national opportunity for patriotic service has been accorded the technical men of this country.**

To conduct the work of compiling this inventory, a Committee on Industrial Preparedness, a sub-committee of the Naval Consulting Board, was constituted, with Howard E. Coffin, Mem.Am.Soc. M.E., as Chairman.

To assist this committee in its work, State Directors were appointed, one in each State for each of the national societies mentioned in President Wilson's letter, and they in turn have been generously assisted by "field aides" composed of other members of the societies who in many cases personally visited the different plants in order to secure the required information. The names of the State Directors representing our Society follow.

ALABAMA. F. H. Crockard, Birmingham. — V. P., Tenn. Coal, Iron & R. R. Co.

ALASKA. George A. Diamond, Nome. — Mgr., Scheid & Co.

ARIZONA. A. G. McGregor, Warren. — Calumet & Ariz. Mining Co.

ARKANSAS. B. N. Wilson, Fayetteville. — Prof. Mech. Eng., Univ. of Ark.

CALIFORNIA. G. W. Dickie, San Francisco.

COLORADO. T. B. Stearns, Denver.

CONNECTICUT. H. B. Sargent, New Haven. — V. P., Sargent & Co.

DELAWARE. R. W. Smith, Wilmington. — V. P., Hilles & Jones Co.

DISTRICT OF COLUMBIA. Dr. W. S. Stratton, Washington. — Director, Bureau of Standards.

FLORIDA. R. E. Chandler, Gainesville. — Prof. Mech. Eng., Univ. of Fla.

GEORGIA. Oscar Elsas, Atlanta. — Pres., Fulton Bag & Cotton Mills.

IDAHO. Geo. F. Waddell, Squirrell, Fremont County.

ILLINOIS. Dr. W. F. M. Goss, Urbana. — Dean, Coll. Eng., Univ. of Ill.

INDIANA. Geo. O. Rockwood, Indianapolis. — Pres., Rockwood Mfg. Co.

IOWA. S. M. Woodward, Iowa City. — Prof. Mech., State Univ. of Iowa.

KANSAS. A. A. Potter, Manhattan. — Dean, Div. of Engineering, Kansas State Agric. College.

KENTUCKY. W. S. Speed, Louisville. — Pres., Louisville Cement Co.

LOUISIANA. A. M. Lockett, New Orleans. — Pres., A. M. Lockett Co., Ltd.

MAINE. P. M. Hammett.

MARYLAND. C. C. Thomas, Baltimore. — Prof. M. E., Johns Hopkins Univ.

MASSACHUSETTS. Ira N. Hollis, Worcester. — Pres., Worcester Poly. Inst.

MICHIGAN. Alex Dow, Detroit. — Pres. & G. M., The Detroit Edison Co.

MINNESOTA. J. J. Flather, Minneapolis. — Prof. Mech. Eng., Univ. Minn.

MISSISSIPPI. R. C. Carpenter, Agricultural College. — Prof. Mech. Eng.

MISSOURI. E. Flad. St. Louis.

MONTANA. C. V. Nordberg, Butte. — Nordberg Mfg. Co.

NEBRASKA. Wm. R. McKeen, Omaha. — Pres., McKeen Motor Car Co.

NEVADA. James G. Scrugham, Reno. — Dean Eng. College, Univ. of Nev.

NEW HAMPSHIRE. Thomas W. Fry, Claremont. — Secy., Sullivan Mchy. Co.

NEW JERSEY. H. L. Gantt, Montclair.

NEW MEXICO. L. J. Charles, Elephant Butte. — C. E., U. S. Reclamation Service.

NEW YORK. W. H. Marshall, New York City. — Pres., American Locomotive Co.

North Carolina. Wm. S. Lee, Charlotte. — V. P. & Ch. Engr., Southern Power Co.

North Dakota. Calvin H. Crouch, University. — Dean Coll. Mech. & Elec. Eng., Univ. of N. D.

Ohio. Frank A. Scott, Cleveland. — V. P., The Warner & Swasey Co.

Oklahoma. J. P. Fisher, Bartlesville, Okla.

Oregon. Bert C. Ball Portland. — Pres. & Mgr., Willamette Iron & Steel Works.

Pennsylvania. Julian Kennedy, Pittsburgh.

Rhode Island. Henry D. Sharpe, Providence. — Treas., Brown & Sharpe Mfg. Co.

South Carolina. J. L. Coker, Jr., Hartsville. — V. P., Carolina Fiber Co.

South Dakota. M. W. Davidson, Vermillion. — Prof. M. E., Univ. of S. Dak.

Tennessee. N. Sanders, Chattanooga. — Newell Sanders Plow Co.

Texas. W. B. Tuttle, San Antonio. — V. P., San Antonio Traction Co. and San Antonio Gas & Elec. Co.

Utah. Wm. Wraith, Salt Lake City. — G. M., Internat. Smelting Co.

Vermont. James Hartness, Springfield. — Pres., Jones & Lamson Mach. Co.

Virginia. W. D. Mount, Saltville. — G. M., Mathieson Alkali Works.

Washington. James V. Paterson, Seattle. — Pres., Seattle Constr. & Dry Dock Co.

West Virginia. Chas. E. Ward, Charleston. — Pres., Charles Ward Eng. Works.

Wisconsin. L. E. Strothman, Milwaukee. — Dept. Mgr., Allis-Chalmers Mfg. Co.

Wyoming. E. G. Hoefer, Laramie. — Head Dept. M. & E. Engrg., Univ. of Wyoming.

The Council records its appreciation of the work of the Society's representatives in this splendid movement, which has now been carried through with complete success.

ENGINEER RESERVE CORPS AND NATIONAL COUNCIL OF DEFENSE

In response to an invitation from General Wood, Commander of the Department of the East, a joint committee was chosen by five national engineering societies to facilitate the carrying out of the organization of a Civilian Engineer Reserve Corps as a part of the military forces of the United States. This committee organized with a joint executive committee of: Wm. Barclay Parsons, representing the American Society of Civil Engineers, *Chairman*, H. S. Drinker, representing the American Institute of Mining Engineers, Wm. H. Wiley, representing The American Society of Mechanical Engineers, B. J. Arnold, representing the American Institute of Electrical Engineers, Ralph D. Mershon, representing the American Institute of Consulting Engineers. The committee made its first report in December 1915, and in July 1916 Chief of Engineers, Lieut-Col. E. Eveleth Winslow of the Army Engineer Corps, in writing to all the engineer officers throughout the country, said, "Congress has now provided a means by which the civilian engineers can more than fully prepare themselves for that highest duty of citizens — the defense of our country." The bill became effective on July 1, 1916, thus directly and permanently establishing the Engineering Profession as an arm of the Government. Public recognition of the engineer is now an accomplished fact.

As a sequel to the work of the Naval Consulting Board, a bill has been passed by Congress providing for a National Council of Defense, consisting of five Cabinet officers and not more than seven civilians.[1]

Standing, Annual and Special Committees

STANDING COMMITTEES

The President made early appointments on the Standing Committees, on which one member's term expires each year and a new member is appointed for five years. The appointments were: *Finance Committee*, George M. Forrest; *Meetings Committee*, D. S. Kimball; *Publication Committee*, Charles I. Earll (reappointment); *Membership Committee*, Fred J. Miller; *Library Committee*, Walter M. McFarland (reappointment); *House Committee*, Maxwell M. Upson; *Research Committee*, C. C. Thomas; *Public Relations Committee* (not appointed); *Standardization Committee*, Henry Hess (1), John H. Barr (2), Chas. Day (3), Carl Schwartz (4) and Wm. F. Kiesel, Jr. (5). Messrs. Barr and Day later resigned on account of urgent other work and appointments were made of H. L. Gantt (2), H. G. Stott (3); *Con-*

[1] The President later made the following appointments: Medicine, Franklin H. Martin; Labor, Samuel Gompers; Transportation, Daniel Willard; Science and Research, Hollis Godfrey; Raw Materials, Bernard Baruch; Munitions, Howard E. Coffin; Supplies, Julius Rosenwald.

stitution and By-Laws, Jesse M. Smith, Geo. M. Basford, I. N. Hollis, F. R. Hutton, J. E. Sague.

Statements covering the work and plans of the Standing Committees follow, and the annual reports of these committees are included as an appendix to this Council Report.

Finance Committee. Last year, when a serious business depression in this country seemed imminent, precautionary measures were taken to conserve the interests of the Society in every proper way. This conservation was carried out a little more than later proved necessary, although not more than seemed wise at the beginning of the year, with the result that we saved $24,000 to go into the surplus. This is made available to increase our working capital, a necessity by virtue of the Society's rapid growth.

All the gifts of money to the Society are intact and invested, and initiation fees have been used during the last few years to retire the certificates of indebtedness issued to pay off our share in the land of the Engineering Societies Building. Our sister societies were each able to pay off, by subscription, their individual share — $180,000 — of the purchase price of this land, but with our subscriptions amounting to only about two-thirds of the gifts received by these societies, we have been able to finance the problem on another basis; yet we were the first to pay off the mortgage on the land, by means of certificates issued to the membership. Our certificates were retired on July 1, 1916.

In the future instead of putting the whole sum from initiation fees into reserve, it is proposed to put one-half into reserve and one-half into the general income of the Society.

Meetings Committee. The culmination of the work of this committee is represented in the Thirty-sixth Annual Meeting held in New York in December, 1915, and the spring meeting held in New Orleans in April, 1916. Too much praise cannot be accorded to the members of the committee for the great amount of time and work they have given to reviewing manuscripts of the papers presented and arranging the programs of these meetings.

The committee has received valuable assistance in its work from its sub-committees on Air Machinery, Cement Manufacture, Fire Protection, Gas Power, Hoisting and Conveying, Industrial Buildings, Machine Shop Practice, Protection of Industrial Workers, Railroads, and Textiles. An account of the work of these sub-committees is given later.

THE ANNUAL MEETING

The Thirty-sixth Annual Meeting was held in New York City, December 7–10, 1915, inclusive. There was a record attendance of 1437, of which 819 were members. Members were present from every section of the country, and fourteen local Sections of the Society were officially represented by delegates, two of whom were from California. Enthusiasm was manifested in various ways; there was a series of conferences of representatives of the Sections, at which "Extend the Local Sections" was distinctly emphasized; at the Council meetings the sentiment expressed was equally enthusiastic; the spirit of democratization was dominant, and the membership was encouraged to participate in the affairs of the Society to a greater extent.

The papers presented were representative of the mechanical engineering work of the country as a whole, having been selected through intimate acquaintance with the subjects to be discussed and on the mature judgment of the expert engineers who constitute the different sub-committees.

At this particular meeting, seven papers were contributed by Sections and thirteen, including reports, by sub-committees.

In the matter of entertainment for the visiting members, an innovation was introduced in the way of a smoker held in the rooms of the Society. The rooms were filled to their utmost capacity, indicating the popularity of such a form of entertainment and the appreciation of the opportunity for members to "get together."

On Thursday evening there was a dinner and dance at the Hotel Astor, which was attended by 300 members and guests. On the third and fourth days of the meeting, excursions were made to various points of engineering interest.

The entertainment features for the ladies were the reception and tea and a special luncheon tendered by a member of one of the ladies' committees.

The committee was ably assisted at the meeting by the House Committee, which conducted the President's Reception, and by the New York Local Committee, which had charge of the other social events.

College Reunions. Friday night of the Annual Meeting has come to be regarded as college alumni reunion night. At the meeting of 1915 ten such reunions were held and the members of the Society were invited to all.

at the Annual Meeting in December, 1916: Standardization of Machine Tools, by Carl G. Barth, and A Proposed Plan of Activities for the Machine Shop Practice Committee. Around these papers the committee hopes to build a comprehensive discussion on the standardization of machine tools.

Protection of Industrial Workers. The Sub-committee on Protection of Industrial Workers conducted one session at the annual meeting of the Society in December, 1915. The papers presented were: Standardization of Safety Principles, by Carl M. Hansen; Modern Movement for Safety from Standpoint of Manufacturer, by Melville W. Mix; The Attitude of the Employer towards Accident Prevention and Workmen's Compensation, by W. H. Cameron; and Industrial Safety and Principles of Management, by W. P. Barba.

The committee has held two meetings during the year 1916.

The result of the request to the Bureau of Standards, the Conference Board on Safety and Sanitation, and the National Safety Council, was the appointment of a representative from each of these organizations.

The committee has conducted a great deal of correspondence with a view to formulating for consideration sets of machinery-safeguarding codes. One code on Cranes has been secured and will be presented at the Annual Meeting, December, 1916; this code was prepared under the supervision of Dr. John Price Jackson, member of the committee.

Other Codes are now in preparation, the plan of the committee being to present a set of Codes as formulated, and the purpose is to so draft them that they will be recognized by the various states and so contribute to uniformity in legislation and state regulations.

This committee is constituted as follows: John H. Barr, *Chairman;* Melville W. Mix, John P. Jackson, John W. Upp, William A. Viall, M. W. Alexander, C. M. Hansen, G. R. Olshausen.

Railroads. Edwin B. Katte, *Chairman;* Geo. M. Basford, Frank H. Clark, C. E. Eveleth, W. F. M. Goss, A. L. Humphrey, Wm. F. Kiesel, Jr., Geo. W. Rink, Norman W. Storer, H. H. Vaughan.

The Sub-committee on Railroads has confined its activities this year to the preparation of a program for the Railway Session to be held at the Annual Meeting, December, 1916. The committee has also under consideration arrangements for a special railroad meeting to be held in February or March, 1917.

Publication Committee. The publications of the year have been

The Journal, which has been issued monthly, the annual volume of Transactions, the Year Book, the sixth annual volume of Condensed Catalogues, and the Power Test Code (reprinted from Vol. 36 of Transactions).

Features in The Journal have been papers and discussions, in whole or in comprehensive abstract, presented at general meetings; papers and discussions presented at meetings of the sections, correspondence departments which were started last year for short original articles and for discussions, and which have developed very successfully; a Society affairs section which has now also become a pretentious department, and an engineering survey section, reviewing the world's technical press. Plans are under way for the further development of The Journal to include such features as will, it is hoped, make this publication a professional necessity to every mechanical engineer.

Volume 36 of Transactions, issued during the summer, was the largest ever published by the Society and contained nearly 1600 pages. The volume contained papers and discussions presented at the spring meeting in Buffalo, June, 1915, the special meeting in San Francisco in connection with the International Engineering Congress, 1915, and the Annual Meeting in New York, December, 1915. It also included the Power Test Code, an account of the memorial service to Frederick W. Taylor, Past-President of the Society, held at the Annual Meeting in 1915, and other features.

Membership Committee. In an appendix to this report is given a detailed summary of the applications considered by the Membership Committee. There were 1879 candidates for membership, 1143 of whom were recommended for acceptance.

The committee has also considered applications from former members desiring reinstatement; these applications are received in the same way as an original application, decisions and recommendations of the committee being presented to the Council. Sixteen reinstatements have been made.

The Committee also passes on cases in which a member having been connected with the Society for 30 years, or having reached the age of seventy and paid dues for 25 years or over, finds it necessary to resign. Such men the Council desires to retain in the membership, and provision has been made to remit dues and keep these names on the active rolls of the Society.

The accompanying table shows the changes in membership during the fiscal year 1915–1916:

MEMBERSHIP, 1915-1916[1]

Grade	Oct. 1, 1915	Losses					Additions				Net Gain	Oct. 1, 1916
		Trans.	Resign.	Lapsed	Deaths	Total	Trans.	Elec.	Reinst.	Total		
Honorary members	14	....	....	.....	2	2	1	......	....	1	1[2]	13
Members	4334	1	35	50	59	145	43	341	12	396	251	4585
Associates	388	2	7	13	3	25	7	38	....	45	20	408
Associate members	448	4	3	.....	2	9	35	211	....	246	237	685
Juniors	1393	79	30	41	7	157	....	316	1	317	160	1553
Total	6577	86	75	104	73	338	86	906	13	1005	667	7244
Affiliate students	919										Decrease 78	841

[1] The data here given cover the Society's fiscal year, October 1, 1915 to September 30, 1916. The data in the forthcoming Year Book will cover the calendar year of 1916.

[2] Loss.

Library Committee. There has been little required of the Library Committee of the Society as such, the work of the administration of the library of the United Engineering Society, of which our library is a part, being conducted under the direction of the Library Board. On this Board our Library Committee forms the representation of the mechanical engineers.

The principal matter of interest to the Board this year has been the consideration of the plans for receiving the library of the American Society of Civil Engineers. With this library added to that of the original Founder Societies, the largest and most complete engineering library in the world will be at the service of every engineer in whatever branch of the profession.

House Committee. The House Committee has considered the alterations required in the rooms of the Society to take care of the supporting columns passing up through the rooms to carry the load of the three stories being added to the building for additional library accommodations and headquarters for the American Society of Civil Engineers, and felt this is the opportune time to make some desirable changes in the decoration of the Council and reception rooms, the Secretary's office and the reception hall. They have also advantageously placed the insurance of the property of the Society after exhaustive investigation; have most painstakingly looked after all other interests of the Society's rooms and property, and arranged the details of the President's reception at the annual meetings. A more detailed report is given in the appendix to this record.

Research Committee. The work of the Research Committee has consisted in lining up into definite duties and plans for sub-committee work. The committee has now several sub-committees including Bearing Metals, C. H. Bierbaum, *Chairman*, H. Diedrichs and John A. Capp; Clinkering of Coal, Lionel S. Marks, *Chairman*, F. C. Hubley, A. V. Bleininger, O. W. Palmenberg, S. W. Parr; Flow Meters, R. J. S. Pigott, *Chairman*, Leo Loeb, F. G. Hechler, G. S. Coffin, Chas. C. Lee, Sidney Fisher; Fuel Oils, Raymond H. Danforth, *Chairman*, Lee E. Barrows, Andrew M. Hunt; Lubrication, Albert Kingsbury, *Chairman*, A. E. Flowers, Mayo D. Hersey, George B. Upton; Cutting Action of Machine Tools, Leon P. Alford, *Chairman*, A. L. DeLeeuw, E. E. Barney; Materials of Electrical Engineering, R. D. Mershon; Safety Valves, Philip G. Darling, Henry D. Gordon, Frederick L. Pryor, Frederick M. Whyte; Worm Gearing, Fred A. Halsey, *Chairman*, L. D. Burlingame, Wilfred Lewis, Walter Rautenstrauch.

The Sub-committees on Materials of Electrical Engineering, Safety Valves, and Worm Gearing are not active at the present time, but their work will be resumed probably within the next year.

Sub-Committees of the Research Committee. In the general report of the Research Committee is presented a synopsis of the plans and work accomplished by the sub-committees of the Research Committee.

Public Relations Committee. The Council has under careful consideration the entire field of the Society's activities in their relation to the public, so that no active work has been taken by the Committee this year. C 55 of the Constitution of the Society has been interpreted as follows: That the Society may not take part in partisan politics, but should coöperate in every way with the public in the promotion of the arts and sciences connected with engineering. This matter is being carefully considered both in connection with the activities of the Society as a whole and with those of its Sections.

Constitution and By-Laws. The Committee on Constitution and By-Laws acts in an advisory relation to the Council in consideration of suggested and necessary amendments to the Constitution, By-Laws and Rules. Where advisable to amend in the opinion of the Council, this Committee is asked to prepare and present a suggested draft or revision, in order that harmony may be preserved in relating to existing laws.

Standardization Committee. This committee became a standing committee of the Society by amendment to the Constitution in the spring of 1915. The action was the result of a report prepared by Mr. Henry Hess, and approved by the Council in December, 1913. The purport of the report is that the Committee shall standardize the method of making and arriving at standards rather than create standards themselves. The committee shall work along the lines of unifying the activities of the Society, proposing national and international coöperation between organizations, proposing governmental coöperation with organizations, and establishing an exchange of standardization knowledge.

The committee will coöperate by representation on a proposed Joint Committee composed of three representatives each from the national engineering societies, to consider and report back to their respective societies suggested means of bring about coöperation in American engineering standards.

NOMINATING COMMITTEE

The Nominating Committee this year was appointed by the President in a way to secure the greatest possible democracy in selecting nominees for executive offices. The Sections of the Society were divided into five geographical groups and one member of the committee was chosen from each group.

The committee consisted of Walter B. Snow, *Chairman*, representing the Boston, New Haven and Worcester Sections; H. M. Montgomery, representing the Chicago, Milwaukee and Minnesota Sections; E. H. Ohle, representing the Atlanta, Birmingham, St. Louis and Cincinnati Sections; J. T. Whittlesey, representing the San Francisco and Los Angeles Sections, and D. Robert Yarnall, representing the New York, Philadelphia and Buffalo Sections.

The report of the committee Chairman follows:

To the Secretary:

Dear Sir: Acting for the Nominating Committee, which you advised upon February 24, 1916, of their appointment by the President, I have the honor to submit the following report.

As the result of several meetings at New Orleans in April, and subsequent personal interviews on the part of individual members, the following names of nominees for the various elective offices next falling vacant under the Constitution, together with the written consent of each nominee, are herewith presented:

For President, for one year:

Ira N. Hollis, Worcester, Mass.

For Vice-Presidents, for two years:

Charles H. Benjamin, Lafayette, Ind.
Arthur M. Greene, Jr., Troy, N. Y.
Charles T. Plunkett, Adams, Mass.

For Managers, for three years:

Robert H. Fernald, Philadelphia, Pa.
William B. Gregory, New Orleans, La.
C. R. Weymouth, Berkeley, Cal.

For Treasurer:

William H. Wiley, New York, N. Y.

Respectfully submitted,

For the Committee,
(*Signed*) Walter B. Snow, *Chairman.*

The President appointed as Tellers of Elections for 1916: Robert H. Kirk, *Chairman;* Erwin S. Cooley, Harry A. Hey.

SPECIAL COMMITTEES

Committee on A.S.M.E. Junior Prizes. Seven papers were submitted for competition for the 1916 prize for the best paper by a Junior Member, and after careful consideration, the committee awarded the prize to:

L. B. McMillan for his paper entitled The Heat Insulating Properties of Commercial Steam-Pipe Coverings.

Honorable mention was awarded to:

Victor J. Azbe for his paper on Power Plant Efficiency, Herbert B. Reynolds for his paper on the Flow of Air and Steam through Orifices.

Committee on A.S.M.E. Student Prizes. Prizes have this year been awarded to:

Boynton M. Green, Leland Stanford University, for his paper on Bearing Lubrication.

Howard E. Stevens, Rensselaer Polytechnic Institute, for his paper on An Investigation of the Dynamic Pressure on Submerged Flat Plates.

M. Adam, Louisiana State University, for his paper on The Adaptability of the Internal Combustion Engine to Sugar Factories and Estates.

Honorable Mention has been awarded to:

M. Boyd Gordon, University of Cincinnati, for his paper on A New Type of Unaflow Engine.

S. C. Williams, Stevens Institute of Technology, for his paper on Photostatic Reproduction.

Charles P. Miller, Pennsylvania State College, for his paper on Investigation of Properties of Low and Medium Carbon Steels.

Biographies Committee. A Biographies Committee is to be appointed to supervise the preparation and editing of biographies of noted engineers.

Boiler Code Committee. By reappointment by the Council, the Boiler Code Committee is constituted as follows: John A. Stevens, *Chairman,* C. W. Obert, *Secretary,* Wm. H. Boehm, Rolla C. Carpenter, Frank H. Clark, Francis W. Dean, Thomas E. Durban, Carl Ferrari, Elbert Curtiss Fisher, Chas. E. Gorton, Arthur M. Greene, Jr., Richard Hammond, A. L. Humphrey, Chas. L. Huston, D. S. Jacobus, S. F. Jeter, Wm. F. Kiesel, Jr., W. F. MacGregor, Edward F. Miller, M. F. Moore, Irving E. Moultrop, Richard D. Reed, Henry G. Stott, H. H. Vaughan.

As the Am.Soc.M.E. Boiler Code comes into use more and more throughout the country, questions are put to the Society regarding the meaning or application of particular rules embodied in it. The procedure in the Committee in handling these questions is that all inquiries be in written form, copies are sent to each member of the Committee, the interpretation is then prepared by the Committee and passed upon at a regular meeting of the Committee, and in turn submitted to the Council; when approved by the Council the reply is sent to the inquirer and is also published in THE JOURNAL.

Monthly meetings of the Committee have been held during 1916, at which 112 inquiries have been considered and interpretations given.

A public hearing will be held in December of this year.

Increasing recognition of the value of the Code is indicated by its use in technical schools as a text or reference book. It is now used as a reference book at Stevens Institute of Technology; at Sheffield Scientific School of Yale University; and in the course in boiler design at the Rensselaer Polytechnic Institute. It is reported that the Code is also in use at Virginia Polytechnic Institute, Georgia School of Technology, University of Texas, and The Tulane University of Louisiana.

The U. S. Government now specifies that boilers in use at the Canal Zone are to be in accordance with the Code. It is operative as a legal construction code in the States of Wisconsin, Indiana, Ohio, Pennsylvania and California. The Code has also been adopted as a standard by the Industrial Commissions of Pennsylvania and California, and by fifteen of the leading boiler insurance companies.

Committee on Filter Standardization. This committee, George W. Fuller, *Chairman;* Jas. C. Boyd, Arthur M. Crane, Philip N. Engel, Martin F. Newman, Wm. Schwanhausser, was appointed to make recommendations for rating the capacity of mechanical filters. The committee records with regret the death in 1915 of Mr. J. C. W. Greth, one of the members of the original committee and who gave a great deal of assistance in the collection of data on the practical state of the art and assisted greatly in his judicially expressed opinion as to the basis of the committee's report.

The committee has made a very wide canvass to secure all information and data available, and has based its recommendations on normal practice.

Increase of Membership Committee. This committee, with its sub-committees in various parts of the country, has been effectively at work, and the chairman reports: "This activity is in a very healthful condition, and, generally speaking, our members are interested in the welfare of the Society, have the activity of this committee in mind, and are doing what they can to help. It is a work which requires constant agitation and publicity, and having once started it is desirable to keep pushing it all the time. The special campaign among the college alumni should be continued, and from time to time the membership of the Society should be circularized in a dignified way as has been done in the past. A judicious use of the circular 'Society Service' will be quite helpful."

The general committee and its sub-committees are:

Increase of Membership. Irving E. Moultrop, *Chairman;* Fred. H. Colvin, James V. V. Colwell, Robert M. Dixon, Wm. R. Dunn, Leigh A. Hunt, John P. Ilsley, Edwin B. Katte, Richard B. Sheridan.

Chairmen of Sub-Committees on Increase of Membership: Atlanta, Park A. Dallis; Boston, A. L. Williston; Buffalo, W. H. Carrier; Chicago, Philip N. Engel; Cincinnati, John T. Faig; Cleveland, Arthur G. McKee; Los Angeles, O. J. Root; Michigan, H. H. Esselstyn; Milwaukee, Fred. H. Dorner; Minnesota, Max Toltz; New Haven, E. H. Lockwood; New York, J. A. Kinkead; Philadelphia, T. C. McBride; Rochester, Lucien Buck; St. Louis, John Hunter; San Francisco, Wynn Meredith; Seattle, Robert M. Dyer; Tennessee, E. C. Patterson; Troy, Albert E. Cluett; Worcester, Edgar H. Reed.

Joint Committee on Standards for Graphic Presentation. A preliminary report of the Committee published for the purpose of inviting suggestions has had a kindly reception. A large number of suggestions have been received. In order to properly arrange suggestions for the future work of the Joint Committee on Standards for Graphic Presentation a volunteer committee has been holding frequent meetings for the past six months. This committee is composed of men whose work requires them to handle a large amount of statistical material where graphic presentation is particularly advantageous.

When the volunteer committee has completed its recommendations regarding the suggestions which have thus far been made for standardization, the recommendations will be acted upon by the Joint Committee on Standards for Graphic Presentation. The work of the volunteer committee should vastly reduce the amount of time required for the meetings of the Joint Committee on Standards for Graphic Presentation, which is composed of nineteen men representing nineteen different associations of national scope.

A number of important journals are using the suggestions of the preliminary report for preparing the illustrations used in their current publications. The experience to date shows advantageous results from this standardization work. Within the last year there has been a noticeably large increase in the volume of graphic presentation printed in technical publications, standard magazines, and even daily papers.

Committee on Power Tests. In 1909 this committee of the Society started work on the formulation of new codes for the testing of power-plant apparatus. A preliminary report, published in December, 1912, was widely discussed both orally and in writing, and in the light of the suggestions attending this discussion an extensive revision was made. The revised report was presented to the Council and accepted. It is included in Volume 37 of Transactions.

As originally constituted, the committee consisted of D. S. Jacobus, *Chairman*, Edward T. Adams, George H. Barrus, L. P. Breckenridge, Wm. Kent, Charles E. Lucke, Edward F. Miller, Arthur West, and Albert C. Wood. Dr. Lucke resigned early in 1912 and the vacancy was not filled. Owing to pressure of 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 chairman in place of Dr. Jacobus. Prof. R. H. Fernald and James W. Parker have been added to the Committee this year, by appointment by the Council.

The Council has directed that this Committee hold, from time to time, meetings in the nature of public hearings of interested users, as is done in the case of the Boiler Code.

Committee on Refrigeration. The work done by this committee to date is in the form of a preliminary report published in Vol. 26 of Transactions. Dr. Jacobus, chairman of the committee, is now working along the lines of a suggested joint report with the similar committee of the American Society of Refrigerating Engineers.

Committee on Sections. Elliott H. Whitlock, *Chairman;* W. F. M. Goss, Louis C. Marburg, Walter Rautenstrauch, D. Robert Yarnall. The spirit of coöperation among engineers has influenced to a remarkable degree the activities of the Sections this year. Local problems of affiliation of Sections with other societies have been successfully worked out, and in practically all of the sixteen centers of established Sections joint meetings have now become the custom.

It is very gratifying to note that in this matter of bringing the local societies together, our Sections have in many cases taken the initiative.

Meetings in Atlanta have been held in conjunction with the affiliated technical societies in that city. Coöperation among the engineering societies of Boston has been followed from the inception of the Section six years ago. In Buffalo the Section has coöperated with the Buffalo Engineering Society. The meetings in Cincinnati have generally been joint meetings with the Engineers' Club, as also in Milwaukee and St. Louis. A recent meeting of the Los Angeles Section was held with all the technical societies in that city, the meeting being under the auspices of our Section. The Minnesota Section has held meetings in three cities, St. Paul, Minneapolis and Duluth. In Philadelphia joint meetings have been held with the Engineers' Club and The Franklin Institute. The meetings in Providence have been those of the Providence Association of Mechanical Engineers, now the Providence Engineering Society, affiliated with our Society. In New York there has been coöperation with other organizations and a number of joint meetings have been held, also in New Haven.

Over one hundred meetings have been held by the Sections during the year and all have been reported in The Journal. Nineteen of the papers presented locally have been published in The Journal in full and fourteen others will be published in due course. Eight papers originally presented at Section meetings during the season were subsequently delivered at general meetings and appear in Vol. 37 of Transactions.

This plan is a definite purpose of the Committee on Meetings, approved by the Council, to enhance the dignity of the Section meetings and develop the interest. On account of the necessarily crowded programs of the general meetings of the Society, for which there is no relief in sight, it is obvious that a more extended discussion can be obtained by presentation of papers before Section meetings.

The same coöperative spirit which has brought the Sections into close contact with other local organizations is now manifest in the relations between the Sections themselves and the Sections and the Society proper.

The first Conference of Sections was held at the 1915 Spring Meeting, and was such a success that a second Conference was arranged at the Annual Meeting in New York, December, 1915. A representative from each of the fourteen Sections then established was present, and also delegates from the Providence Engineering Society and from several industrial centers where members of the Society were interested in the formation of Sections. Three business meetings were held and the delegates had opportunity to discuss matters with the officers and members of the Council.

The Nominating Committee for officers of the Society was this year nominated by the Sections at the request of President Jacobus.

Two fine examples of coöperation between the Sections and the parent organization were the receptions accorded the Society at San Francisco and Birmingham, the former on the occasion of the September, 1915, meeting of the Society and the latter when the officers and members were en route to the Spring Meeting in New Orleans.

The plan for having all the Sections elect their officers at a uniform time and by a uniform method, either by vote at an open meeting or by letter ballot, has now been consummated, and the officers for all the Sections for the ensuing year assumed their duties on July 1, 1916.

There has been evidenced a necessity for a Code which would tend to put the activities of all the Sections upon a more uniform basis, and the Committee on Sections has been engaged in formulating such a Code. The Committee is hopeful of securing the coöperation of all the national engineering societies in putting their respective Sections upon the same general basis.

New Sections have been established during the year at Detroit and Indianapolis, and Sections are under contemplation at Baltimore, Erie, Kansas City, Meriden and Rochester.

During the year the President has visited Sections at Philadelphia, Chicago, Milwaukee, Detroit, St. Paul and Buffalo, to consult the members upon matters of import to the Society, and has also attended meetings for the purpose of considering Sections at Erie, Indianapolis and Baltimore. The Secretary has accompanied him on some occasions and has also attended meetings of the New Haven, Philadelphia, Providence, Buffalo, Boston and Birmingham Sections.

Particular mention should be made of the self-sacrificing and effective work of the Chairman of the Sections Committee, E. H. Whitlock, who has been untiring in his interest and whose efforts have accomplished so much in bringing into definite shape the sections organization.

Committee on Standard Flanges and Pipe Fittings. During the year the Manufacturers' Standards Committee, representing the principal companies manufacturing pipes and fittings, requested the coöperation of the Society's Flange Committee in the standardization of flanges for hydraulic work; later our Committee was asked to include flanged fittings for ammonia apparatus and also steel fittings and, by request of the American Railway Master Mechanics' Association, pipe unions, which has been granted.

The title of the committee was changed to the more comprehensive one of Committee on Standards for Flanges and Pipe Fittings. This committee is constituted as follows: Henry G. Stott, *Chairman*, Arthur R. Baylis, Stanley G. Flagg, Jr., E. M. Herr, Arthur M. Houser, Julian Kennedy, E. A. Stillman, W. M. White, A. S. Vogt.

The work of this committee to date has included investigations of the strength of the various sizes and weights of rolled-steel piping obtainable in the market; thickness of pipe walls for cast steel and semi-steel pipe and fittings; proportions for flanges, bolting and fittings for hydraulic pressures of 800, 1200 and 3000 lb. working pressure per square inch.

A 50-pound schedule has also been considered which proposes a slight reduction in diameters of bolts, thicknesses of flanges and wall metal from sizes 12 in. to 100 in. inclusive. The diameters of flanges and bolt circles as well as the number of bolts remain the same as called for in the 125-pound standard. This proposed 50-pound standard, however, is identical with the 125-pound standard for all sizes 10 in. and below. Other topics under discussion are pipe unions, threads, etc.

All the above data have been placed in the hands of each member of the committee with a view toward a final agreement.

Committee on Tolerances in Screw Thread Fits. Soon after its organization, this committee held a number of meetings and outlined a general plan of procedure, including the securing of data on present practice. To carry out these various lines of investigation sub-committees were appointed, one having responsibility for the three main divisions of the work — present practice of satisfactory limits for commercial work, requirements for higher-grade work where conditions are more exacting, and for lower-grade work of rough character; investigation of methods of measuring taps, screws, nuts, etc., and to establish a nomenclature.

These sub-committees have been at work and a great amount of material has been collected on which much time has been spent in an endeavor to find some practical basis for the proposed standards.

The committee is attempting to reconcile conflicting positions, standing as it were between the manufacturer and the user, and to reconcile the interests of both. The committee has also been in communication with the British Engineering Standards Committee, whose standards for limits and tolerances as first given required changes, indicating the difficulty of the problem. The committee is constituted as follows: Luther D. Burlingame, *Chairman;* Ellwood Burdsall, Frederic G. Coburn, Fred H. Colvin, A. A. Fuller, James Hartness, W. R. Porter, Frank O. Wells, Walter F. Worthington, Chas. D. Young.

Committee on Student Branches. The Society has now affiliated with it forty Student Branches, which are independent organizations of students in the following educational institutions:

Armour Institute of Technology	Chicago, Ill.
Bucknell College	Lewisburg, Pa.
Carnegie Institute of Technology	Pittsburgh, Pa.
Case School of Applied Science	Cleveland, Ohio
Colorado State Agricultural College	Fort Collins, Colo.
Columbia University	New York, N. Y.
Cornell University	Ithaca, N. Y.
Georgia School of Technology	Atlanta, Ga.
Kansas State Agricultural College	Manhattan, Kan.
Lehigh University	So. Bethlehem, Pa.
Leland Stanford Jr. University	Stanford University, Cal.
Louisiana State University	Baton Rouge, La.
Massachusetts Institute of Technology	Cambridge, Mass.
New York University	New York, N. Y.
Ohio State University	Columbus, Ohio
Pennsylvania State College	State College, Pa.
Polytechnic Institute of Brooklyn	Brooklyn, N. Y.
Purdue University	Lafayette, Ind.
Rensselaer Polytechnic Institute	Troy, N. Y.
State University of Iowa	Iowa City, Ia.
State University of Kentucky	Lexington, Ky.

Stevens Institute of Technology	Hoboken, N. J.
Syracuse University	Syracuse, N. Y.
Throop College of Technology	Pasadena, Cal.
University of Arkansas	Fayetteville, Ark.
University of California	Berkeley, Cal.
University of Cincinnati	Cincinnati, Ohio
University of Colorado	Boulder, Colo.
University of Illinois	Urbana, Ill.
University of Kansas	Lawrence, Kan.
University of Maine	Orono, Me.
University of Michigan	Ann Arbor, Mich.
University of Minnesota	Minneapolis, Minn.
University of Missouri	Columbia, Mo.
University of Nebraska	Lincoln, Neb.
University of Wisconsin	Madison, Wis.
Virginia Polytechnic Institute	Blacksburg, Va.
Washington University	St. Louis, Mo.
Worcester Polytechnic Institute	Worcester, Mass.
Yale University	New Haven, Conn.

Johns Hopkins University has recently held an organization meeting with the idea of establishing a Student Branch.

Institutions in which the entrance requirements are equivalent to those established by the Carnegie Foundation are eligible for Student Branches. Bodies of students in the engineering departments of such colleges are invited to apply to the Secretary of the Society for information regarding the form of petition to the Council for a Student Branch.

For the first time in the history of the Society a joint meeting of Student Branches was held at the headquarters of the Society on Friday evening, April 14, 1916. Five Student Branches located at the following institutions participated:

Columbia University
New York University
Pennsylvania State College
Polytechnic Institute of Brooklyn
Stevens Institute of Technology

More than 250 undergraduates and members of the faculties of the various colleges were in attendance.

The object of the meeting was to bring the students of the different engineering schools into closer touch with the parent Society and at the same time foster the habit of forming new acquaintanceships and friends among those in the Engineering Profession. In turn this develops the coöperation which is of paramount importance to the continued progress of the profession.

It is planned to make this joint meeting of the Student Branches located in the Metropolitan District an annual affair, and further that whenever possible the branches in other sections of the country arrange to hold occasional joint meetings.

Employment Work. The employment work carried on through the medium of the Secretary's office and of THE JOURNAL has been continued this year, and the work has increased in volume so that the department has now become quite a pretentious one.

An indication of the magnitude of the work of this department may be obtained from the fact that an average of thirty positions available and forty men available are published in THE JOURNAL each month. This represents only a small part of the work of the department, as many positions available are filled and men available recommended to positions before THE JOURNAL goes to press each month. Upwards of 700 positions available have been registered since the first of the year, and the Society has received 117 letters of thanks from men who have secured positions through the department.

COÖPERATION WITH OTHER ORGANIZATIONS

WORK OF JOINT COMMITTEES, ETC., ON WHICH THE SOCIETY IS REPRESENTED

American Association for the Advancement of Science. The representatives of the Society on the council of the Association are W. B. Jackson and, by appointment this year, D. S. Jacobus. This representation is honorary and advisory, but has taken a more active form this year in the plans of the Association for its meeting to be held in New York in December.

Classification of Technical Literature. The Joint Committee on Classification of Technical Literature has held several meetings during the year. The Society is represented on this committee by Fred. R. Low, *Chairman;* William W. Bird, L. P. Breckenridge, Alfred E. Forstall, Edwin J. Prindle. Mr. Low is also Chairman of the joint committee.

The committee has reached an agreement on a tentative outline schedule of the divisions of technical literature, and the development of the parts of this schedule has been assigned to certain of the societies represented on the committee.

Through the courtesy of the Library Board of the United Engineering Society, an assistant has been assigned to the duties of secretary of this committee and will devote full time to the work.

Some of the schedules for divisional subjects are now near completion, and it is hoped to have them ready for submission to the committee in a short time.

Joint Conference Committee of National Engineering Societies. This committee was formed in 1912 for the purpose of considering matters pertaining to the Engineering Profession in general, in distinction to those concerning one branch of it and not extending beyond the scope of the corresponding organization.

The committee consisted formerly of two members from each of the constituent societies, but has now been increased by one additional member from each society. The representatives of our Society are Charles Whiting Baker, *Chairman*, A. M. Greene, Jr., and D. S. Jacobus.

It is planned that the reorganized committee hold regular meetings each month, and that the scope of the joint conference committee be enlarged so as to recommend action on all general or public matters relating to the welfare of the Engineering Profession and in which joint action seems desirable.

Under instructions of our Council, recorded actions of the joint conference committee are reported by our representatives for approval at the meeting next following of our Council.

Conference Committee on Electrical Engineering Standards. An important matter considered in this committee (Henry G. Stott, *Chairman*, Albert F. Ganz, and Carl Schwartz) was the proposal to form a joint standards committee, with representation from all the national engineering societies, to act as a senate and give final approval to any standards proposed by the constituent societies.

Engineer Reserve Corps. Shortly after appointment the Committee on Engineer Reserve Corps held two meetings. The committee lost by death John A. Hill, who was acting-secretary, and James M. Dodge, member of the committee. The committee as now constituted is: William H. Wiley, *Chairman;* W. F. M. Goss, H. A. Giles, Alex. C. Humphreys, Wilfred Lewis.

The work of the members of the committee has been advisory to the chairman, who in turn is the representative of the Society on the Executive Committee of the Joint Committee on Engineer Reserve Corps. This joint committee is composed of committees from the national engineering societies, and the Executive Committee is constituted of the chairmen of these committees.

The Executive Committee has made two trips to Washington to confer with the heads of departments of the Government, and two to Governor's Island to meet General Leonard Wood; they have also attended some ten or fifteen meetings.

The culmination of the joint committee's work is now reached, as the Engineer Reserve Corps was legally established by a provision in the National Defense Act, which became law on July 1, 1916.

The Engineering Foundation. The Society's representatives on The Engineering Foundation are W. F. M. Goss and E. Gybbon Spilsbury.

The Foundation has made radical changes in its policy of such a nature as will, it is felt, greatly increase its usefulness.

The facilities of the organization and of the office of the Foundation have been placed at the service of the National Research Council and accepted. The Foundation will thus take up actively the details of the work planned by the Research Council, which will place the Foundation, and in turn the United Engineering Society, behind the great national movement represented in the Research Council.

This new work requiring the entire time of a secretary, which Dr. F. R. Hutton, who has acted as temporary secretary of the Foundation since its inception, did not care to give, he resigned in September, 1916, and Dr. Cary T. Hutchinson has been engaged as Secretary of the Foundation.

John Fritz Medal Board of Award. The Society is represented on the Board by four appointees, the term of one of which expires at the annual meeting of the Board in January of each year. Dr. F. R. Hutton was reappointed this year for a second term of office; the Society's other representatives are John R. Freeman, Ambrose Swasey and Dr. John A. Brashear.

The award of the John Fritz Medal has this year been made to Dr. Elihu Thomson, of Swampscott, Mass., for "achievements in electrical inventions, in electrical engineering and in industrial development, and in scientific research."

Committee on Metric System. Representatives on the Joint Conference Committee on the Metric System, proposed by the American Institute of Electrical Engineers, were appointed to confer upon the advisability of adopting the Metric System as a working standard in engineering.

The Committee on Metric System of this Society is composed of Luther D. Burlingame, *Chairman*, E. M. Herr, F. A. Halsey, J. Sellers Bancroft, and A. L. De Leeuw. Two members of this committee are to be chosen later to represent the Society on the Joint Conference Committee.

United Engineering Society. The Trustees of the United Engineering Society representing our Society are John R. Freeman, E. Gybbon Spilsbury, and Henry G. Stott, the two latter being the new appointees this year. The Trustees this year invited the American Society of Civil Engineers to join the Founder Societies in the ownership and administration of the property of the United Engineering Society, in the occupancy of the Engineering Societies Building, and in the administration of the Library. By letter ballot of the membership of the Civil Engineers, it was voted to accept the offer. The work of alteration to the building to accommodate the new Founder Society, including adding three stories to provide quarters for it and take care of the enlarged library, was immediately begun and is now well under way.

The hope of Mr. Andrew Carnegie, the donor of the building, that it should become the headquarters of the engineering profession in America, is thus on the point of realization.

NEW JOINT COMMITTEE WORK THIS YEAR

Joint Conference Committee of the National Academy and the National Engineering Societies. A Joint Conference Committee of the National Academy of Sciences and the national engineering societies has been created to develop ways and means for a closer coöperation between the Academy and these organizations. Mr. Gano Dunn was temporary chairman of the committee.

At a meeting on September 19, the Joint Conference Committee organized with J. J. Carty, Past-President of the A.I.E.E., as permanent chairman, and Calvin W. Rice as permanent secretary. At this meeting it was announced that the Academy had this year created a section of engineering and would elect engineers to the Academy.

Dr. Ira N. Hollis is the representative of the Society on this committee.

National Research Council. At its annual meeting in April, 1916, the National Academy of Sciences volunteered to the President of the United States to organize the scientific resources of the educational and research institutions of the country in the interest of national preparedness. This offer was immediately accepted, and President

Wilson in turn invited the President of the National Academy of Sciences to appoint a National Research Council. This council was chosen mainly, if not entirely, from members of the National Academy of Sciences and of the national engineering societies.

The purpose of the council is to bring into coöperation existing governmental, educational, industrial, and other research organizations, with the object of encouraging the investigation of natural phenomena, the increased use of scientific research in the development of American industries, the employment of scientific methods in strengthening the national defense, and such other applications of science as will promote the national security and welfare.

The following members of this Society received appointment on the National Research Council: Dr. John A. Brashear, Honorary Member of the Society, Mr. Gano Dunn, President, The J. G. White Engineering Corporation; Dr. W. F. M. Goss, Dean of Engineering, University of Illinois; Dr. S. W. Stratton, Director, Bureau of Standards; Mr. Ambrose Swasey, Honorary Member of the Society.

The Engineering Foundation of the United Engineering Society passed resolutions commending the purposes of the National Research Council and offered the organization the use of an office in the Engineering Societies Building, and the services of Dr. Cary T. Hutchinson, secretary of the Foundation. This offer was accepted, and Dr. Hutchinson was appointed secretary of the council.

Lectures on Military Engineering. Early in the year a course of military engineering lectures was organized under the auspices of the officers of the national engineering societies by officers of the U. S. Army provided by Major General Wood, Commander of the Department of the East, and the course was given in New York City in the Engineering Societies Building and the building of the American Society of Civil Engineers.

The success of the lectures exceeded the most sanguine expectations. Over 3000 men attended.

The movement spread beyond the metropolis. On March 29 in Detroit a course of five weekly lectures for engineers was commenced by Col. M. M. Patrick and Major P. S. Bond, Corps of Engineers, U. S. A. In Chicago eleven weekly events, including excursions to a citizen's training camp established near Indianapolis, were inaugurated on April 20; correspondence courses and practical work in field engineering and maneuvers by members of the training camp were organized during the summer.

Courses of lectures were also given in Pittsburgh, Buffalo and Los Angeles.

In all cases the work was carried out under the direction of committees on Military Engineering Lectures, and the activities of these committees and similar committees included organization of and participation in "preparedness" parades.

Coöperation with the Architects. The American Institute of Architects requested and received the coöperation of the Council in a protest against the proposed power plant in the park area of Washington, D. C.

The Institute has also suggested that a clearing house be formed for coöperation in matters of mutual interest to engineers and architects. This suggestion has been referred to the Joint Conference Committee of the National Engineering Societies.

Pan-American Engineering Committee. A permanent committee has been constituted to foster greater engineering and scientific intercourse between South and Central America and the United States and to promote the welfare of the Engineering Profession in these countries.

The committee is composed of five representatives from each of the four national engineering societies. The appointees of our Society are W. H. Bixby, C. T. Plunkett, S. W. Stratton, Ambrose Swasey and C. C. Thomas.

Conference Committee on Electric Power. The purpose of this committee is to coöperate with the Sub-committee on Cost of Electric Power of the Committee on Standards of the American Institute of Electrical Engineers. The representatives of the Society on this committee are D. C. Jackson, A. H. Kruesi, R. J. S. Pigott, John A. Stevens and B. F. Wood. H. G. Stott, Mem.Am.Soc.M.E., is chairman of the joint committee.

John Ericsson Memorial. Early in the year a bill was presented before Congress to erect a monument in Washington to the memory of Captain John Ericsson. A hearing of this bill was held in Washington on March 13, at which Erik Oberg, C. A. V. Carlsson, Gustave Fast, H. A. Gillis, O. Ohlson, C. von Philp and A. H. Raynal, representing the Society, were present. Our thanks are due these gentlemen for their efforts to attend this hearing and for their effective service.

It is of interest to note that Captain Ericsson was a member of the Society, joining in 1881. At the time of his election the distinction of honorary membership was offered to him but he refused it, desiring, as he put it, "to come into the rank and file."

Joseph A. Holmes Memorial. The Joseph A. Holmes Safety Association, formed to perpetuate the memory and work of the first director of the U. S. Bureau of Mines, who died in July, 1915, was formally established by old associates on March 3, 1916.

The association will meet once each year in Washington, when awards and medals will be publicly announced, with honorariums to those who perfect the most efficient devices in the mining, quarrying and metallurgical industries.

Dr. John A. Brashear, Honorary Member and Past-President, is our representative on the committee of awards. He is also a member of the Executive Committee of the Association. General Wm. H. Bixby, Mem.Am.Soc.M.E., has represented Dr. Brashear and the Society at the conferences.

Massachusetts Institute of Technology. On June 12 to 14 the Institute celebrated its semi-centennial and dedicated its new buildings in Cambridge. On invitation, the Society was represented by the President, Dr. Jacobus.

Rutgers College. The 100th anniversary of the founding of Rutgers College was celebrated on October 13 to 15, and at the invitation of the college the Society was represented by Spencer Miller, member of the Council.

Engineering Coöperation. Two conferences on engineering coöperation were held this year by representatives of various societies, under the auspices of a committee self-constituted with Dr. Frederick H. Newell, Mem.Am.Soc.M.E., Chairman, and C. E. Drayer, Secretary. Over thirty technical and engineering societies, clubs and like organizations were present at the second conference, which was held in Chicago.

H. C. Gardner was the delegate of this Society at the second conference, at which discussion upon the plan of attaining the object in mind centered upon whether a separate organization should be formed or how best the activities of the existing societies could be correlated. The representative of our Society argued for the development of engineering coöperation through existing organizations.

No. 1531

MEETINGS JANUARY—JUNE

MEETINGS OF SECTIONS

The spirit of coöperation among engineers influenced to a remarkable degree the activities at the Section meetings during 1916. Local problems of affiliation of Sections with other societies were successfully worked out, and in practically all of the centers of established Sections joint meetings have now become the custom.

Meetings in Atlanta were held in conjunction with the affiliated technical societies in that city. Coöperation among the engineering societies of Boston has been followed from the inception of the Section six years ago. In Buffalo the Section coöperated with the Buffalo Engineering Society. The meetings in Cincinnati were generally joint meetings with the Engineers' Club, as also in Milwaukee and St. Louis. One meeting of the Los Angeles Section was held with all the technical societies in that city, the meeting being under the auspices of our Section. The Minnesota Section held meetings in three cities, St. Paul, Minneapolis and Duluth. In Philadelphia joint meetings were held with the Engineers' Club and The Franklin Institute. The meetings in Providence were those of the Providence Association of Mechanical Engineers, now the Providence Engineering Society, affiliated with our Society. In New York there was coöperation with other organizations and a number of joint meetings were held, also in New Haven.

BUFFALO, JANUARY 5

Subject: Iron in Antiquity and Today, Dr. John A. Mathews, Mem.Am.Soc.M.E.

NEW YORK, JANUARY 11

Paper: Standardization of Power Plant Operating Costs, Walter N. Polakov, Mem.Am.Soc.M.E. Published in this volume.

BOSTON, JANUARY 14

Address: Naval Lessons of the Great War for America, Dr. Ira N. Hollis, Mem.Am.Soc.M.E.

CHICAGO, JANUARY 14

Paper: High Temperature Insulation, P. A. Boeck, mechanical engineer, Kieselguhr Company of America, Chicago, Ill. Published in The Journal, August, 1916.

LOS ANGELES, JANUARY 18

Subject: Patents, F. W. Harris, Mem.Am.Soc.M.E. Preceded by a report by W. W. Smith on the Conference of Sections at the Annual Meeting.

ST. LOUIS, JANUARY 19

Joint meeting of the St. Louis Engineering Societies. Subject: Certain Phases of Scientific Management of Machine Shops, Carl G. Barth, Mem.Am.Soc.M.E.

BUFFALO, JANUARY 20

Subject: Electric Railway Signalling, Henry M. Sperry, General Electric Railway Company.

MINNESOTA, JANUARY 20

Banquet at St. Paul. Charles W. Tubby, Mem.Am.Soc.M.E., Chairman, Minnesota Section, gave a history of the Section; Prof. Wm. H. Kavanaugh, Mem.Am.Soc.M.E., spoke on engineering education; and C. L. Pillsbury, Mem.Am.Soc.M.E., spoke of the value of engineering societies.

CINCINNATI, JANUARY 21

Joint meeting with the Engineers' Club of Cincinnati. The Committee on Concrete presented a report of its investigations.

BIRMINGHAM, JANUARY 26

Dinner meeting. General discussion relative to the affairs of the Section.

PROVIDENCE, JANUARY 26

Meeting of the Providence Association of Mechanical Engineers. Franklin H. Wentworth, secretary of the National Fire Protection

Association, spoke on the work of the Underwriters' Laboratories in the direction of fire protection. Illustrated by moving pictures.

BUFFALO, FEBRUARY 2

Illustrated Address: Public Service Problems, Morris L. Cooke, Mem.Am.Soc.M.E.

PHILADELPHIA, FEBRUARY 3

Joint meeting with The Franklin Institute. Paper: The Development of the Pumping Engine, Prof. Arthur M. Greene, Jr., Manager, Am.Soc.M.E.

ST. LOUIS, FEBRUARY 6

Joint meeting of the St. Louis Engineering Societies. F. W. Doolittle, consulting engineer, New York, N. Y., spoke on the Cleveland Street Railway situation. Illustrated.

BIRMINGHAM, FEBRUARY 7

Appointment of Executive Committee, Committee on Program and Papers, and Membership Committee. Major General Leonard Wood spoke on Preparedness and the Military Training Camp at Chattanooga.

BOSTON, FEBRUARY 8

Seventh 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: Samuel W. McCall, Governor of Massachusetts, Col. W. E. Craighill, of the Corps of Engineers, U. S. A., William Barclay Parsons, consulting engineer, New York, N. Y., John R. Freeman and Dr. John A. Brashear, Past-Presidents, Am.Soc.M.E., and Calvin W. Rice, Secretary, Am.Soc.M.E.

NEW YORK, FEBRUARY 8

Paper: Ways of Presenting Data for Executive Purposes, T. Russell Robinson, statistical engineer, W. S. Barstow and Company, Inc., New York, N. Y.

ST. LOUIS, FEBRUARY 9

Joint meeting of the Engineers' Club of St. Louis with the local societies. Address: Recent Practice on Concrete Roads, J. B.

Marcellus, engineer of the Association of the American Portland Cement Manufacturers. Illustrated.

MILWAUKEE, FEBRUARY 10

Joint meeting of the local engineering societies, under the auspices of the Milwaukee Section of the American Institute of Electrical Engineers. Paper: The Design and Application of Electromagnets for Industrial Purposes, Arthur Simon, Mem.Am.Soc.M.E. Illustrated by moving pictures.

MINNESOTA, FEBRUARY 10

Illustrated lecture given at Minneapolis: The Practical Chemistry and Modern Manufacture of Portland Cement, George P. Diekmann, chief chemist of the Northwestern States Portland Cement Company, Mason City, Ia. Published in THE JOURNAL, May, 1916.

BOSTON, FEBRUARY 16

Joint meeting under the auspices of the Boston Society of Civil Engineers. Paper: The Spillways of the Panama Canal, Edward C. Sherman. Illustrated.

BUFFALO, FEBRUARY 16

Lecture: Uses of Modern Worm Gearing, Cornelius T. Myers, Mem.Am.Soc.M.E.

WORCESTER, FEBRUARY 16

Address: What Constitutes a Well-Rounded Fleet? Dr. Ira N. Hollis, Mem.Am.Soc.M.E.

PROVIDENCE, FEBRUARY 23

Meeting of the Providence Association of Mechanical Engineers. Address: The Handling of Steam Coal from the Mine to the Consumer, Prof. James A. Hall, Mem.Am.Soc.M.E.

CINCINNATI, FEBRUARY 24

Joint meeting with the Engineers' Club of Cincinnati. Illustrated lecture: A Rapid Transit System and Interurban Facilities for the City of Cincinnati, Morse W. Rew, assistant engineer of the Board of Rapid Transit Commissioners.

BUFFALO, MARCH 1

Paper: The Problems of a Consulting Engineer, T. Kennard Thomson, Mem.Am.Soc.M.E.

ST. LOUIS, MARCH 1

Subject: The Collection and Disposal of City Refuse, Hiram Phillips.

BIRMINGHAM, MARCH 2

Business meeting at which new Constitution and By-Laws were adopted.

NEW HAVEN, MARCH 8

Informal meeting followed by noon-day luncheon.

MINNESOTA, MARCH 9

Held at the offices of the St. Paul Gas Light Company, St. Paul. Paper: Uniflow Engine, Herman F. Mueller, Mem.Am.Soc.M.E.

BOSTON, MARCH 14

Paper: Steam Safety Valves, George H. Clark, instructor in mechanical engineering, Massachusetts Institute of Technology. Published in this volume.

NEW YORK, MARCH 14

Joint meeting with the New York Section of the Illuminating Engineering Society to discuss the Code of Lighting for Factories, Mills and Other Work, prepared by committees of the latter society.

BUFFALO, MARCH 15

Illustrated Address: History of Pumping Stations, Prof. Arthur M. Greene, Jr., Manager, Am.Soc.M.E.

CINCINNATI, MARCH 16

Joint meeting with Engineers' Club of Cincinnati. Papers: The Sales Engineer in His Relation to Production and Machine Design, Arthur J. Baker, Mem.Am.Soc.M.E., published in The Journal, November, 1916; Recent Developments in the Recovery of Casing-head Gasoline, Paul Diserens, Jun.Am.Soc.M.E.; Why is an Engineer? W. G. Franz, Mem.Am.Soc.M.E.; Probable Future Requirements in Machine Tools, A. M. Sosa.

CHICAGO, MARCH 17

Paper: Heavy-Oil Engines, S. B. Daugherty, Mem.Am.Soc.M.E. Published in The Journal, October, 1916.

PROVIDENCE, MARCH 22

Meeting of the Providence Association of Mechanical Engineers. Address: Vacuum Steam Heating, J. W. Hook, Assoc-Mem.Am. Soc.M.E.

LOS ANGELES, MARCH 28

Paper: Motor Trucks and Their Use in Southern California, W. H. Clapp, Mem.Am.Soc.M.E. Published in THE JOURNAL, October, 1916.

BUFFALO, MARCH 29

Stereopticon Lecture: The Engineering Arm of Our Army, Major L. V. Frazier of the United States Corps of Engineers.

NEW YORK, MARCH 29

Dinner and Dance. Speakers: Fred. R. Low, Mem.Am.Soc.M.E., Spencer Miller, Manager, Am.Soc.M.E., member of the Naval Consulting Board, and Calvin W. Rice, Secretary, Am.Soc.M.E.

BOSTON, APRIL 5

Subject: The Shoe Industry, Joseph J. Gillespie. Illustrated.

NEW HAVEN, APRIL 5

Spring meeting, held jointly with the Civil, Electrical and Mining Engineers of New Haven. Paper at afternoon session: The Water Powers of New England, H. I. Harriman, civil engineer, of Boston, Mass., illustrated. Followed by a general discussion and by a brief talk by Calvin W. Rice, Secretary, Am.Soc.M.E., upon the work that the Society is doing in coöperating with the other national engineering societies toward the welfare of the country.

Following this session, a dinner was served in the hall of the Yale Dining Club. At the evening session, Samuel Insull, president of the Commonwealth Edison Company of Chicago, presented a paper on The Progress of Economic Power Generation and Distribution. Published in THE JOURNAL, November, 1916.

ST. LOUIS, APRIL 5

Subject: Engineering Geology, Professor McCort of Washington University.

PHILADELPHIA, APRIL 10

Joint meeting with the Philadelphia Section of the American Institute of Electrical Engineers. General Subject: Some Prime

Movers Now Under Development. Papers: The Uniflow Engine, Allan D. Skinner, Mem.Am.Soc.M.E.; Future Possibilities of the Large Steam Turbine Generator in Electric Generating Service, W. C. Eglin, Mem.Am.Soc.M.E.; The Diesel Engine, L. B. Harris of the Southwark Foundry and Machine Company.

BUFFALO, APRIL 12

Subject: Bringing a Shop up to Date, Fred G. Kent, works manager of the Lodge and Shipley Company, Cincinnati, O.

ST. LOUIS, APRIL 12

Subject: Railroad Signals, B. H. Mann, Missouri Pacific R. R.

MINNESOTA, APRIL 14

Joint meeting with the American Institute of Electrical Engineers and the American Society of Civil Engineers at Minneapolis. Illustrated talk by A. L. Rohrer, Mem.Am.Soc.M.E., on the Electrification of the Butte, Anaconda and Pacific Railway. George H. Hill, assistant engineer of the railway traffic department of the General Electric Company, also spoke on the Electrification of the Chicago, Milwaukee and St. Paul Railroad. M. E. Simpson, mechanical superintendent of the Minneapolis Steel Machinery Company, spoke on the Manufacture of High Explosive Projectiles, and Edwin Rollman, Captain, Cadet Corps, University of Minnesota, on the Method of Firing Projectiles. Both of these talks were illustrated.

NEW YORK, APRIL 20

Joint meeting with the Society of Automobile Engineers. Topic: The Four-Cylinder vs. the Multi-Cylinder Motor. Papers: Internal Combustion Motors; The Four-Cylinder Motor of Today and Future Possibilities, Finley R. Porter, Mem.Am.Soc.M.E.; Automobile Applications of Four-Cylinder Motors, F. E. Watts.

CINCINNATI, APRIL 21

Meeting under the auspices of the Engineers Club of Cincinnati. Paper: Prehistoric Engineering in Southwestern Ohio, Fred W. Hinkle.

PHILADELPHIA, APRIL 23

Informal dinner followed by an illustrated talk on Society Affairs by Dr. D. S. Jacobus, President, Am.Soc.M.E.

BUFFALO, APRIL 26

Subject: National Preparedness. Speakers: Elmer A. Sperry, Mem.Am.Soc.M.E., on the work of the Naval Consulting Board, and Calvin W. Rice, Secretary, Am.Soc.M.E., on Industrial Mobilisation.

PROVIDENCE, MAY 3

First annual dinner of the Providence Engineering Society, reorganized from the Providence Association of Mechanical Engineers. Henry A. Wise Wood spoke on Preparedness. Additional speakers were: President W. H. P. Faunce of Brown University, Mayor Gainer of Providence, Arthur H. Annan, Mem.Am.Soc.M.E., Luther D. Burlingame, Mem.Am.Soc.M.E., Calvin W. Rice, Secretary, Am.Soc.M.E., Prof. Hardy Cross and Dr. Charles F. Scott, Mem. Am.Soc.M.E.

LOS ANGELES, MAY 4

Luncheon Meeting. Subject: Industrial Preparedness. W. W. Smith, chairman of the Section, made a brief address, and a letter was read from Howard E. Coffin, Mem.Am.Soc.M.E., chairman of the Committee on Production, Organization, Manufacture and Standardization of the Naval Consulting Board.

NEW YORK, MAY 9

Paper: Report upon Efficiency Tests of a 30,000-kw. Cross-Compound Steam Turbine, Henry G. Stott, Mem.Am.Soc.M.E., and W. S. Finlay, Jr., Mem.Am.Soc.M.E. Published in this volume.

BIRMINGHAM, MAY 12

Reelection of officers of the Section, followed by a general discussion concerning letters received from the State Committee of the Naval Consulting Board and from Thomas E. Durban, Mem.Am. Soc.M.E., chairman of the Uniform Boiler Law Society.

CHICAGO, MAY 15

Paper: The Use of Powdered Coal as a Fuel, Joseph Harrington, Mem.Am.Soc.M.E. Illustrated. Published in The Journal, October, 1916.

CINCINNATI, MAY 18

Illustrated Address: Engineering in Antiquity. Prof. John M. Burnham, Professor of Latin, University of Cincinnati.

ST. LOUIS, MAY 20

Dinner Meeting. Speaker: W. A. Layman, president of the Wagner Electric Company, on Mobilization of American Industries for National Protection, followed by brief talks on the same subject.

BOSTON, MAY 23

Paper: Steam-Flow Measurement, Ervin G. Bailey, Mem.Am. Soc.M.E. Published in THE JOURNAL, October, 1916.

PHILADELPHIA, MAY 23

Paper: Efficiency of Propulsive Machinery and Late Developments in Naval Engineering, Lieut-Commander H. C. Dinger, U. S. N., published in this volume.

CHICAGO, MAY 24

Special meeting. Illustrated address by Dr. D. S. Jacobus, President, Am.Soc.M.E., on Society Affairs.

PROVIDENCE, MAY 24

Meeting of the Providence Engineering Society. Illustrated address on The Fiskeville Weir, by William Pickersgill, designing engineer of the Providence Water Supply Board, followed by Prof. A. E. Watson of Brown University on The New Wireless Installation at Brown University.

BIRMINGHAM, MAY 30

Subject: From Ore to Finished Product; pictures shown by the National Tube Company, followed by an informal meeting.

LOS ANGELES, JUNE 1

Joint meeting of the technical societies in Los Angeles. Subject: Industrial Preparedness. Speakers: Rev. George N. Davidson; Earle Remington, president of the Aero Club of Los Angeles; Capt. Charles T. Leeds, U. S. A. (retired); Major George B. Pillsbury, U. S. A., in charge of fortifications, San Pedro; George W. Dickie, Vice-President, Am.Soc.M.E., and Willis H. Booth, Vice-President of the Security National Bank.

CINCINNATI, JUNE 15

Joint meeting with the Engineers' Club of Cincinnati. Subject: Preparedness. Speaker, Major P. S. Bond, Corps of Engineers, U. S. A. Illustrated.

PROVIDENCE, JUNE 28

Providence Engineering Society. Illustrated lecture: The Scituate Water Supply, by F. E. Winsor, chief engineer of the Providence Water Supply. Other speakers on same subject were G. T. Seabury, engineer of the Water Supply Board, and F. E. Waterman, also engineer of the Board.

NEW HAVEN, JUNE 29

Illustrated Address: The Diesel Motor, F. J. G. Reuter, Assoc-Mem., Am.Soc.M.E.

THE SPRING MEETING

NEW ORLEANS, LA., APRIL 11 TO 14

The Spring Meeting of the Society, held for the first time in New Orleans, La., convened April 11 to 14, with headquarters at the Hotel Grunewald. There was a good attendance, the registration totaling 485, of which 114 were members. Ninety-two of those in attendance were from out of the city.

On the Monday preceding the convention, the Birmingham Section provided a unique entertainment for those journeying to the convention by way of that city. The program at Birmingham included inspection trips to manufacturing plants in the vicinity, and a barbecue dinner was served at Bayview, on the shore of an artificial lake used for water supply. An informal dinner was held in the evening, at which several addresses were made. R. E. Brakeman was Chairman and Paul Wright was Secretary of the Committee which had this program in charge.

On the afternoon of the day of arrival at New Orleans trips were made to the French quarter of the city, and in the evening an informal reception and dance was held in the ballroom of the Hotel Grunewald.

At the first session on Wednesday morning, addresses of welcome were made by Chairman W. B. Gregory of the Local Committee of Arrangements, by A. M. Shaw, past-president of the Louisiana Engineering Society, and by Hon. Martin Behrman, Mayor of the City of New Orleans. Dr. D. S. Jacobus, President, Am.Soc.M.E., responded, following which a session on Organizing for Industrial Preparedness was held. Mr. Spencer Miller, Manager, Am.Soc. M.E., and Representative of the Society on the U. S. Naval Consulting Board, read a paper on this subject, and discussion of it continued throughout the morning. In the evening, after an address by Commissioner W. B. Thompson, more time was given to the consideration of the subject. The Society was honored by receiving a communication from the President of the United States, congratulating the engineers on the movement for national preparedness being undertaken by them, and heartily commending it. The Secretary of the Navy also sent a letter expressing his appreciation of the service which the Society is rendering to the country.

On Wednesday afternoon an inspection trip was made to the harbor of the Port of New Orleans formed by the Mississippi River. During the trip a landing was made at the recently constructed public cotton warehouses. Many interesting facts about the harbor and the warehouses were brought out in the lecture by Commissioner Thompson, given in the evening, on The Debt of New Orleans to the Engineer.

The Thursday morning session was devoted to engineering subjects of local interest, with papers on multiple evaporators, low-lift pumping plants and the mechanical equipment in the Port of New Orleans.

On Thursday afternoon the members and guests were taken in automobiles to the Country Club, from whence they proceeded to the Newcomb School of Art, a department of Newcomb College, the women's college of Tulane University.

Two simultaneous sessions, at which six papers were presented, were held on Friday morning. On Friday afternoon a party was taken to the New Orleans Lake Shore Land Company's project to inspect the reclamation work in process there. Some of the visitors remained in New Orleans until Saturday and continued their trips about the city.

The local membership of the Society, the Association of Local Members of the American Society of Civil Engineers, and the Louisiana Engineering Society were the hosts at the Spring meeting and the arrangements were perfectly worked out through their local committee, W. B. Gregory, Chairman, A. L. Black, Treasurer, and H. L. Hutson, Secretary. The chairmen of the several local subcommittees were George H. Davis, Finance Committee, H. F. Rugan, Committee on Hotels, A. M. Lockett, General Entertainment Committee and Committee for Entertainment for Ladies, R. T. Burwell, Reception Committee, E. L. Jahncke, Committee on Printing and Publicity.

PROGRAM

Tuesday Morning, April 11

Registration of members and guests at headquarters.

Tuesday Afternoon

Inspection trips to points of interest in the city.

Tuesday Evening

Informal reception and dance.

Wednesday Morning, April 12

BUSINESS MEETING

Committee reports, discussion, new business.

PROFESSIONAL SESSION

ORGANIZING FOR INDUSTRIAL PREPAREDNESS, Spencer Miller.

Wednesday Afternoon

Trip to inspect harbor of the Port of New Orleans.

Wednesday Evening

Lecture by Commissioner W. B. Thompson on The Debt of New Orleans to the Engineer.

Thursday Morning, April 13

PROFESSIONAL SESSION

CAPACITY AND ECONOMY OF MULTIPLE EVAPORATORS, E. W. Kerr.

THE EVOLUTION OF LOW-LIFT PUMPING PLANTS IN THE GULF COAST COUNTRY, William B. Gregory.

MECHANICAL EQUIPMENT USED IN THE PORT OF NEW ORLEANS, William von Phul.

Thursday Afternoon

Trip to Country Club and tea at Newcomb College Art Pottery.

Thursday Evening

Reception and dance.

Friday Morning, April 14

SIMULTANEOUS SESSIONS

ON MEASUREMENT OF FLOW OF FLUIDS

ESTABLISHING A STANDARD OF MEASUREMENT FOR NATURAL GAS IN LARGE QUANTITIES, Francis P. Fisher.

DEVIATION OF NATURAL GAS FROM BOYLE'S LAW, Robert F. Earhart and Samuel S. Wyer.

SOME EXPERIMENTS ON WATER FLOW THROUGH PIPE ORIFICES, Horace Judd.

MISCELLANEOUS SESSION

DYNAMIC BALANCE, N. W. Akimoff.

THE MEASUREMENT OF VISCOSITY AND A NEW FORM OF VISCOSIMETER, H. C. Hayes and G. W. Lewis.

ON THE TRANSMISSION OF HEAT IN BOILERS, E. R. Hedrick and E. A. Fessenden.

No. 1532

ORGANIZING FOR INDUSTRIAL PREPAREDNESS

By Spencer Miller,[1] New York, N. Y.
Member of the Council of the Society

The President of the United States, Woodrow Wilson, speaking from his intimate contact with foreign affairs, recently said, "There may come a time when I cannot preserve *both* the honor and the peace of the United States." He also urged that we should be "very adequately prepared, not for war but for defense."

Both Ex-Presidents Taft and Roosevelt have declared for the immediate and adequate preparedness of our army and navy.

Our highly trained army and navy experts, whose life work entitles them to speak with authority, have been most earnest for far more effective military defenses.

The public press reflecting largely public opinion is overwhelmingly in favor of adequate preparedness.

With so formidable an array of authoritative opinion engineers patriotically, quietly and earnestly ask the question, "What may we do to serve our country?"

Engineers are for peace with honor and will make personal sacrifices to insure it.

THE UNITED STATES NAVAL CONSULTING BOARD

It was a proud day for the engineers and scientists when Secretary Daniels of the U. S. Navy Department invited Thomas A. Edison, our own Society and ten other engineering and scientific societies to form a civilian advisory board of twenty-three members to aid in mobilizing the engineering and industrial world for preparedness. This invitation was eagerly accepted and the U. S. Naval Consulting Board is now fully organized with Mr. Edison as Chairman. Ten members of our Society are members of this board; two

[1] Member U. S. Naval Consulting Board.

Presented at the Spring Meeting, New Orleans, La., April 1916, of The American Society of Mechanical Engineers.

representing it officially.[1] This board serves without monetary compensation, even paying its own expenses.

The Naval Consulting Board has resolved itself into committees, one of which, headed by Howard E. Coffin, member of this Society, is the Committee on Production, Organization, Manufacture and Standardization. This committee formulated a plan for aiding in organizing the industries for preparedness. This plan, laid before the President of the United States, prompted him to invite the five leading engineering societies, whose combined membership is about 35,000, to effect such an organization. This he did by writing identical letters to each of the following societies:

The American Society of Mechanical Engineers
The American Society of Civil Engineers
The American Institute of Electrical Engineers
The American Institute of Mining Engineers
The American Chemical Society

THE PRESIDENT'S CALL

The President's letter to our Society, dated White House, January 13, 1916, reads as follows:

> The work which The American Society of Mechanical Engineers has done, through its members on the Naval Consulting Board, is a public service which is deeply appreciated. It has been so valuable that I am tempted to ask that you will request the society to enlarge its usefulness to the Government still further by nominating, for the approval of the Secretary of the Navy, a representative from its membership for each state in the Union to act in conjunction with representatives from the American Institute of Mining Engineers, the American Society of Civil Engineers, the American Institute of Electrical Engineers and the American Chemical Society, for the purpose of assisting the Naval Consulting Board in the work of collecting data for use in organizing the manufacturing resources of the country for the public service in case of emergency.
>
> I am sure that I may count upon your cordial coöperation.
>
> (Signed) Woodrow Wilson

This letter appeared everywhere in the public press and was received with a glow of patriotism in the hearts of engineers generally.

THE PRESIDENT'S INVITATION ACCEPTED

Each society named has accepted the President's invitation. Each has selected one director from every state of the Union, for the approval of the Secretary of the Navy. The five directors in each state will constitute a Committee of State Directors for Industrial

[1] W. L. R. Emmet and the author.

Preparedness. These State Directors will report to the Naval Consulting Board through the Committee of which Mr. Coffin is chairman.

AN INDUSTRIAL INVENTORY

The immediate work of these State Directors will be to obtain an accurate inventory of all the facts necessary to be known to the army and navy of the resources of our nation to supply munitions of war in case of need.

To obtain this census the State Directors will be assisted by the members of each of the five representative societies residing within the state. In many states there are more members than industries in which the army and navy may be interested. This indicates that this census will be rapidly completed.

ITS IMPORTANCE

The importance of such a census can scarcely be overstated. It has been estimated that for every million of soldiers and sailors at the front at least three million of workers will be needed to maintain them efficiently with the necessary implements and supplies of warfare.

Major General Leonard Wood, U. S. A., estimates that a serious war would require at least 2,000,000 men. Under such circumstances 6,000,000 men would be required for producing munitions of war.

SOME STATISTICS ABOUT WARS IN THE UNITED STATES

War of the Revolution	1775–1783	Number of troops	309,791
War of 1812	1812–1815	Number of troops (Incl. Navy)	576,622
Civil War	1861–1865	Number of troops (Incl. Navy)	2,778,302

Cost of Civil War to date, including pensions (about)	$16,000,000,000
Present estimated wealth of the U. S.	$200,000,000,000
Present estimated population of the U. S.	100,000,000
Present estimated wealth per capita	2,000
Public debt (less cash in treasury)	1,027,000,000
Public debt per capita	10
Appropriation in 1915, navy (about)	$145,000,000
Appropriation in 1915, army and forts (about)	108,000,000
Cost of navy per capita	1.45
Cost of army per capita	1.08

David Starr Jordan, the eminent pacifist, recently said, "*In all ages war costs all that it can.*"

Benjamin Franklin observed, "*War is not paid for in war time, the bill comes later.*"

The Civil War cost about sixteen billion dollars, or about as much as the whole estimated wealth of the nation at that time.

Have we any right to assume that a war with one or more first-class powers finding us unprepared would cost less than one-quarter of our national wealth, namely fifty billions of dollars?

Such a cost represents one-quarter of the wealth of each of our hundred million population on an average, to say nothing of lives lost and personal property destroyed.

To insure this country against successful invasion, a much greater navy is generally demanded. In fact, a navy superior to that of any nation save only England. Some declare even that our navy should exceed that of England on the ground that we may have to fight two great nations at once.

Today our navy costs about $1.50 per capita per annum, average. Even though our navy be doubled, we would only pay $3.00 per capita, average.

Our per capita wealth is about $2000. Is $3 per year a heavy tax on $2000 worth of property if such a tax would provide a navy adequate to defend us against successful invasion?

MUNITIONS OF WAR

The term munitions of war covers the whole field of army and navy requirements: guns and ammunition, rifles and uniforms, food and blankets, tents and cooking equipment, automobiles and aeroplanes, battleships and submarines, coal and oil, medical supplies and hospital equipment, even mules and horses. Thus it will be apparent that this volunteer army of engineer census takers must needs visit woolen mills as well as machine shops, boot and shoe factories as well as foundries, coal mines as well as ship yards.

INDUSTRIAL INVENTORY

A detailed printed form has been prepared for taking the industrial inventory. It will be " a strictly confidential, non-partisan, non-political and wholly patriotic inventory of our country's manufacturing and producing resources. The information given upon this form will be used by the United States Naval Consulting Board in aiding in effecting an industrial organization necessary to the plans for national defense. The information contained in these blanks is not to be used in any way to affect the business of the concern reporting, or for comparison with any other report of any kind previously filed by it. The value of this patriotic work can best be insured by making this report complete in every detail. We must deal with the problems of an adequate national defense as we deal with the problems of our everyday business life. We must face facts — not theories. We must do now, in time of peace, quietly, efficiently and thoroughly, those things which all know must be done to achieve true industrial preparedness, and which if postponed until an outbreak of hostilities must result in tremendous losses in lives and money."

The form properly filled out will give:

1 The names, ages and citizenship of the officers and directors, or owners, the valuations of the plant, representatives abroad, and banking connections.

2 Location and description of plant, its possibilities for expansion, its fire protection, and telegraph and telephone facilities of the plant.

3 Mechanical equipment, character, quantities and sources of raw materials, character and volume of products, proportion shipped abroad, whether munitions of war have ever been produced, the quantities and ultimate possible capacity.

4 Character of labor — union or open shop — total employees and citizenship — if women are employed and the possibilities for further employment of women.

5 Transportation facilities — rail and water.

6 Under the heading "Possible future arrangements" appears the following which is quoted in full.

POSSIBLE FUTURE ARRANGEMENTS

1 Would consider bidding upon regular U. S. Army and Navy contracts in time of peace?..

2 Would consider accepting regular U. S. Army and Navy contracts in time of war? ....

3 Would consider accepting "Minimum educational order" (see *Clause A* below)? ....

4 Would consider accepting payment in accordance *Clause B*? ....

5 Would consider constructing jigs and tools in accordance *Clause C*?.......

(If not applicable to this plant, write word "None")

6 Would consider enrolling skilled labor in "Industrial Reserve" (see *Clause D*)?..

7 Would consider agreements to limit profits in time of war in accordance with governmental regulations (on basis of cost plus a reasonable profit)? ....

8 Would consider agreements to restore existing labor agreements at close of war? ....

9 Would consider inserting clause in all civil contracts making them contingent upon governmental needs in time of war?........................

Clause A Minimum order for annual production will be accepted with the understanding that such order will be restricted to that product for which the manufacturer's equipment is best fitted. Also, that such order shall be for only such a quantity of product as will insure familiarity with the work upon the part of the manufacturer's organization. The manufacturer agrees that this minimum annual educational order shall be put through the factory in regular course and in such manner that foremen and those holding positions of responsibility shall become familiar with the peculiarities incident to the manufacture of these goods. In time of war the manufacturer will be expected to concentrate upon this same product and it is essential, therefore, that his entire organization, including purchasing, manufacturing, inspection, shipping, engineering, cost keeping and administrative departments, be made familiar with the work. Minimum orders will not be of sufficient quantities to interfere with manufacturer's regular production.

Clause B Payments for "Minimum annual orders" covered in *Clause A* shall be made upon the basis of the actual cost of production, inclusive of all special tools, jigs, etc., plus a reasonable profit.

Clause C The manufacturer will agree to make and preserve one set of special jigs, tools, gages or fittings necessary for the production of these goods, and corrected drawings shall be kept on file in the engineering department of the plant covering such special jigs, tools, gages or fittings. In short, the engineering or designing department shall maintain at all times corrected drawings from which the shop may, upon short notice, construct the necessary equipment for quantity production.

Clause D In war as now waged the industrial force has become quite as important as the fighting army. Skilled mechanics in all lines of production work must be kept from enlistment in the regular Army and must be retained in the factories, mills and mines for the production of munitions. It is essential, therefore, that the names of these skilled workmen be listed and that the men themselves be enrolled in the Industrial Reserve. A button or other distinguishing mark will be supplied by the government in the event of war to skilled workmen enrolled in the Industrial Reserve, and such enrollment will be considered to carry with it honors equal to enrollment in the fighting army. A government card will be issued to each man enlisted.

7 A summary by classes of tools, types of machines, etc., and their capacity.

THE FACTS COLLATED

With the facts in this census properly collated we shall learn where the nation is weak and where strong. Not only will the government be instantly enabled to determine where munitions can be obtained, but how rapidly. We shall also know from this inventory:

1 Whether America is independent of foreign countries for raw or manufactured war material, and if not, wherein we are lacking and to what degree.

2 In what kind of munitions we have ample manufacturing facilities and wherein we must provide means for making up the deficiency.

3 Whether it be prudent for the government to build new arsenals and other works in the central part of the United States.

TOOLS, GAGES, JIGS AND TEMPLATES

This inventory will probably show that many shops well equipped to manufacture ammunition will not employ the skilled machinist to make the required tools, jigs, gages and templates. Possibly then it will transpire that the most efficient plan would be for a government factory to make standard gages, jigs and templates and loan or sell them to certain shops for use in the manufacture of shells, rifles, etc.

MACHINE TOOLS

This work of industrial preparedness should serve to inspire our machine-tool builders to perfect and improve their product and perhaps to standardize their machine tools to the end that those who must needs purchase them would find an adequate and immediate supply available in case of emergency.

STANDARD BOOKS ON MANUFACTURE OF AMMUNITION

The various engineering societies contain in their membership men who could prepare most valuable papers on the subject of ammunition manufacture. For the present there may be many secret processes which for the time being may not be forthcoming in the way of books and papers, which, however, might be prepared in advance and held until the hour of need, when through patriotic motives they could naturally be published for the benefit of the nation at large. The old saying that "Necessity is the mother of invention" is now applicable to the nations at war and it is reasonable to assume that they have discovered superior methods, processes and tools for the rapid and economical manufacture of shells, etc. The best method of overtaking such a lead would be by a combination of talent and experience such as might be brought about by a congress of ammunition manufacturers, assembled for the purpose of exchanging experiences, etc. The best way for the manufacture of each element should then be arrived at and the widest publicity should be given to descriptions of such methods.

FUELS FOR INTERNAL-COMBUSTION ENGINES

Wholly aside from the needs of war, there is a growing demand for liquid fuels for automobile and aeroplane motors. An interesting and profitable inquiry might be directed towards securing other fuels than gasoline, such as alcohol from sawdust and other wood wastes, sugar-house waste and cornstalks. Here the chemist will find ample fields for investigation.

GOOD ROADS AND BRIDGES

These State Directors will be an influence in every state by urging the military value of good roads especially along and near tidewater, and they should not forget that the bridges should be made strong enough for heavy automobile trucks and extra heavy cannon.

SETTING AN EXAMPLE FOR BETTER CITIZENSHIP

Such an army of engineer census takers actively at work cannot fail to breed a better spirit of citizenship. Working without compensation they will set an example of patriotism which is bound to be felt. This spirit of patriotism will be transmitted to the workingman producing war supplies. These workers are surely a part of the defensive force of the country and should receive adequate recognition as such.

Ex-Senator Elihu Root recently said, "Eternal vigilance is the price of liberty. The principles of American liberty today stand in need of a renewed devotion on the part of the American people. ....I want to see in my country the spirit that beat in the breasts of the men at Concord Bridge — who were just and God-fearing people but who were ready to fight for their liberty, and if the hundred million people of America have the spirit and it is made manifest, they won't have to fight."

DISCUSSION[1]

CHARLES WHITING BAKER, who presented the paper in the absence of the author, discussed briefly the history of the last half-century in connection with the movement for preparedness. He cited as an example of the interest manifested by engineers in preparedness the courses of lectures on military engineering given in various cities throughout the country in the early months of 1916.

Regarding the industrial census, he thought that there were so many other important things regarding which the advice of engineers was needed, that there was no reason why they should undertake the detail work which the Government itself, when shown, should undertake by its own paid agents. The best service that a Board of Engineers could do for the country would be to take care of the big things, and let the little things be done by somebody else.

FRANK B. GILBRETH thought that the problem of preparedness was not one of merely preparing for war; it was also one of preparing for efficient living at any time. It was a problem of national elimination of unnecessary waste, of adequate general and individual effort toward utilization and conservation of our great resources. Just as in education it was psychologically sound to develop general abilities to a certain extent in order that special abilities might later reach their greatest height, so in preparedness it was most efficient to develop general preparedness in order that special preparedness for an emergency might be most quickly and satisfactorily developed.

Efficient preparedness meant the preparedness of every man, woman, and child in the country, beginning at the earliest age available. It was not enough that the equipments and tools of the sudden need be made available. The methods of handling them must be taught in such a manner that the workers might gain not only a knowledge of the machine that made the machines, but an idea of motion economy and of personal effectiveness.

What should shock us was not what might happen to us in a war, but what was happening right now, while we had general peace. National waste, lack of general and specific education, lack of easy means for obtaining the fullest education in the vocation for the worker in that vocation, lack of correlation!

[1] This discussion which is here published in abstracted form will be found in full in THE JOURNAL for JUNE, 1916.

Ernest H. Peabody, in a written discussion, called attention to the awakening, in this country, to new conditions which make invasion by a foreign power possible, and indorsed the author's question, "What May We Do to Serve Our Country?" He thought, however, that the author answered this question with a complacency which might lead others to believe that the work done by the Naval Consulting Board and the engineering societies was entirely sufficient, whereas the contrary was the case, and it was a mistake to presuppose that the enormous work of organizing for industrial preparedness could be effectively projected and completed, with the dispatch which the circumstances made imperative, by the engineering fraternity of this country as represented by the engineering societies working in the odd moments snatched from the task of making a living, without compensation, and paying its own expenses.

While it was a satisfaction to all to know that the important movement of industrial preparedness had been initiated, and was being pursued to good purpose by the engineers of the country acting in their private capacity, there was not time to carry out the work to completion in this way. Only the Government could do this work, and the whole question of preparedness should be made a Government policy, it being the duty of the engineering societies to stimulate public opinion for the purpose of inducing the Government at Washington to establish the necessary department.

Besides the industrial census, there should be a census of men, and every man in the country should be listed in it. The Government should find out how each man could best serve the country in case of hostilities, — whether in active service with the colors or in the equally important field of supplies; and the men should not only be selected, but each should be assigned to his job and should be preparing to take up the work when the call came.

Oberlin Smith said that in his opinion military and naval preparation, and the enormous expenditures involved, should go on with the utmost speed and to such an extent that there would be no danger of any nation attacking us. Preparedness was simply a matter of insurance.

Our tools for munitions and other military supplies should be standardized much more than they had been, so that if ever the time should come when our Government had to set all its industries to work, and do it by government consent and under government protection and supervision, the matter would be easy for all concerned, and there could be almost immediate production. Many little machine shops all over the country might have proper gages and standards supplied by a bureau of the Government. Then parts could be made by the thousands and assembled in a few hours.

The Naval Consulting Board was a decided step forward, and he did not see why the army should not have a similar board and the government personnel be brought together into one Commission to handle all matters pertaining to munitions under the general direction of experienced generals and admirals.

One very important class of machine tools not mentioned in the paper was the presses with which brass cartridge cases were made. These presses were almost exactly the same as had been made for years for all sorts of drawing work, and would be necessary if a great many munitions were needed for our own Government. Many of them doubtless would be found in existence when needed and could be drawn upon.

John H. Barr, in a written discussion, stated that the uniform standard vital in producing guns and ammunition required a corresponding uniformity in the quality of the gages and tools for producing these. Such gages and tools should be constructed in specially equipped government tool factories controlled by a highly expert staff and manned by workmen of corresponding skill. This would develop a large corps of mechanics skilled not only in the making of these tools but very familiar with their use and the requirements to be met by their output. They would be a source from which to draw inspectors and instructors for duty in the private shops.

James A. Campbell pointed out that members employing mechanics in large numbers could get signed statements from those willing in time of war to take positions in shell factories or wherever machinists, etc., would be required. These mechanics, he wrote, could be classified so that firms requiring men could see from the lists the experience and class of men available.

F. O. Hoagland wrote that the shortage of arms and ammunition in this country, in case of immediate need, was very apparent. As a precaution he believed that orders for arms and ammunition of sufficient size to keep a small unit plant at work constantly should be placed with several manufacturers without delay, in order to maintain equipment in good repair and to have a number of trained workmen to form a nucleus for building up an organization in case of need.

It took about twelve months to prepare the special equipment necessary for the production of, say, 100 military rifles per working day, and about half that time to prepare for 100,000 cartridges per day, even when a factory had at the start a fair organization for similar work; and a good many months additional might be spent before manufacturing difficulties were overcome and a product turned out that would pass inspection.

H. V. Haight suggested that in letting "minimum annual order contracts" in time of peace, (1) the Government should be the general contractor and let only sub-contracts. (2) Shops should be encouraged to make their own equipment, which might include even machine tools. In this way the equipment could be rapidly increased in time of war. (3) A provision that manufacturers should visit each other's plants should be incorporated in government contracts. (4) The cost-plus-a-fixed-sum basis of payment was open to very grave objections in time of war, the most serious of which was that it would not give adequate incentive for low costs or large production.

Frank O. Wells and Charles E. Smart expressed their belief that the government should own all of the drawings, books of instructions, gages, jigs and fixtures, in fact all the special equipment necessary to produce munitions of war, and that these should be kept in the vaults in all the arsenals and navy yards ready to be distributed to the various plants at short notice. Operations and methods of handling the work, they wrote, should be standardized, and the present arsenals used more as experimental stations from which the private plants could get their instructions and information. There should be a number of private plants which should be given small yearly orders for parts which they were best adapted to produce, so as to keep their organizations familiarized with the product.

John Younger wrote calling attention to the tremendous importance of good roads in time of war. He believed the organization of our transport demanded the attention of our engineering bodies just as much as did the mobilization of our industrial resources. The standardization of our road systems, the standardization of the controls on our locomotives and motor trucks, so that men could change from one to another with all confidence, the education and disciplining of numbers of men to the principles of transportation, were all matters that could only be handled by engineering executives.

BERNARD M. FINE suggested in a written communication that the Society appoint a Sub-Committee on Ordnance Manufacturing, which should secure and present papers not only on modified standards but on methods of supervision, jigs, design, etc., and anything that would facilitate manufacture in time of need, and should even go so far as to organize a volunteer force of trained engineers to act as supervisors and inspectors.

ALFRED L. P. DENNIS[1] outlined in writing a plan providing for the formation of a Council for National Defense, whose membership should include experts in special fields as well as military and naval officers.

ARTHUR G. MCKEE thought that the biggest problem now confronting us and one that would have to be solved in case of war, or a threatened war, was the establishment of an organization which could coördinate, systematize, and regulate the manufacture of the different items entering into the schedule of preparation.

HARRY E. HARRIS thought it not only necessary to enlist the engineers and manufacturers, but also the loyal support of every skilled workman. It was the duty of employers who were patriotically inclined to reach their men by individual talks, thus obtaining their views and learning of their willingness to enlist as members of the Industrial Reserve if called upon. As an example of what might be done, he gave an account of a meeting of the workmen of the company with which he was connected.

D. ROBERT YARNALL said that he would like to see a committee appointed to coöperate with other institutions engaged in the study of and the elimination of the causes of war. Industrial preparedness and the preparation of an industrial inventory might be desirable steps, but along with them he would like to see the Society enter into this even greater constructive work.

HENRY A. WISE WOOD[2] wrote in advocacy of the development of the shipbuilding facilities of the nation and of the merchant marine

[1] Professor of History, University of Wisconsin, Madison, Wis.

[2] 25 Madison Ave., New York City.

as important elements in Industrial Preparedness. Our naval preparedness, he said, should include the upbuilding of such a merchant marine, not only that we might have it for use in war, but that in times of peace we might gather into our own hands our own overseas carriage of goods.

HAROLD V. COES thought that the Government should construct in its own factory the necessary jigs, templates, gages, and fixtures required and adaptable to special or standard machine-tool equipment, and rent, loan, or sell them to manufacturers equipped to manufacture shells, rifles and other war material.

H. L. GANTT said that it had become perfectly clear that the principles underlying industrial and military efficiency were the same, and that if a nation was to be efficient in a military sense it must first be efficient industrially.

In the present European war, when the supreme test came Germany was found to be a nation of people who, in general, knew what to do and how to do it; while the industries of England were, in too many cases, controlled by people who understood only their commercial side.

We, following the footsteps of England, had regarded financial strength as the most important strength; but in the new world which was being ushered in by the great struggle now taking place the engineer was destined to be the supreme power, for it was becoming increasingly clear that, in the future, the man who *owned things* would not be as important a factor in the world as the man *who could do things.*

WM. B. JACKSON thought that the sooner the idea that the Government should own its own manufacturing establishments was rooted out the better. Were the United States to be attacked by a formidable foe, it would require not only the most intense activity of any reasonable development that the Government might have in the line of manufacture of munitions and supplies, but also the most intensive activity of every private manufacturing establishment that could be pressed into the service.

FRED E. ROGERS wrote that he believed comprehensive plans should be prepared for the manufacture of shrapnel, high-explosive shells, cartridge cases, rifles, field guns and all the varied equipment of war. Every step of the operations should be laid down, specifying in each instance the tool and the machine on which the work should be done and the limits of accuracy required, including an estimate of the possible production. It would be a great undertaking, but how much better it would be to spend a few million dollars working out plans to provide means for industrial mobilization and rapid production than to run the risk of a complete smash-up in time of stress.

K. A. JUTHE thought that the author's suggestion for a central plant wherein tools and gages could be manufactured and sent out to different plants was one to which engineers should give careful consideration. No manufacturing, he wrote, could be started until gages had first been made. The manufacture of the gages should, however, be carried out in one plant laid out for that particular purpose and under the supervision of one head, as it would be necessary that all gages bear the same relative characteristics.

The proposition of toolmaking under central supervision, however, both as regarded small tools, jigs and fixtures, was an entirely different one. Jigs and fixtures were largely made by individual concerns to suit their own particular machine tools for rapid interchangeable production. As practically no two tool-manufacturing plants had the same facilities, it was practically impossible to standardize tools for making the jigs and fixtures, and the latter would be of no value unless they were interchangeable under the majority of conditions existing.

PERCY E. BARBOUR[1] commented on the prodigious waste of the less common and semi-rare metals in the war. Brass for cartridge cases must necessarily be composed of the best brands of copper and zinc because their drawing was the severest test to which such metal could be put. But specifying high-grade materials for everything required in military work was wholly unnecessary.

For example, the British had used a brass plug for closing the opening in the nose of the fuse temporarily between the time when a shell was completed ready for shipment and the time of its fusing prior to the action in which it was required. Study, however, had shown that wood plugs could be used to equal advantage and at an enormous saving. Similarly, Russian specifications had required that all projectiles be nickel plated.

As an effective means of reducing these wastes he thought speci-

[1] Managing Editor, Engineering and Mining Journal, New York City.

fications should be referred to the U. S. Naval Consulting Board, representing the engineering societies, whose specialists would scrutinize them and suggest substitutes and improvements.

L. P. Alford wrote that from the experience of the allied nations in purchasing machine tools it seemed justifiable to lay down principles for the standardization and procurement of machine tools in organizing for American industrial preparedness. The first of these were:

1 Organize at once, in skeleton form, an industrial committee to control the standardization, design and preparation of machine tools for the production of American munitions.
2 Through joint action of this committee, The American Society of Mechanical Engineers, and the National Machine Tool Builders' Association, standardize the details of regular machine tools and design whatever additional special machine tools may be necessary for the rapid and economical production of American munitions.
3 Immediately on the outbreak of war, prohibit the exportation of any machine tools from the United States, and also their importation except under license and control of the committee mentioned under 1.
4 Order all machines abroad through this committee or its representatives in the capitals of Europe, and entrust these men with the responsibility of securing the desired deliveries and quality.
5 Order no machine tools abroad except to standardized American designs, either for the complete machine or the essential details as the committee may determine.

As to the standardization of machine tools, he believed that the Society should initiate and prosecute the work.

Chester B. Hamilton, Jr., wrote strongly advising that one or more efficient and up-to-date sample plants be maintained by the government for each of the principal classes of munitions and be worked actively under an approximation to war conditions. The size should be the minimum economical unit for the machinery involved. At the start he and other Canadian munition manufacturers had no idea at just what speeds the different operations could or should be performed, and it took months of work to find out. He also emphasized the importance of providing in advance the skeleton of an inspection organization.

HARRINGTON EMERSON told of methods successfully employed in recruiting workers for a certain new munitions plant located in a non-manufacturing section of the South, drawing the conclusion therefrom that there was abundant opportunity to find in America the raw material to produce efficient workmen provided the search was undertaken in the right way.

RALPH E. FLANDERS said that in Canada the munition manufacturers at first had great difficulty in getting enough gages, the rapidity of wear proving much more severe than expected. Another difficulty lay in the lack of coördination in the furnishing of materials and supplies.

Canadian experience indicated the impracticability of standardizing methods, at least so far as the making of shells was concerned. The demand was so tremendous that the problem could not be approached from the ideal standpoint. Every imaginable kind of tool had to be used. For instance, the finishing of the inside of a shell might be done on the engine lathe, the hand turret lathe, the automatic lathe, the drill press, or on special machines made for the purpose. It would be folly to confine the doing of this work to any one machine and leave the resources of the other machines untouched. He was inclined to think that too much haste in the matter of standardizing too closely the processes of shell making would greatly cut down their rate of production.

C. S. WILLIAMSON, JR., said that although nitric acid was absolutely indispensable in the manufacture of modern explosives, we nevertheless depended on nitrates imported from Chile for our supply. To produce nitric acid from the atmosphere required cheap electric power. As regarded water power for this purpose, in Alabama alone there was 1,000,000 h.p. capable of development. He also enumerated the resources of the South in the way of fuels, naval stores, etc.

ADOLPH L. DE LEEUW recommended that in addition to the census of factories, materials, machines, men, executives, etc., there should also be taken a census or list tabulating the conditions that prevented the speedy and easy manufacture of those materials and products which were wanted; citing for illustration defects generally found in Government specifications in the matter of allowances and limitations.

MAJOR W. GOFF CAPLES[1] said that the nation would need the coöperation of every one, from the manufacturer, the engineer and the chemist clear through to the laborers in the mills, because wars now were fought not by armies but by nations. The proposed industrial census was a needed step in the right direction, for it would show us some of our weaknesses and give us a better idea how to remedy our defects.

FRED J. MILLER wrote that in 1913 only three nations spent more money for army purposes than the United States. Yet we were told by people who should know that our army was a negligible factor. Engineers had always paid far more attention to efficient organization than had any others, and it seemed to him that they might render their best possible service by suggesting such improvements in organization as would result in getting more for the money and effort expended than was at present secured.

ELMER H. NEFF believed that the United States Government needed a manufacturing department headed by a secretary who should be a cabinet officer. With a Secretary for Manufacture and civilian experts (not army or navy officers) of the quality that private firms would employ for similar work, placed in charge of the great manufacturing plants which the Government owned, a long step towards efficiency, economy, and preparedness would have been taken. He strongly urged the immediate operation of the rifle plants at Springfield and Rock Island, for it was an easy matter to obtain men to defend the country, but very difficult to provide rifles and ammunition in quantities needed in case of war, without great delay.

J. H. BROPHY[2] wrote that he believed it would take years of pounding away at the facts to enlighten people on this great undertaking of Industrial Preparedness. An educational campaign seemed absolutely necessary, and the newspapers were the best channels through which to reach the people. To utilize the industrial resources of this country to the very best advantage, there must be a complete unification of all who participitated. This could never be

[1] Queen and Crescent Bldg., New Orleans, La.
[2] Cleveland Automatic Machine Co., Cleveland, O.

accomplished by the aid of a few individuals without the assistance of the Government.

COMMANDER H. T. WRIGHT[1] said that in the South instead of getting materials it was rather a question of securing men educated in the trades who were able to step into the shops and perform the work necessary to the preparation of war materials. The South had been very lax in its training of men for trades, and it would take a long time to build up that force of workingmen who must be the backbone of any industrial preparedness no matter how many raw materials were available or how many machine shops were lying idle.

LEWIS A. RILEY, 2D, called attention to the possibilities of obtaining an independent nitrate supply from the bituminous fuel resources of this country. The catalytic oxidation of coal-gas ammonia to nitric acid was being commercially operated in Germany, and there was available a process of treating bituminous fuel known as the by-product producer gas system, by which 70 per cent of the available nitrogen in the fuel could be converted into ammonia in such a way that the products of the plant would consist of a useful fuel gas, ammonia, and tar only. Such plants would produce not less than four times the amount of valuable ammonia per ton of fuel that could be recovered from a corresponding by-product coke plant.

WILLIAM KENT wrote suggesting the establishment of a General Industrial Staff to perform the same service for the manufacturing and transportation industries of the country that the Agricultural Department did for the farmers. He thought information should be gathered from which to prepare a card index of every engineer in the United States who was available for service in the Army or Navy or in any department of industry that could in case of need supply munitions, clothing, motor cars, aeroplanes or anything else that might be required by the Government. We should also have a Board of Inspection to review the specifications for every part of our munitions and make requirements which were reasonable and capable of being met by the available tools and manufacturing methods.

GEO. R. HENDERSON wrote that there were many members acting in a consulting capacity whose technical services would probably be of greater advantage to the Government than would

[1] U. S. Naval Station, New Orleans, La.

their digging trenches or shouldering rifles, and he believed that they should be listed, with their professional records, so that the very best use might be made of their services in the navy yards, arsenals and other plants where mechanical skill was of the highest importance.

THE AUTHOR said that the discussion brought out proved a statement frequently made, viz.: "The discussion is likely to be the most valuable part of a paper;" and he wished to express his thanks to those who in giving their facts and suggestions have added real and enduring value to his brief paper.

No. 1533

CAPACITY AND ECONOMY OF MULTIPLE EVAPORATORS

By E. W. Kerr, Baton Rouge, La.

Member of the Society

During the last few years, the author has carried on a large number of investigations of both the capacity and economy of multiple evaporators and vacuum pans. This work has been done in the laboratory on a small apparatus, and in sugar factories on apparatus of commercial size and under regular operating conditions. The results of much of the laboratory work were given in a former paper before this Society,[1] and the object of the present paper is to give results of experiments made since the first paper was written, together with certain general observations.

2 The process of evaporating in sugar factory work consists of two distinct parts: *First*, the juice with a density of 12 deg. to 18 deg. Brix[2] is concentrated by evaporating off part of its water to approximately 55 deg. Brix in a multiple evaporator. *Second*, the syrup produced is then taken to a vacuum pan working single-effect, and where further water is removed by evaporation under conditions suitable for crystallizing the sugar, the final density being in the neighborhood of 95 deg. Brix.

3 This paper will consider the first part of the process, namely, the evaporation in multiple effect. Fig. 1 shows diagrammatically the general arrangement of a quadruple effect. Steam is admitted to the first body and is condensed, and the latent heat is transmitted in succession through the heating surface of the four bodies, the boiling temperature in each body being lower than in the preceding body.

[1] Tests upon the Transmission of Heat in Vacuum Evaporators, Trans. Am.Soc.M.E., vol. 35, p. 731.

[2] The Brix saccharometer or hydrometer indicates directly the percentage of sugar dissolved.

Presented at the Spring Meeting, New Orleans, La., April, 1916, of The American Society of Mechanical Engineers.

4 The thin juice enters the first body, where a portion of the water is removed. The juice then passes in succession to the other bodies, in each of which a portion of the water is evaporated, and leaves the last body at a density which is hand-regulated by means of valves in the liquor lines entering the different bodies. The juice entering each body, except the first, is hotter than the boiling temperature in that body, consequently some heat is delivered by the liquid as well as the steam. Whether heat enters the first body in the liquid or not depends upon the temperature of the entering juice.

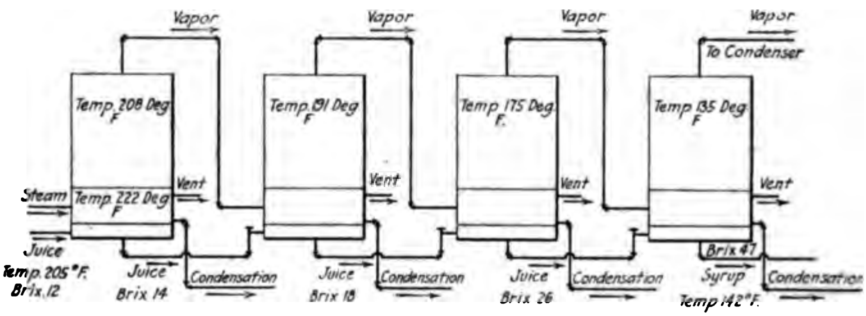

FIG. 1 QUADRUPLE EFFECT FOR SUGAR JUICES. TEMPERATURES AND DENSITIES OBTAINED FROM TEST

CAPACITY

5 The capacity of evaporators depends on two general factors: *Coefficient of heat transmission* and *temperature fall.*

6 The factors affecting the coefficient of heat transmission, discussed in the first paper,[1] may be enumerated as

1 *a* Steam distribution; *b* Steam velocity
2 Density of heating steam
3 Quality of heating steam (whether superheated or not)
4 Presence of air or other incondensable gases in the steam, which may be expressed mathematically by the ratio $P_s \div P_t$
5 Presence of condensation on the heating surface
6 Cleanliness of heating surface (steam side)
7 Material, thickness and molecular structure of tubes

[1] Tests upon the Transmission of Heat in Vacuum Evaporators, Trans. Am.Soc.M.E., vol. 35, p. 731.

8 Cleanliness of heating surface (liquor side)
9 Velocity of liquor circulation

7 The temperature fall depends on the following factors:

10 Pressure of steam supplied to first body
11 Vacuum in last body
12 Hydrostatic head (height of boiling)
13 Presence of air or other incondensable gases in the steam (see item 4 above)
14 Density of liquor being boiled
15 Purity of liquor being boiled.

8 The heat transmitted from steam through a metal wall to a liquid being boiled takes place in three steps: *a* From the steam to the initial surface of the metal; *b* From the initial surface through the metal wall to the cooler surface; *c* From the cooler surface to the liquid being boiled. Items 1 to 6 affect *a*; item 7 affects *b*; items 8 and 9 affect *c*, and items 10 to 15 affect temperature fall.

9 Although in many respects evaporators are similar to surface condensers, there are certain fundamental differences. In surface condensers the density of the heating steam varies between very small limits, corresponding to vacuums of say 24 to 29 in., whereas in evaporators the steam pressure varies from about 5 lb. per sq. in. gage in the first body to a vacuum of about 15 in. in the last body, with corresponding variations in density. In surface condensers air is the only incondensable gas, and it enters with the steam, also by leakage through joints and metal pores; in evaporators, other incondensable gases, such as ammonia, sulphur fumes, etc., entering with the juice, have to be contended with. Evaporator shells also furnish larger areas in contact with the atmosphere per unit of capacity than do surface condensers, thus increasing the danger of air leakage. In surface condensers the danger of scale or other fouling of tubes is confined mainly to the steam side, the high velocity of the cooling water preventing any fouling on the other, even with bad water; in evaporators, however, the larger part of the fouling is on the liquor side. The tendency to fouling depends on the condition of the juice as to density, materials used in clarification, etc. In surface condensers the velocity of the cooling water is produced by a pump or other mechanical means, whereas in evaporators the circulation is produced by convection and steam currents. In surface condensers the factors hydrostatic head, density of liquid and purity of liquid do not enter, whereas they are important in connection with evaporators.

10 The effect of superheat in steam on the coefficient of heat transmission is not very well known, the results of different experimenters being more or less contradictory. The author's experiments seem to show that superheat does not affect the coefficient of heat transmission, and results by other recent experimenters substantiate this. Operators frequently state that greater capacity can be obtained with exhaust steam than with live steam. Most evaporators are arranged to use live steam in case the exhaust steam supply is insufficient; boiler steam is throttled down to the necessary low pressure, resulting in superheating the steam, and the importance of the matter lies in this. Table 1 gives data obtained by the author in experiments on a laboratory apparatus, where the amount of superheat was varied by injecting a spray of water into the steam supply pipe to the calandria. A curve plotted with coefficients of heat transmission as ordinates and with degrees superheat as abscissae is practically a horizontal straight line.

11 In another series of tests on the laboratory evaporator, exhaust steam from a steam turbine and live steam direct from the boiler were used in alternate tests. The results corroborate the conclusions derived from Table 1, that the coefficient of heat transmission is not less for throttled live steam (superheated) than for exhaust (moist) steam. In all probability, the apparent reduction in capacity noticed with the use of live steam is due to the live-steam pipe being too small to supply steam to the evaporator at a pressure equal to that in the exhaust main.

12 The presence of air and other incondensable gases is doubtless one of the most fruitful causes of low heat transmission. It is probable also that less is known about this factor than any other involved. Air-free steam is practically impossible, and, as stated, the conditions in this respect are less favorable in evaporators than in surface condensers.

13 As the result of a large number of experiments on surface condensers, George A. Orrok states that the heat-transmission coefficient varies according to the expression $(P_s \div P_t)^2$, in which P_s represents the partial steam pressure and P_t the total pressure of steam and air combined. Experiments by the author on laboratory evaporators seem to show that the exponent in the above expression lies somewhere between 3 and 4.[1]

[1] Tests upon the Transmission of Heat in Vacuum Evaporators, Trans. Am.Soc.M.E., vol. 35, p. 731.

14 A considerable amount of experimenting on this subject has been done recently by the author. Temperatures were measured at different portions of a laboratory calandria (Fig. 2), on the basis

TABLE 1 EFFECT OF SUPERHEAT IN STEAM ON COEFFICIENT OF HEAT TRANSMISSION

DATA OBTAINED IN TESTS ON A LABORATORY EVAPORATOR

No.	Abs. Pressure in Calandria, In. Mercury	Temperature Fall	Head	Temperature Steam Pipe	Superheat, Deg. Fahr.	Coefficient of Heat Transmission
Calandria *A* (Without Downtake)						
1	30	16.75	16	216.5	4.5	529
2	30	16.7	16	217.0	5.0	554
3	30.1	16.75	16	220.1	8.1	514
4	30	16.7	16	227.1	15.1	506
5	30	16.75	16	239.5	27.5	543
6	....	16.5	16	239.7	27.7	507
Calandria *B* (With Downtake)						
1	30.06	17.2	16	215.9	3.9	855
2	30.0	16.7	16	216.0	4.0	790
3	30.04	16.7	16	217.2	5.2	923
4	30.0	16.5	16	225.1	13.1	844
5	30.1	16.7	16	233.2	21.2	906
6	30.0	16.78	16	243.7	31.7	696
Calandria *B* (With Downtake)						
1	28.1	17.38	12	235	23.9	878
2	29.42	19.67	12	214	4.3	717
3	29.15	18.73	12	215.1	36.9	832
4	28.8	18.46	12	212.5	4.5	395
5	29.77	20.5	12	215.4	5.7	926
6	28.65	18.75	12	226.6	16.2	887
7	28.5	17.7	12	245	34.3	865

Calandrias *A* and *B* were used with the laboratory apparatus. See Figs. 3 and 4, Tests upon the Transmission of Heat in Vacuum Evaporators, Trans. Am.Soc.M.E., vol. 35, p. 731.

that the greater the amount of air in any locality of the steam compartment, the lower the temperature there. Typical results are given in Table 2. The thermometers were inserted through the shell at points indicated in the table, and were made to extend a few inches into the steam compartment. The figures show very clearly that there was considerable variation of temperature in different parts of the calandria, and also a shifting of temperatures. It will be noted that some temperatures were found higher than the saturation temperature and some lower; the former were due to superheat and the latter to the presence of air. These data seem to show that there are air pockets, changing from place to place. Also, the last

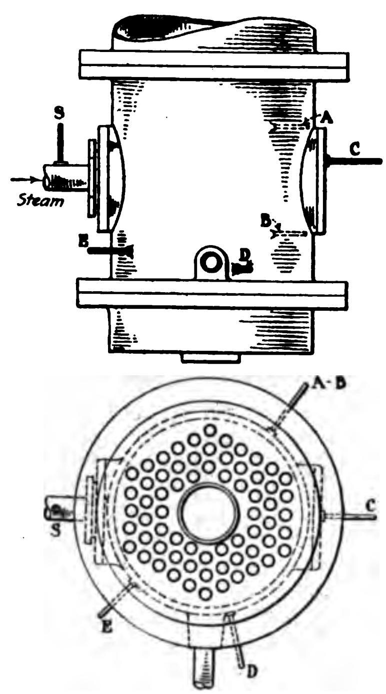

FIG. 2 LOCATION OF THERMOMETERS

two sets of readings show practically the same temperatures at all points and it is interesting to note that when these latter readings were taken, the evaporator was boiling at a high rate, the velocity of steam being much greater than with the previous observations.

15 The prevention of these air pockets is a matter for the consideration of designers. It seems reasonable to expect that air pockets will form in dead spaces, i.e., where there is little or no movement of steam, and for this reason good distribution is very

TABLE 2 TEMPERATURES IN DIFFERENT PORTIONS OF A CALANDRIA

FOR LOCATION OF THERMOMETERS, SEE FIG. 2

No.	Abs. Pres. in Calandria, In. Merc.	Saturation Temp. Due to Pressure in Calandria	Temperature at (Fig. 2)						Valve A Turns[1] Open	Valve B Turns[2] Open	Remarks
			S	A	B	C	D	E			
1	22.4	197.75	199.0	155.0	164.0	163.0	158.0	....	1½	0	No air injec.
2	12.27	170.75	178.0	164.0	164.0	164.0	160.5	170.0	3	0	No air injec.
3	22.14	197.2	199.0	179.0	177.0	180.0	158.0	170.0	1½	0	No air injec.
4	19.64	191.5	194.7	175.0	175.0	176.5	175.0	169.0	1½	¼	No air injec.
5	22.90	198.8	200.0	170.0	167.0	178.0	173.0	170.0	1½	0	No air injec.
6	23.9	200.9	202.0	168.0	168.0	175.0	174.0	194.0	1½	0	Air injec.
7	23.9	200.9	202.0	167.0	169.0	177.0	164.0	194.0	1½	0	Air injec.
8	20.24	192.9	197.2	176.5	176.0	178.0	179.5	173.5	1½	¼	Air injec.
9	19.5	191.17	....	179.0	175.0	180.0	180.0	174.0	1½	¼	Air injec. No water
10	20	192.4	202.0	191.0	182.0	185.0	185.0	180.0	2	¼	No air. No water
11	21	194.7	203.0	192.0	183.0	194.0	182.0	182.0	2	¼	No air. No water
12	19.34	190	204.0	190.0	191.0	191.0	191.0	191.0	3	¼	No air. No water
13	20	192.4	203.5	190.5	191.0	191.5	192.0	192.5	3	¼	No air. No water

[1] A = 1-in. valve in condensation drain. [2] B = ¾-in. valve for venting.

important. High velocities produced by means of baffles, etc., should result in the prevention of air pockets. In practice some of the air is removed by means of the condensation pumps which draw from the bottom of the steam compartment, and some is vented from the top through small openings connected either to the next body or direct to the condenser. This statement applies particularly to the more common types, — vertical juice-tube evaporators and horizontal steam-tube evaporators. Other special types make use of various devices, in some cases each tube having its individual vent. In some types the vents can be controlled, while in others they cannot. One of the great difficulties is the impossibility of removing the incondensable gases without removing steam along with them and reducing the economy. In addition to its insulating effect, air in the steam reduces heat transmission by causing a temperature fall lower than the apparent fall. This is because the temperature of the heating steam is lower than the saturation temperature corresponding to the pressure.

16 The loss in capacity due to the presence of condensation on the heating tubes depends upon the type and design of evaporator. The bottom portions of long vertical tubes will be affected more than those of shorter ones. In vertical juice-tube evaporators, the water, after reaching the lower tube sheet, must zigzag amongst the tubes to reach the outlet. Tube sheets are inclined by some makers, to facilitate the removal of condensation. Occasionally the tubes themselves are inclined so that the condensation runs along the underside of the tube in attenuated form to the bottom, the idea being to reduce the amount of surface in contact with the water. The removal of water is also facilitated by using multiple outlets, reducing the distance the water has to travel along the bottom tube sheet. In horizontal steam-tube evaporators, the condensation gravitates to the lower side of the tube and is blown through by the steam.

17 As stated, the fouling of the heating surface on the steam side is mainly due to oil in the exhaust steam coming from the engines. An oil separator in the supply pipe to the evaporator aids in overcoming this difficulty, and when steam turbines are used as prime movers, the difficulty is avoided. In vertical juice-tube evaporators there is little opportunity for cleaning the oil scale from tubes as the latter are difficult of access. A very efficient means of cleaning them is to fill the steam compartment with molasses during the off-season; the acid in the molasses attacks the scale and removes it. Steam-tube evaporators have the advantage that me-

chanical cleaning by means of swabs is possible. This can be done with the tubes in place. Tubes in this type are usually removable, however, and this facilitates cleaning.

18 Evaporator tubes are generally made of copper or brass, more often the former. The thickness is usually about $\frac{1}{16}$ in., ranging from 14 to 18 B.W.G. Within this range of thickness the variation in the coefficient of heat transmission is practically negligible. In standard evaporators the most common size is 2 in. diameter by 48 in. long. Steam tubes are usually of smaller diameter and much longer, a common size being $\frac{3}{4}$ in. diameter by 12 ft. to 14 ft. long.

19 The fouling of tubes on the liquor side presents a difficult problem. The rapidity with which scale forms depends upon several factors, such as method of clarification used, amount of lime used, quantity of gums remaining in the juice after clarification, velocity of juice circulation, etc. This scale collects on the inside of juice tubes and on the outside of steam tubes. It can be removed by filling the juice compartment with a solution of soda or hydrochloric acid and boiling; it is generally necessary to do this about once a week. The juice scale is usually worse in the last bodies on account of the increased density of the juice. Makers of some special types of evaporators claim for them that the juice scale can be removed by reversing the flow of vapors. The cool body is thus made the hot body, and expansion of the metal of the tubes cracks the scale. A very convenient way to clean the tubes of standard horizontal evaporators is to remove them and crack the scale by mechanical means.

20 Rapid juice circulation is also a factor in securing efficient heat transmission. The heat transmitted through the tube walls produces vapor bubbles which tend to collect on the surface of the tube and in turn prevent heat transmission. A rapid circulation brushes these bubbles away and increases the number of contacts between the cooler surface and the liquid. As stated, circulation is produced in evaporators by convection and steam currents, except in a few special types. In vertical juice-tube evaporators, the velocity varies with the proportions of the tubes, that is, with the ratio of the heating surface per tube divided by the carrying area of the tube.

21 A matter of considerable importance is that of entraining or priming. Vertical juice tubes, especially, are likely to *spout* if they are forced too hard. This can be overcome, however, by the

use of baffles in the vapor space and by means of separators in the vapor pipes. The circulation in horizontal evaporators is less rapid and not so well defined as in vertical evaporators having usually a central downtake or circulation tube.

22 Important among the factors affecting temperature fall are initial steam pressure and vacuum in last body. The former is limited, 5 lb. gage being an average, with 10 lb. the maximum. If sugar juice is boiled at a temperature above 220 to 230 deg. fahr. there is danger of injuring the sugar. High pressure also reduces the capacity of the mill and other engines due to the high back pressure. The vacuum in the last body is generally about 26 in. With 10 lb. initial steam pressure this would give a total temperature fall of 114 deg., and a pressure of 5 lb. would give a total fall of 102 deg. which must be divided among the several bodies of a multiple effect. This means greater temperature fall and capacity per unit of heating surface in a triple than in a quadruple, and still more in a double. Simple calculations will show that a decrease of pressure at the condenser end will increase the total temperature fall much more than will an equal increase of initial pressure. Some have contended that high vacuum is desirable and it is not uncommon to specify a vacuum of 27 in. for evaporator installments. It is doubtful, however, if a vacuum higher than 26 in. is desirable, especially in submerged-tube evaporators which operate with considerable hydrostatic head.

23 Hydrostatic head (submergence of the heating surface) decreases heat transmission by increasing the temperature at which the liquor boils. This makes the temperature fall less than that obtained by subtracting the temperature corresponding to the vapor pressure from the temperature of the steam in the calandria (apparent temperature fall). This loss varies in the different bodies inversely with the absolute pressure of boiling, that is, with a given height of boiling the loss will be greatest in the last body and proportionately less in preceding bodies. Fig. 3 shows the theoretical loss due to hydrostatic head for the three bodies of a triple effect with apparent temperature falls of 20 deg, 30 deg. and 50 deg, respectively in the first, second and third bodies, with heads varying from 0 to 48 in. Developments in evaporator design have been largely affected by this question. The loss due to hydrostatic head is greatest in vertical submerged-tube evaporators, and is less in horizontal steam-tube evaporators, while film evaporators avoid it entirely. In view of the large loss due to this cause when the

absolute pressure is very low, film evaporators can make better use of high vacuum in the last body than can submerged-tube evaporators.

24 High density of the liquor being boiled causes reduced heat transmission by making the actual temperature fall lower than the apparent fall. This loss is still greater with sugar solutions of low purity. For example, in a sugar solution with a density of 70 deg. Brix and a purity of 100 per cent, the boiling temperature is about 12.5 deg.

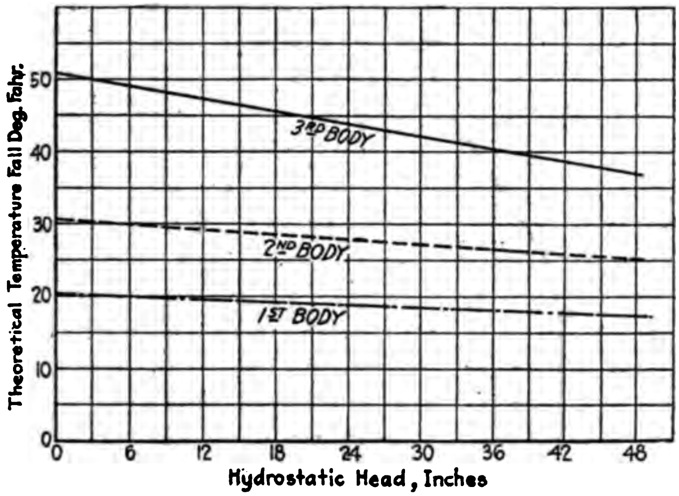

FIG. 3 THEORETICAL LOSS IN "TEMPERATURE FALL" DUE TO HYDROSTATIC HEAD IN THE THREE BODIES OF A TRIPLE EFFECT

above the saturation temperature of steam, whereas with the same density and a purity of 60 per cent the temperature of boiling is approximately 18.5 deg. above the saturation temperature of steam.

TESTS OF EVAPORATORS IN SUGAR FACTORIES

25 The object of these tests was to gather data regarding the capacity and economy of different types under regular operating conditions. Observations were made of practically all factors which might affect the results. A typical log included observations of the pressure or vacuum in the steam or vapor space of each body; temperature and density of juice entering and leaving; weight of juice entering or leaving; weight of condensed steam from the first

body, etc. This form of log was used in tests where data regarding both capacity and economy were sought. It was not possible to weigh the condensed steam from the first body in all cases, so that

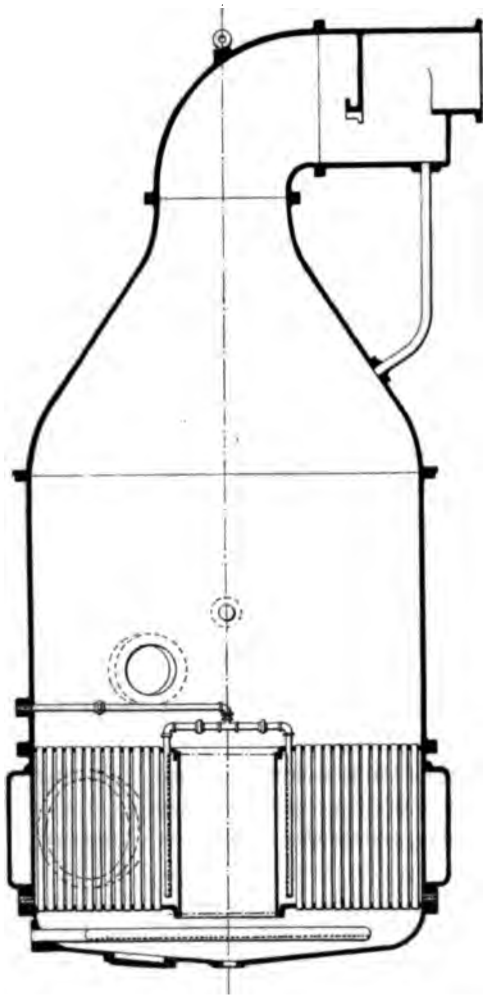

FIG. 4 ONE BODY OF VERTICAL SUBMERGED-TUBE EVAPORATOR WITH BELT STEAM DISTRIBUTION. (TYPE A)

only capacity data were obtained in many of the tests. In the logs for such tests the weight of the condensed steam from the first body was omitted.

26 The weight of juice was determined by measuring the volume in factory tanks and then calculating the weight from the densities. The condensation from the first body (steam supplied) was determined by weighing or by a venturi meter, the former in most of the tests.

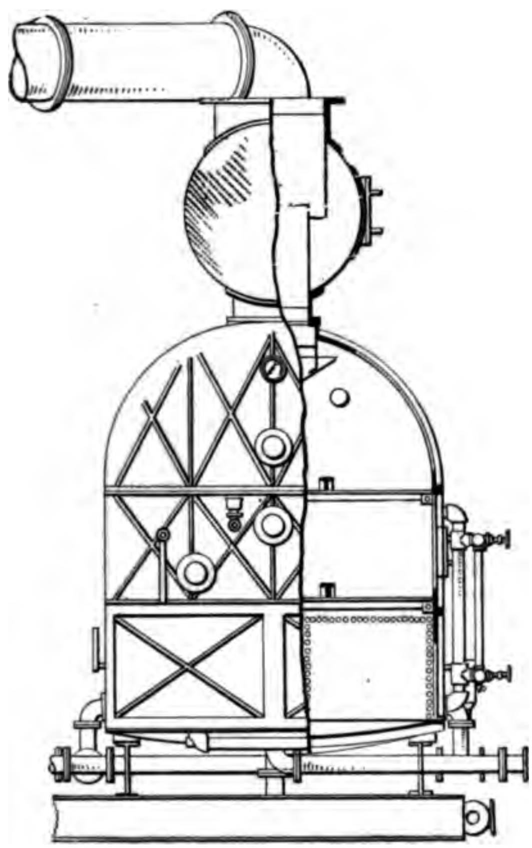

FIG. 5a ELEVATION OF HORIZONTAL EVAPORATOR (TYPE B)

The instruments for measuring pressures, vacuums, temperatures, etc., were calibrated.

EVAPORATORS TESTED

27 The evaporators tested included double, triple and quadruple effects, also the following types:

A Vertical submerged tube (so-called Standard, Fig. 4).
B Horizontal steam tube (Fig. 5).
C Horizontal film (Fig. 6).
D Vertical film (Fig. 7).
E Standard type with special baffle steam distribution, running under vacuum (Fig. 8).

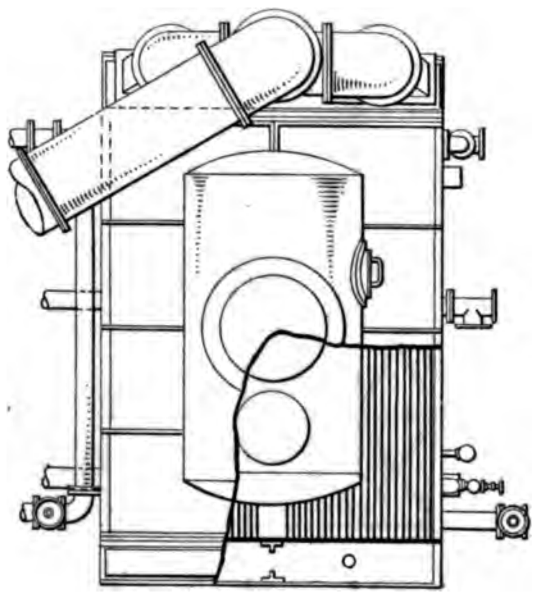

FIG. 5b PLAN VIEW OF EVAPORATOR, FIG. 5a

F Same as E but with atmospheric pressure in last body, no vacuum apparatus being used.
G Vertical steam tubes with special means for venting each tube (Double Tube, Fig. 9).

28 Types A and B (Figs. 4 and 5) are so familiar that it is not necessary to describe them. It may be well to state, however, that the majority of the sugar-house evaporators used in practice belong to one or the other of these two general types.

29 In type C, Fig. 6, the horizontal tubes are expanded at one end into a thick tube plate, and at the other are closed except for a small vent which releases the incondensable gases into the vapor space. The water of condensation leaves the tube at the end where

the steam enters.. Juice is supplied by centrifugal pumps to a distributor above the tubes and falls in a film from one tube to another. The piping is such that the juice may be recirculated over the tubes until the desired concentration in each body is obtained.

30 In the vertical film type D, Fig. 7, the juice level is near the bottom of the tubes, the latter being approximately 20 ft. long. A

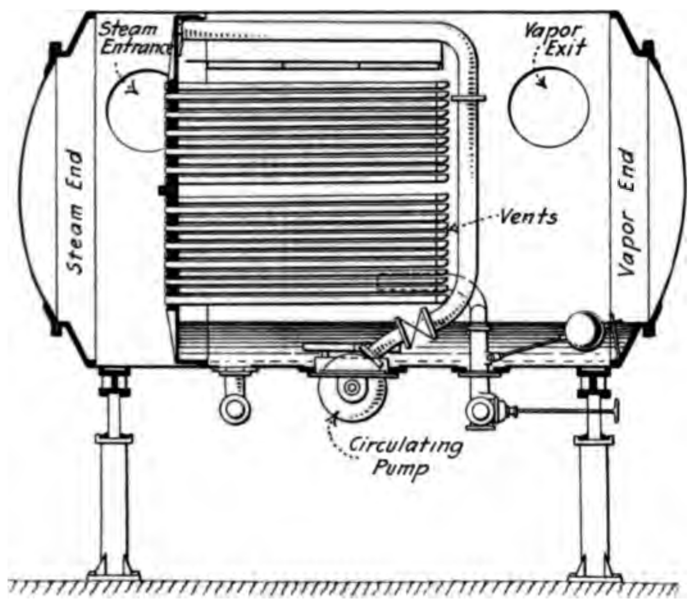

FIG. 6 ONE BODY OF TYPE C EVAPORATOR

rapid current of steam rising through the tubes sweeps juice along with it. The pressure of the steam on the interior of the tube keeps this juice as a film (often referred to as *climbing film*) on the tube surface. The juice thus projected upward, on issuing from the tops of the tubes, enters a special separator which separates it from the vapor. The juice and vapor then pass from body to body in the usual manner.

CONDITIONS OF OPERATION

31 In order to interpret the results of the tests, it is important to note some of the details of conditions of operation. The most important of these are:

1 Method of removing condensation
2 Method of venting the incondensable gases
3 Condition of the heating surface.

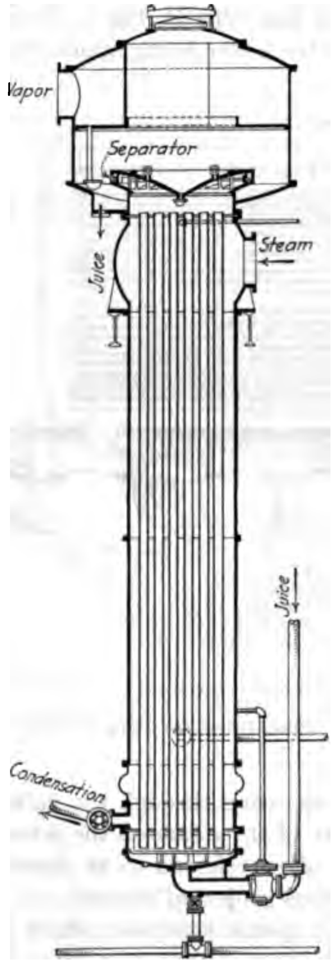

FIG. 7 ONE BODY OF TYPE D EVAPORATOR

32 Referring to Fig. 1, the removal of condensation from the steam compartments may be effected by:

a Pumps from each body
b Siphoning the condensation from body to body, a pump being used to remove it from the last body

c Barometric leg pipe; this method can be used only in case the evaporator is high enough to give a column of water sufficient to balance the vacuum in the space being drained.

33 The venting of incondensable gases may be effected by:

a Small pipes tapped through the shell just under the top tube sheet

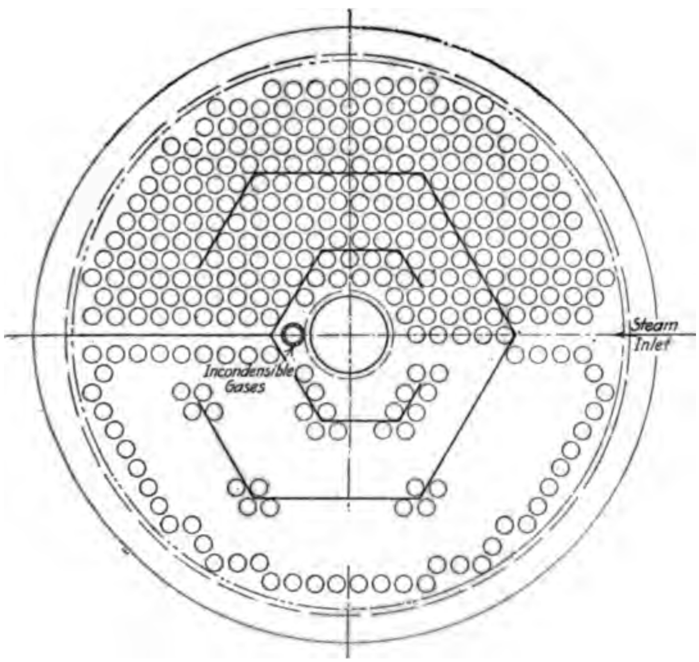

FIG. 8 STEAM DISTRIBUTION (TYPES E AND F EVAPORATORS)

b Small pipes tapped into the top tube sheet and located near the downtake. Usually two or four of these tubes are used, which are connected together inside of the vapor space with a single pipe brought through the shell and a valve placed on the outside for control

c In horizontal evaporators the steam chest opposite the steam entrance is vented

d Special methods depending on the type of evaporator.

34 In all of the above the vented gases may be taken direct to the condenser or to the next body, though so far as heat trans-

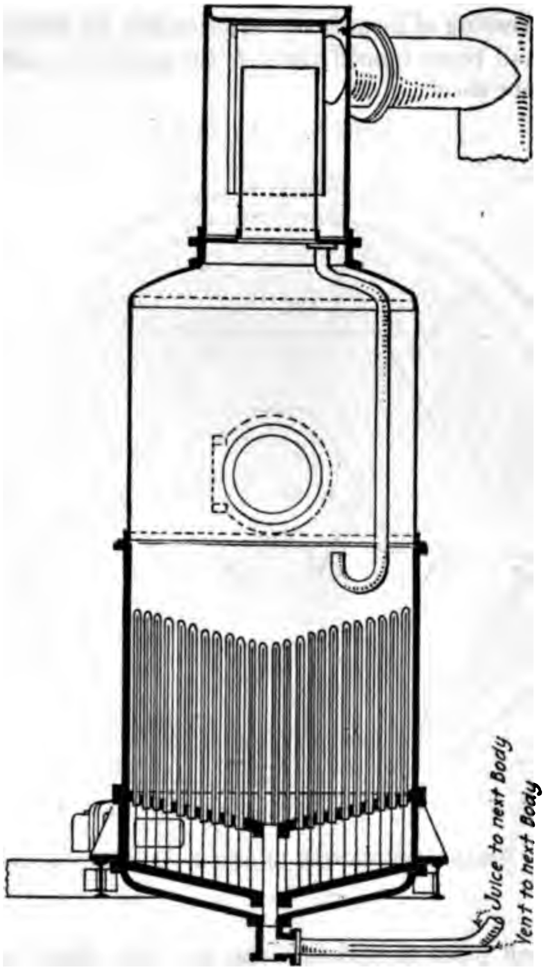

FIG. 9 DOUBLE-TUBE EVAPORATOR (TYPE G)

See also Fig. 6, Tests upon the Transmission of Heat in Vacuum Evaporators, Trans. Am.Soc.M.E., vol. 35, p. 731.

mission is concerned, it is doubtless better to vent direct to the condenser. Usually valves are placed in vent lines in order that control may be exercised. In some cases the venting is continuous and in other cases intermittent, depending upon the ideas of the operator. In types C and G special means of venting are used which have been explained above. Data regarding the methods of venting are given in the tables.

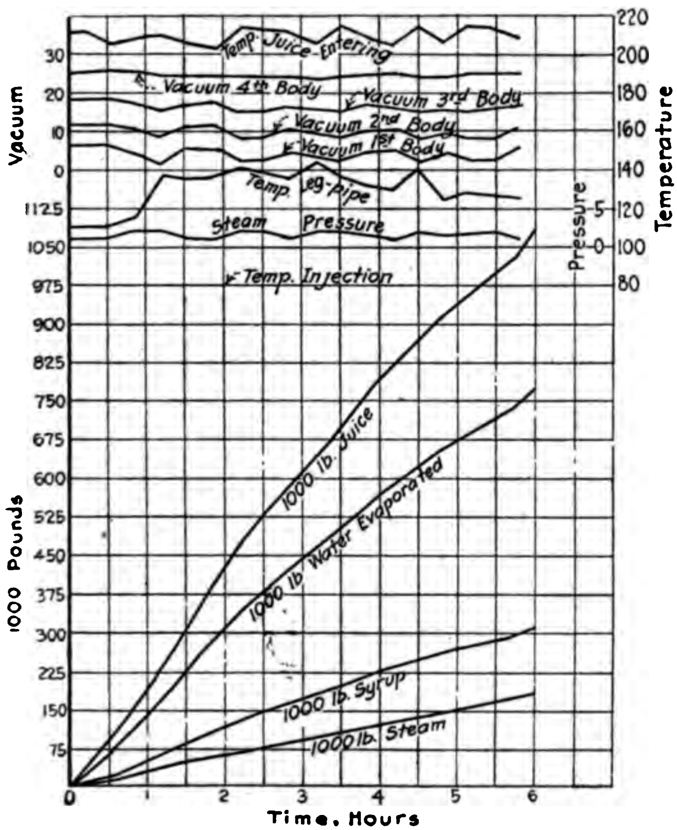

FIG. 10 GRAPHICAL LOG OF TYPICAL EVAPORATOR TEST

RESULTS OF TESTS

35 Fig. 10 is a graphical log of a typical test. Tables 3 and 4 give the more important data from the tests on multiple evaporators.

36 The percentage evaporation by weight was calculated by means of the formula $X = \frac{(B-b)}{B} \times 100$, in which B is the Brix of the juice leaving, b the Brix of the juice entering, X the per cent

TABLE 1 PRINCIPAL DATA FROM TESTS ON SUGAR-HOUSE EVAPORATORS

No.	Duration, Hr.	Type	Number of Bodies	Rated Capacity, Gal. per 24 Hr., 75 Per Cent Evaporation	Length of Tubes, Ft.	Diam. of Tubes, In.	Heating Surface, Each Body, Sq. Ft.	Steam Pressure 1st Body, Lb. Abs.	Steam Pressure Last Body, In. Abs.	Temperature, Deg. Fahr.		Density Degrees Brix at 17.5° C.		Method of Removing Condensation*	Method of Venting*	Number of Days Since Cleaned	Per Cent Evaporation		Juice Treated per 24 Hr. Equiv. to 75 Per Cent Evap., Per Cent of Rated	Water Evaporated, Lb.	Water Evaporated per Lb. of Steam Supplied, Lb.	Water Evaporated per Sq. Ft. Heating Surface per Hr., Lb.	Thermal Efficiency, Per Cent
										Juice Entering	Juice Leaving	Juice Entering	Juice Leaving				By Weight	By Volume					
1	7.13	A	4	290,000	4.5	2	4504.8	16.3	1 8	183.5	101	13.7	50.2	*f*	*bc*	6	72.7	76.6	137	587,914	....	5.44	....
2	6.00	A	3	270,000	4	2	3740	14.6	3.6	185.9	134.2	15.2	55.5	....	......	3	72.6	77	......	350,584	....	5.19	....
3	6.66	A	3	290,000	4	2	3983	15.3	3.2	188.6	....	13.2	52.2	*e*	none	3	74.7	78.6	102	505,304	....	6.35	....
4	12.0	A	3	185,000	4	2	2579	15.6	3.4	198.4	....	15 4	61.1	*e*	*bd*	0	74.7	79.4	138	933,684	....	10.38	....
5	5.75	A	3	280,000	4	2	....	16.7	4 9	183 8	....	13.6	51	*e*	note 1	0	73.4	77.4	83	327,466	....	4.96	....
6	4.33	A	3	210,000	4	2	2855	14.7	3.5	194	124 5	13.3	58.2	*e*	none	7	77.2	81.3	87	195,763	....	5.27	....
7	4.92	A	2	125,000	4	2	1901	14.9	5.2	206.7	128.9	15.9	56	*e*	*ad*	....	71.6	76.1	145	104,775	2.09	5.67	....
8	3.5	A	2	120,000	4	2	1808	13.6	5.1	172.7	137.1	16.9	42.4	*e*	*ac*	12	60.3	64.4	86	88,255	....	6.95	....
9	6.0	A	2	130,000	4	2	2137.6	14.4	4.1	176.8	123.2	13.6	52 4	*e*	none	0	74.1	78.1	63	124,474	....	4.85	....
10	4 5	A	2	130,000	4	2	2137.6	14.5	3.8	174.3	123.1	13 2	51.3	*e*	none	6	74.3	78.2	42	54,504	....	3.19	....
11	6.0	A	2	100,000	4	2	1585.6	14 9	6.2	185.2	143.4	15	48.3	*e*	*bd*	3	68.9	73.01	64	95,857	....	5.04	....
12	6.16	A	2	100,000	4	2	1585.5	14.7	6.9	188.0	144	14.1	49.4	*e*	*bc*	0	71.5	75.5	90	141,888	....	7.26	....
13	5.0	A	2	125,000	4	2	2042	14.7	5.0	188.6	140	16.3	54 2	*e*	*bd*	3	75.4	79.4	151	239,478	....	11.72	....
14	4.0	B	4	145,000	9½	¾	2000	22.9	3.1	200	119.8	18 1	55.9	note 2	*d*	6	67.6	72.5	109	155,198	3.52	4 85	85.06
15	4 0	B	4	145,000	9½	¾	2000	21.4	3.8	199.7	129.2	17.5	62.9	note 2	*d*	3½	72.2	77.2	174	248,701	3.7	7.77	89.05

16	6.0	B	4	145,000	9½	¾	2000	23.4	4.0	200.5	127.3	16.0	47.2	note 2	d	7	66.1	70.3	107	225,586	....	4.69	....
17	6.0	B	4	250,000	13	⅞	3432	18.7	3.7	181.0	....	13.0	57.5	e	c	0	77.5	81.4	86	314,062	3.4	5.08	90.48
18	4.5	B	4	250,000	13	⅞	3432	17.4	5.0	196.5	140	13.0	59.3	e	c	0	78.1	82.1	116	319,833	....	5.18	....
19	6	B	3	125,000	11	¾	1560	15.5	7.5	186.2	135	13.2	54.3	e	c	....	75.7	79.7	62	113,716	2.86	4.05	100.4
20	5.33	B	3	200,000	13	1¼	2849	20.9	3.3	195	130	10.6	58.3	e	c	1	83.5	85.1	191	502,311	....	11.03	....
21	6.00	B	3	30,000	10	1¼	654.3	20.1	3.3	195	131	11.7	58.5	e	none	1	79.9	83.1	290	130,669	....	11.09	....
22	5.00	B	2	100,000	13	1¼	1633.6	15.5	4.6	172.7	137.1	16.2	53.7	e	none	10	69.8	74.3	97.9	116,729	....	7.1	...
23	6.00	C	4	400,000	7¼	4½	2570	19.1	5.9	194	141	12.7	53.2	g	spec.	0	76.1	79.9	94	556,923	3.71	9.02	96.9
24	6.00	C	4	400,000	7¼	4½	2570	17.7	6.5	191.8	144	13.1	58.7	g	spec.	0	77.7	81.7	94	553,877	3.95	8.98	97.5
25	6.06	C	4	400,000	7¼	4½	2410	17.9	4.6	188.8	131	12.7	65.4	g	spec.	0	80.6	84.6	76	450,293	3.69	7.71	97.5
26	6	C	4	275,000	7¾	3	1940	9.4	4.0	199.3	132	14.9	52.3	f	spec.	0	71.6	75.8	62	264,780	....	5.68	....
27	7.13	D	4	400,000	....	....	4000	17.9	5.2	205.0	141.8	11.6	47.3	f	note 3	3	75.5	78.9	138	837,836	4.21	7.34	97.2
28	6.03	D	4	400,000	....	....	4000	17.9	4.9	207.7	136.9	12.6	46.2	f	note 3	2	72.7	76.7	124	748,958	4.24	7.76	94.8
29	6	D	4	400,000	....	....	4000	18.1	5.2	203.8	132.6	12.8	45.4	f	note 3	0	71.8	75.5	131	790,172	4.28	8.23	93.3
30	4	D	3	200,000	23	....	2112	14.8	2.3	212.2	125	10.5	46.2	f	none	....	64.3	68.5	93	191,585	2.91	7.77	86.5
31	5	D	3	200,000	23	....	2112	13.6	2.3	213	123.1	15.2	50.1	f	none	....	69.7	73.9	95	247,400	3.09	7.81	90.6
32	5.87	D	3	145,000	26	2	....	18	3.6	175.4	124.5	16.5	54.3	f	none	1	69.7	74.3	85	173,133	....	6.26	....
33	6	E	3	220,000	4	2	3054	14.6	4.6	171	136	14.9	58	f	Fig. 8	1	75.4	78.8	105	338,859	2.49	6.27	97.4
34	5.39	E	3	......	4	2	2683	11.06	5.0	184.6	129.2	15	51	f	note 4	2	70.6	74.8	......	190,517	....	4.4	...
35	4	E	3	......	4	2	2683	6.07	3.2	179	....	14.7	50.8	f	note 4	7	73	75.2	......	156,176	....	5.3	...
36	6	F	2	......	4	2	1003.1	24	30.0	204.4	....	15.2	45.3	e	a	0	66.5	70.5	......	73,502	....	6.11	....
37	6	F	2	......	4	2	1007.0	24	30.0	192.6	....	15.9	46.2	e	a	1.5	65.6	69.8	......	77,161	....	6.43	....
38	5.69	G	4	290,000	4	2	1529	15.4	5.8	188.1	147.4	15	64.2	e	Fig. 9	3	76.7	81.2	51	206,917	....	5.95	....

* a — through shell at side; b — through top tube sheet; c — direct to condenser; d — body to body; e — pumps, each body; f — siphon, body to body. g — barometric leg pipe.

Note 1 — vented through 12 ⅟₁₆-in. holes direct to vapor space.

Note 2 — centrifugal pumps, each body.

Note 3 — 2-in. vents at side, bottom and top, body to body.

Note 4 — see Fig. 8.

TABLE 4 RELATIVE TEMPERATURE FALL AND COEFFICIENT OF HEAT TRANSMISSION

DATA FROM TESTS ON SUGAR-HOUSE EVAPORATORS

No.	Temp. Fall 1st Body, Deg. Fahr.	Relative Temperature Fall				Coefficient of Heat Transmission					No. of Days Since Cleaned
		1st Body	2d Body	3d Body	4th Body	1st Body	2d Body	3d Body	4th Body	Average	
1	17.4	1.0	0.67	0.87	4.41	315	430	372	62	172	6
2	14.8	1.0	1.89	3.21	......	556	163	111	......	200	3
3	33.4	1.0	0.27	1.62	......	174	663	114	......	184	3
4	17.3	1.0	1.22	3.31	......	447	355	147	......	247	0
5	14.0	1.0	1.85	3.08	......	371	170	113	......	174	0
6	4.4	1.0	8.30	11.61	......	1185	128	107	......	166	7
7	28.1	1.0	1.74	......	......	181	111	......	......	136	......
8	14.0	1.0	4.28	......	......	501	108	......	......	182	12
9	32.6	1.0	1.60	......	......	145	90	......	......	111	0
10	32.0	1.0	1.75	......	......	98	55	......	......	70	6
11	35.7	1.0	0.98	......	......	134	141	......	......	137	3
12	30.5	1.0	1.15	......	......	226	203	......	......	214	0
13	29.4	1.0	1.64	......	......	380	234	......	......	289	3
14	18.9	1.0	0.69	1.85	2.87	239	270	97	79	130	6
15	21.1	1.0	0.52	1.22	2.50	321	508	203	90	202	3.5
16	19.2	1.0	0.66	1.88	2.24	192	344	115	113	154	7
17	17.9	1.0	0.95	1.26	2.45	220	202	161	87	146	0
18	5.7	1.0	3.71	3.57	7.42	899	218	260	124	236	0
19	8.6	1.0	1.23	5.26	......	482	360	87	......	184	......
20	23.2	1.0	0.74	1.93	......	460	582	160	......	287	1
21	26.9	1.0	0.92	2.14	......	393	399	197	......	291	1
22	23.4	1.0	2.61	......	......	320	105	......	......	164	10
23	15.0	1.0	0.80	0.93	2.93	545	648	600	207	394	0
24	14.0	1.0	0.95	1.07	2.52	610	593	552	286	449	0
25	11.6	1.0	1.02	1.27	4.62	680	644	505	143	334	0
26	10.7	1.0	1.18	1.29	2.63	446	430	405	207	329	0
27	14.5	1.0	1.14	1.10	2.76	464	408	448	193	325	3
28	15.2	1.0	1.11	1.14	2.60	444	405	426	209	328	2
29	18.0	1.0	0.67	1.01	2.17	397	605	378	244	353	0
30	16.1	1.0	1.50	4.14	......	402	287	107	......	192	......
31	16.8	1.0	1.64	3.43	......	366	260	132	......	205	......
32	12.2	1.0	1.84	5.48	......	416	295	113	......	190	1
33	19.8	1.0	0.75	2.01	......	317	393	152	......	244	1
34	5.0	1.0	3.29	6.38	......	769	219	130	......	217	2
35	7.6	1.0	1.18	4.91	......	623	558	149	......	284	7
36	7.9	1.0	1.99	......	......	776	374	......	......	509	0
37	11.1	1.0	1.32	......	......	596	424	......	......	498	1.5
38	8.0	1.0	1.13	1.88	5.38	682	571	365	138	293	2

water evaporated in terms of the weight of the juice. In determining the percentage evaporation by volume, the formula $X = 100 - 100\,(gb/GB)$ was used, g being the specific gravity of the juice entering, and G the specific gravity of the juice leaving. In most cases the volume of the juice leaving was measured in factory tanks, the weight of juice entering and the weight of water evaporated being calculated by the use of the first formula above.

37 It is customary to make evaporator guarantees on a basis of 75 per cent evaporation by volume. As the initial and final densities of the juice are seldom such as to give exactly 75 per cent evaporation, it is necessary to determine the "equivalent volume of juice in gallons per 24 hr." by calculation. There seems to be no standard unit for rating evaporators. Sometimes an evaporator is guaranteed to handle a stated number of gallons of juice per 24 hr. with 75 per cent evaporation, nothing being stated as to the temperature of the juice; in other cases the temperature of the juice is specified. Probably the best standard rating would be "equivalent gallons of juice handled per 24 hr. with 75 per cent evaporation." Primarily a multiple evaporator is supposed to evaporate and not heat; actually, however, the juice may enter the first body at a temperature lower or higher than that of boiling.

38 The following example shows the method used in calculating "equivalent juice treated with 75 per cent evaporation" (see Table 3).

Boiling temperature, first body, deg. fahr.	194.07
Temperature juice entering first body	172.71
Temperature rise	21.36
Specific heat juice entering	0.889
B.t.u. per hour, first body, for heating 41,832 × 21.36 × 0.889	794,349
Latent heat at 194.07 deg.	981.35
Equivalent evaporation, lb., $\frac{794,349}{981.35}$ =	809.4
Total per 24 hr., lb.	19,425.6
Actually evaporated per 24 hr., lb	605,174.4
Equivalent, had all heat been expended in evaporating, lb.	624,600.0
Equivalent gallons at 194 deg.	73,456.3
Equivalent juice at 75 per cent evaporation, 73,456 ÷ 0.75 =	97,941.0
Rated capacity, gallons per 24 hr	100,000
Per cent rated capacity actually developed	97.9

39 The term *thermal efficiency* (Table 3, last column) is used to indicate the heat efficiency of the evaporator as a whole. It may be

determined by dividing the actual weight of water evaporated per pound of steam supplied by the theoretical weight of water that would be evaporated per pound of steam supplied were there no heat losses.

40 In order to illustrate the method used in calculating efficiency, the following example is worked out for a double effect. The general data used in working out the heat balance on which the efficiency determination is based are:

Total juice fed, lb.	94,706.9
Total steam condensed, lb.	36,302.0
Brix of juice entering	12.2
Specific heat[1]	0.919
Temperature juice entering	186.06
Calandria, first body	
Steam pressure, lb. abs.	15.46
Steam temperature	214.56
Total heat above 32 deg.	1,151.33
Juice side, first body	
Vapor pressure, in. merc. abs.	25.65
Vapor temperature	204
Total heat above 32 deg.	1,147.3
Latent heat above 32 deg.	975.18
Juice side, second body	
Vapor pressure, in. merc. abs.	7.94
Latent heat above 32 deg.	1,006.2

The heat balance was calculated as follows:

First Body

B.t.u. entering in steam, $36{,}302 \times 1{,}151.33 =$	41,795,581.6
B.t.u. entering in juice, $94{,}706.9 \times 154.06 \times 0.919 =$	408,710.8
Total	55,204,292.4
B.t.u. leaving in condensation, $36{,}302 \times 182.56 =$	6,627,293.1
B.t.u. in juice in body, $94{,}706.9 \times 0.919 \times 172.32 =$	14,997,981.6
Sum	21,675,274.7
B.t.u. leaving in vapors $(55{,}204{,}292.4 - 21{,}625{,}274.7) =$	33,579,017.7
Vapor leaving, lb., $33{,}579{,}017 \div 975.18 =$	34,433.7
Juice to second body, lb., $94{,}706.9 - 34{,}433.7 =$	60,273.2
Brix of juice entering second body, $\dfrac{94{,}706.9 \times 12.2}{60{,}273.2} =$	19.17
Specific heat at 19.17 Brix	0.874

Second Body

B.t.u. entering in vapors, $34{,}433.7 \times 1{,}147.4 =$	39,510,260.4
B.t.u. entering in juice, $60{,}273.2 \times 172.32 \times 0.874 =$	9,077,606.8
Total	48,587,867.2

[1] The values for specific heat of sugar solutions with varying density were taken from Kopp's tables.

B.t.u. leaving in condensation, 34,433.7 × 172.32 =	5,933,615.2
B.t.u. in juice in body, 60,273.2 × 120 × 0.874 =	6,321,453.2
B.t.u. leaving in vapors...........................	36,332,798.8
Vapors leaving, lb., 36,332,798.8 ÷ 1006.2 =	36,108.9
Juice leaving second body, lb........................	24,164.3
Brix leaving second body, $\frac{60,273.2 \times 19.17}{241,164.3} =$	47.8
Total water evaporated, lb., theoretical, 34,433.8 + 36,108.9 =	70,542.7
Per cent of total evaporation, in first body...........	48.8
Per cent of total evaporation, in second body.........	51.2

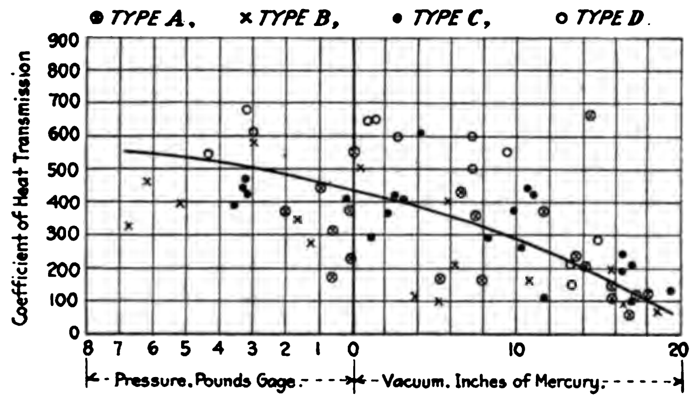

FIG. 11 COEFFICIENTS OF HEAT TRANSMISSION, ALL TESTS ON SUGAR-HOUSE EVAPORATORS

Total water evaporated (theoretical) per lb. of steam 70,542.6 ÷ 36,302 =.........................	1.9432
Total water evaporated (actual) per lb. of steam......	1.9108
Thermal efficiency $\frac{1.9108}{1.9432}$ 100 =	98.33

HEAT BALANCE OF DOUBLE EFFECT

Heat entering in steam, B.t.u......................	41,795,581.6
Heat entering in juice............................	13,408,710.8
Total...	55,204,292.4
Heat leaving in condensation, first body.............	6,627,293.1
Heat leaving in condensation, second body...........	5,933,615.2
Heat leaving in vapors, second body.................	36,332,798.8
Heat leaving in syrup, second body..................	6,321,453.2
Total...	55,215,164.3

41 The coefficient of heat transmission, the B.t.u. transmitted per square foot of heating surface per hour per degree difference in temperature, is obtained for each body by dividing the heat in B.t.u. transmitted during the test by the number of hours, by the number of square feet of heating surface and by the apparent temperature fall in that body. The coefficients of heat transmission obtained in the tests are given in Table 4 for each body, also the average for all bodies. As the apparent temperature fall was used, the coefficients are termed *apparent*.

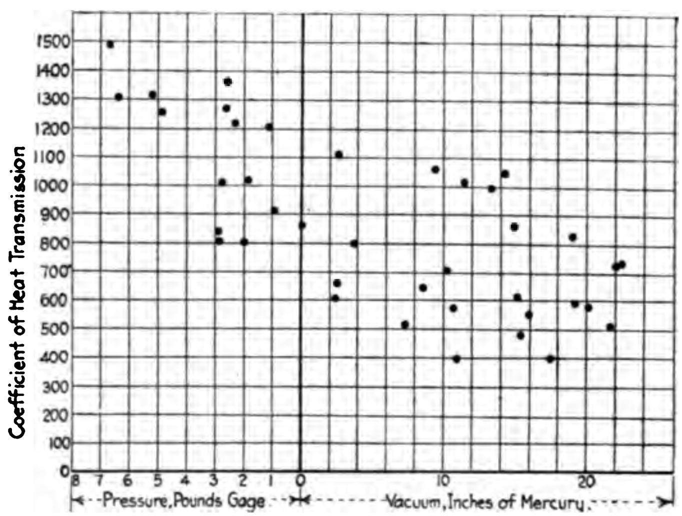

FIG. 12 COEFFICIENTS OF HEAT TRANSMISSION, LABORATORY TESTS

42 The coefficients for all tests are plotted against pressure of steam in Fig. 11. The variation is shown to be very wide, the principal value of this plot being to fix limits. Similarly Fig. 12 gives the results of the laboratory tests given in the former paper. It will be noted that the coefficients for the laboratory tests are higher than those from the full-size factory evaporators. This is doubtless due to the fact that the heating surface in the laboratory apparatus was always clean, and that there was a better distribution of steam with the smaller calandria. Some of the tests were made with the apparatus operating under conditions so poor that to consider the data from them in comparing types would be unfair. In nearly all cases the tests were made without special preparation, the time since

cleaning, etc., varying considerably. It is interesting to note that the coefficients obtained are higher than with ordinary surface condensers and equally as high as with modern types of high-vacuum surface condensers.

43 In order to afford at least an approximate comparison of the types tested, the curves of Figs. 13 to 18 inclusive have been plotted. In making up these curves a considerable number of the tests given in Table 3 have been omitted, only those operating under average or above average conditions being included.

TABLE 5 COEFFICIENT OF HEAT TRANSMISSION
(ACTUAL)

Type	1st Body	2d Body	3d Body	4th Body
A	410	340	265	145
B	400	330	240	135
C	680	630	525	220
D	430	395	320	180
F	685	400	....	....
G	682	571	365	138

44 Table 5 gives the coefficients for each body for quadruples of the different types taken from these curves. Table 6 gives the average, maximum and minimum coefficients for each type taken from the same tests. It will be noted that the highest actual coefficients were obtained from type F (atmospheric double effect). These high coefficients were doubtless due to the high steam pressure and the corresponding high density of the steam. As *apparent* temperature fall was used in determining the coefficients, they not only indicate the relative heat-transmitting ability of the heating surface in the different types, but include also the effect of peculiarities due to type, such as hydrostatic head, method of removing condensation, method of venting, etc. It will be noted that the film evaporators (types C and D) gave coefficients considerably higher than did the submerged-tube types.

45 Comparing types A and B it will be noted that the latter has some advantage, the average actual coefficient for type B being some 8 per cent greater than the average for type A. The tests of type E show an average coefficient of 248, which is some 25 per cent greater

than the average for type A. Comparing types C and D, the average coefficient for the former is 392, which is some 47 per cent greater than that of the latter. Type C requires centrifugal circulating

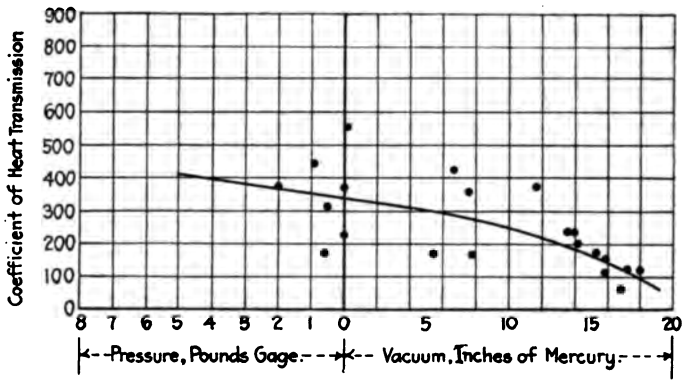

FIG. 13 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE A. TESTS 1, 2, 3, 4, 5, 12, 13

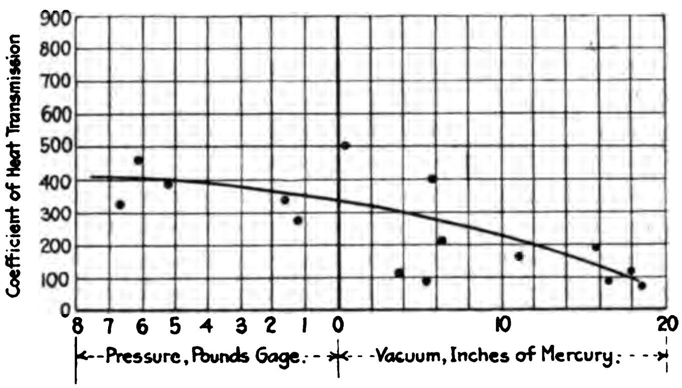

FIG. 14 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE B. TESTS 14, 15, 16, 20, 21

pumps for handling the juice, something not required by other types. Tests of motors driving these pumps showed 0.1 h.p. per 1000 gal. of juice treated per 24 hr. This should be kept in mind in comparing this type with others. Only one test was made upon type

G and this under conditions which could hardly be called favorable, the amount of juice supplied being far below its rated capacity and the juice head greater than would have been used had it been operated with an amount of juice nearer its normal capacity.

46 It will be noted that with type A only four evaporators out of a total of eleven gave coefficients above 200. In these four the conditions of operation were fair as regards cleanliness, drainage and venting. The number of days since cleaning is given for each test, though this can hardly be relied upon to give an accurate idea of the condition of the heating surface, for the reason that all of the evaporators had not been cleaned with equal thoroughness. In practice, the time consumed in boiling out, also the strength of the soda and acid, vary greatly. In some cases the thoroughness of cleaning is

TABLE 6 COEFFICIENT OF HEAT TRANSMISSION
(ACTUAL)

Type	Average	Maximum	Minimum	No. of Tests Included in Average	Relative Coefficient (Actual) Standard = 100	Relative Coefficient (Corrected) Standard = 100
A	197	289	172	7	100	100
B	213	291	130	5	108	74
C	392	449	334	3	199	160
D	266	353	190	6	135	119
E	248	284	190	3	126	180
F	503	509	498	2	255	131
G	293	....	....	1	149	139

not sufficient to prevent a progressive fouling of the heating surface toward the end of the season.

47 It will be noted that practically all of the tests on type A were made with very low steam pressure in the first body, usually near atmospheric pressure, and in some cases less than atmospheric pressure. With most of the other types the pressure was higher, which puts type A at a disadvantage in making comparisons with the average coefficients given. According to laboratory tests, the coefficient of heat transmission varies according to the expression $U = 225 + 17{,}500\,D$, where D = density of heating steam in pounds per cubic foot. This formula was used in calculating the *corrected* coefficients given in the last column of Table 6. These *corrected* coefficients may perhaps give a better means of comparing types than the actual coefficients, especially as regards types A and B. However, too much dependence should not be placed on these

corrected coefficients, as the correction formula was determined from laboratory experiments.

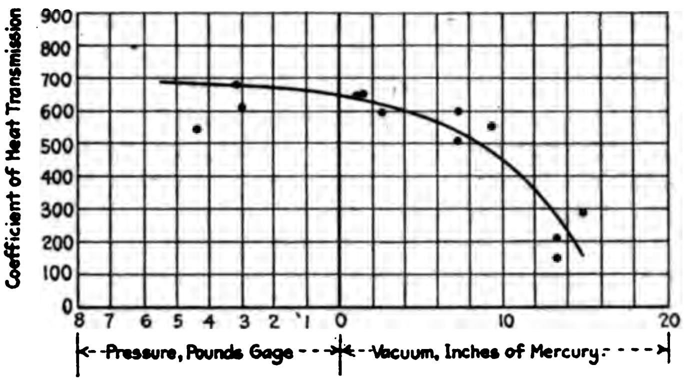

FIG. 15 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE C. TESTS 23, 24, 25

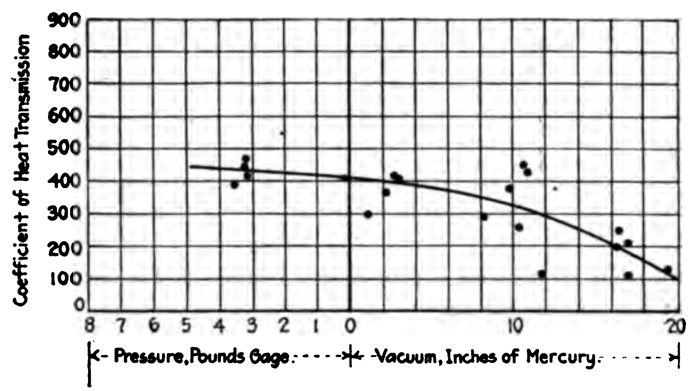

FIG. 16 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE D. TESTS 27 TO 32

48 Considering the four tests giving the highest coefficients for types A and B, the rating is:

(Actual)

Type A.......... 100
Type B.......... 106

(Corrected for difference in initial steam pressures)

Type A.......... 100
Type B.......... 96

49 The fairest comparison of types C and D will be obtained by using only tests 23 to 25 and 27 to 29, which were made on evapo-

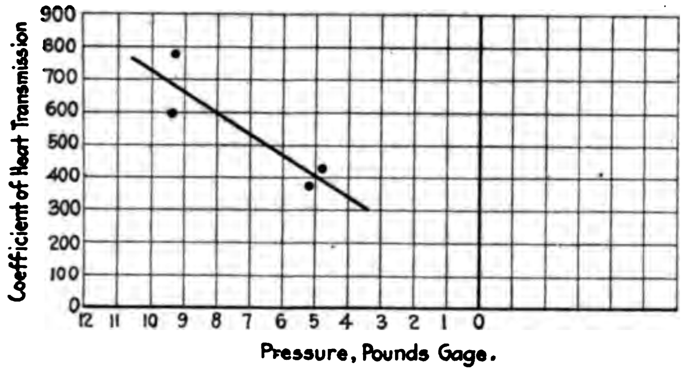

FIG. 17 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE F. TESTS 36 AND 37

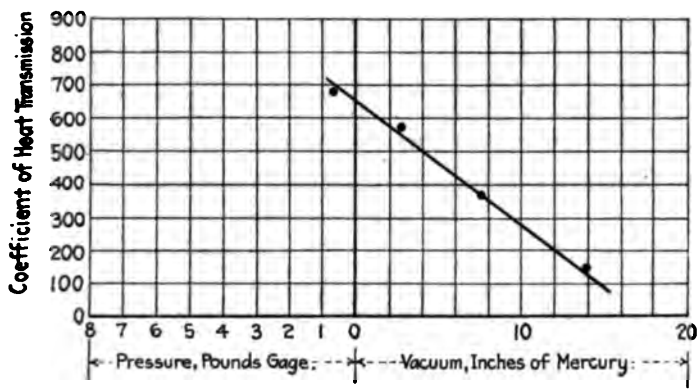

FIG. 18 COEFFICIENTS OF HEAT TRANSMISSION, TYPICAL TESTS, TYPE G. TESTS 38

rators at the same factory and where the operating conditions were equally favorable.

(Actual)

Type C..........	117
Type D..........	100

. 50 The evaporators of type B were of two classes, namely, *a* with small tubes (¾ in. and ⅞ in.) and *b* with larger tubes (1¼ in.). Tests 15 and 16 with small tubes, and tests 20 and 21 with large tubes, were made under conditions that should afford a fair comparison. The ratings are:

(Actual)

Small tubes.......	100
Large tubes.......	142

(Corrected for difference in initial steam pressures)

Small tubes.......	100
Large tubes.......	111

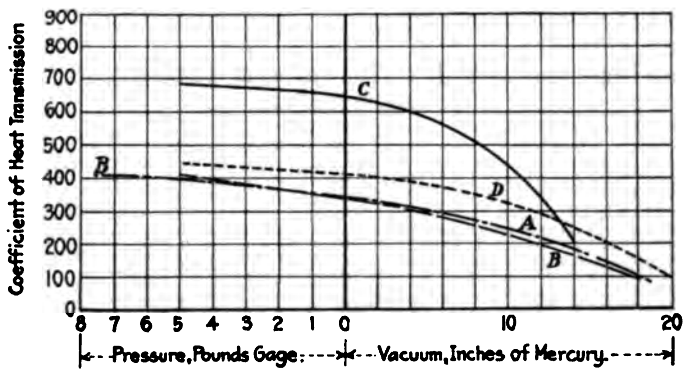

FIG. 19 COMPARISON OF HEAT-TRANSMISSION COEFFICIENTS, TYPES A, B, C AND D EVAPORATORS

RATE OF EVAPORATION PER SQUARE FOOT OF HEATING SURFACE PER HOUR

51 So far all comparisons have been made in terms of the coefficient of heat transmission. The unit "pounds of water evaporated per square foot of heating surface per hour" is convenient for rating evaporators, and is fairly satisfactory provided the steam pressure and the vacuum do not vary greatly. Table 7 gives averages of water evaporated per square foot of heating surface per hour for each type tested. The *actual* values do not furnish a fair comparison on account of the widely varying steam pressure and vacuum. An attempt has been made to determine the equivalent evaporation that would have resulted had all tests been made with

a steam pressure of 5 lb. gage in the first body and a vacuum of 26 in. in the last body. This equivalent value is termed *corrected* water evaporated per square foot of heating surface per hour. The method of calculating the *corrected* value is as follows:

52 For example, in test 27, the total temperature fall for the conditions of the test was 87 deg. and for the *corrected* conditions 102.5 deg. was assumed. The actual average coefficient for the conditions of the test was 325, and for the *corrected* conditions 343. The *corrected* rate sought is

$$\frac{8.75 \times 102.5 \times 343}{87 \times 325} = 10.7$$

TABLE 7 WATER EVAPORATED PER SQUARE FOOT OF HEATING SURFACE PER HOUR

FROM TESTS PLOTTED IN FIGS. 13 TO 18 INCLUSIVE

Type	Actual Equiv. to 75 Per Cent Evap.			Corrected 5-Lb. Gage and 26-in. Vac.		
	Maximum	Minimum	Average	Maximum	Minimum	Average
A Quadruple	....	....	5.55	....	....	6.18
A Triple	8.31	5.07	6.56	10.8	8.15	9.2
A Double	11.64	7.20	9.42	19.4	13.6	16.7
B Quadruple	7.89	4.82	5.89	5.61	3.87	4.55
B Triple	11.18	11.18	11.18	12.61	10.68	11.49
C Quadruple	9.20	7.90	8.75	11.42	9.56	11.06
D Quadruple	8.75	7.75	8.21	10.87	9.45	10.06
D Triple	7.38	6.62	7.11	9.6	7.45	9.21
E Triple	6.33	4.85	5.46	10.63	27.65	16.93
F Double	6.74	5.43	6.085	17.8	17.0	17.4
G Quadruple	....	....	6.092	....	....	9.67

53 It is not claimed that these *corrected* values can be absolutely accurate, but rather an approximation to the results that would be obtained with the conditions assumed.

Thermal Efficiency

54 In thirteen of the evaporator tests recorded in Table 3, the condensed steam from the first body was weighed. Partial data regarding steam consumed for these tests are given in the same

table. Table 8 gives more complete data relating to steam consumption in these tests.

55 The factors which affect the weight of steam required to evaporate a given weight of water are: *number of bodies; radiation losses; losses due to vents; method of handling condensation; temperature of juice entering.*

56 One pound of steam entering the first body of a multiple evaporator causes the evaporation of approximately 1 lb. of water in each body; that is, 1 lb. of steam evaporates approximately 2 lb. of water in a double effect; 3 lb. in a triple effect; and 4 lb. in a quadruple effect. This relation does not hold exactly, for reasons which will be discussed later.

57 The loss due to radiation varies directly as the difference between the temperatures inside the evaporator and the atmosphere outside. This temperature difference is of course greater with atmospheric evaporators than with vacuum evaporators. It is also greater in the first body of a vacuum evaporator than in succeeding bodies. The radiation loss also varies directly as the ratio of radiating surface to water evaporated. This ratio will be less with large evaporators than with small ones, and is affected by type also. Reference to Table 8 will show something regarding the value of this ratio for the evaporators tested. Naturally the radiation loss is affected by the kind and amount of insulating covering used. The loss due to radiation in the first bodies of an evaporator is multiplied; for example, the loss from the first body of a quadruple is multiplied by 4; that from the second body is multiplied by 3, etc. In view of this, it is evident that the total radiation loss from a quadruple will be greater than that from a triple, etc., other conditions being the same. A theoretical calculation will show that this loss should be relatively small. A rough calculation based on an average coefficient of radiation of about 2.8 B.t.u. per square foot of radiating surface per hour per degree fahrenheit difference shows the following losses in percentage of the heat delivered in the steam to the first body:

	Uncovered	Fully Covered	Three-fourths Covered
Double............	1.06	0.26	0.46
Triple.............	4.2	1.05	2.07
Quadruple.........	9.8	2.7	5.0

These figures are for a 10-ft. standard effect including the vapor pipes, with steam at 5 lb. gage and a vacuum of 27 in.

TABLE 5 DATA FROM TESTS RELATIVE TO ECONOMY

SEE FOOT NOTES, TABLE 3

Test No.	Type	No. of Bodies	Rated Capacity, Gal. per 24 hr.	Actual Capacity, Per Cent of Rated	Radiating Surface per 100,000 Gal. per 24 hr., Actually Evaporated, sq. ft.	Portion of Radiating Surface Covered, Per Cent of Total	Steam Temperature 1st Body, Deg. Fahr.	Method of Venting	Method of Removing Condensation	Steam* Supplied to 1st Body for Heating Per Cent of Total	Water Evaporated per Lb. of Steam, Lb.	Thermal Efficiency, Per Cent
7	A	2	125,000	145	....	....	212.5	*ad*	*e*	−6.13	2.09	94.5
14	B	4	145,000	109	1921	73.50	234.9	*d*	note 2	8.9	3.52	85.1
15	B	4	145,000	174	1207	73 50	232.8	*d*	note 2	5.89	3.7	89.1
17	B	4	250,000	86	1780	0	224.5	*c*	*e*	11.73	3.4	90.5
23	C	4	400,000	94	832	100	225.4	spec.	*g*	8.06	3.71	96.9
24	C	4	400,000	94	832	100	221.5	spec.	*g*	7.83	3.95	97.5
25	C	4	400,000	76	1026	100	222.0	spec.	*g*	9.67	3.69	97.5
27	D	4	400,000	138	694	95	222.0	note 3	*f*	1.19	4.21	97.2
28	D	4	400,000	124	774	95	222.1	note 3	*f*	−0.50	4.24	94.8
29	D	4	400,000	131	734	95	222.8	note 3	*f*	−2.37	4.28	93.3
30	D	3	200,000	93	946	0	212.4	none	*f*	−12.98	2.91	86.5
31	D	3	200,000	95	923	0	208.3	none	*f*	−9.85	3.09	90.6
33	E	3	220,000	105	990	40	211.9	Fig. 8	*f*	7.46	2.49	97.4

* Minus values are for initial juice temperatures higher than the boiling temperature.

58 Primarily, vent pipes are for the purpose of removing incondensable gases only. If this were carried out the vent losses would be small, in fact they are probably small in general practice, even though some steam is vented with the gases. This loss, like the radiation loss, is multiplied for all bodies except the last. The vents may be connected to the next body or to the condenser direct. The former is probably more economical, though the latter is better as far as heat transmission is concerned. This loss is apt to be greatest in the early bodies, especially if the vent openings are too large. In order to reduce losses due to venting, it is important to design evaporators so that effective separation of gases from steam may be brought about. Attention has already been called to the fact that this has been attempted in types C, E, F and G.

59 Of the three general methods of handling the condensed steam from evaporators, *b* (see footnote, Table 3) evidently should be most economical for the reason that all of the condensation leaves the last body at the lowest possible temperature, the saving due to this arrangement being multiplied in all except the last body. It is common practice to use the condensation from the first body for boiler feed, though theoretically it would pay to run this water from body to body along with that from later bodies.

60 Primarily, evaporators are designed to vaporize water, though, in fact, some heating is generally done in the first body because the temperature of the juice entering is lower than that of boiling. It is evident that the amount of heating done in this manner will affect the weight of water evaporated per pound of steam supplied to the first body. It should be remembered also that the steam supplied for heating in this manner does not work multiple as does that used for evaporation.

61 Table 8 gives the results of the tests as regards economy. It also includes information regarding the main test conditions which affect economy. A unit commonly used for measuring economy is "pounds of water evaporated per pound of steam supplied to the first body." This unit does not furnish a fair means of comparing the economy of different evaporators, however, for the reason that it does not account for variation in conditions. Thermal efficiency is more accurate, and the method of calculating this item has already been given. It will be noted that it varies from a minimum of 85.06 to 98.3, the average being 94 per cent. The table shows that the efficiency of type C was higher than type D, although the water evaporated per pound of steam supplied was greater in the latter

than in the former. This apparent contradiction is due to two things: (1) the temperature of the juice entering the first body in the tests on type D was higher than in those on type C, more steam being required for heating in type C; (2) the condensation in type D was taken from body to body, beginning with the second, whereas it was drained away separately from each body of type C. The low efficiencies obtained in the tests on type B quadruple, tests 14 and 15, are probably due to the small size, whereas both the C and D quadruples were of large size. Attention is called to the data in Table 8 regarding the ratio of radiating surface to water evaporated in the different types.

62 Test 17 on a type B quadruple evaporator gave an efficiency of 90.48 per cent. This is relatively low for an evaporator of this size and is doubtless due to the fact that it was not insulated. The low efficiencies obtained in tests 30 and 31 on a type D triple are doubtless due also to the fact that it was not insulated. These tests were made soon after the apparatus had been installed and before the covering had been applied. An inspection of this table will show the effect of high rates of evaporation in increasing heat efficiency; in fact, some of the irregularity in the results can only be explained in this way.

DISCUSSION

M. B. De Pass[1] (written). The process of sugar making is divided, we might say, into four parts:

1 Extracting, by crushing process, of juice from the cane
2 Clarification by defecation and sometimes filtration
3 Evaporation (the subject under consideration)
4 Granulation and drying.

The capacity and economy of any of the stages depend very materially on the efficiency of the preceding stage.

In some climates during the crushing stage a larger volume of water must impregnate the crushed cane to dissolve the sugar that is not in solution. This reduces the density of the juice and increases the volume of water that must needs be evaporated from it.

One factor, the paper tells us, is the fouling of the heating surface in effects which is caused by foreign substance that has not been eliminated during the clarification stages — in fact some of it is injected during this stage; hence the amount of water added in crush-

[1] Joubert & Goslin Mch. & Fdy. Co., New Orleans, La.

ing and the nature of the juice clarified entering into the effects largely determines the size and style that is to be adopted and should govern the manufacturer's guarantee.

There are two factors governing the capacity and economy of multiple effects which are not referred to in the paper. A very important one is the size of the vapor pipes and save-alls which regulate the velocity and vapor tension and fix the boiling point in each effect. Faulty vapor pipes cause entrainment and loss in capacity.

As the efficiency of a multiple effect depends very largely on the vacuum in the last body, and this in turn on the completeness of the condensation, attention to the design of the condenser is important. Owing to the low boiling point in the last vessel, the velocity of the vapors is greater and there is therefore a greater chance of entrainment. Syrup thus carried off into the condenser is never reclaimed, which is not the case in the other vessels where the sweet water may be trapped and re-evaporated or used in the first stage of extraction for saturation. So not only is a good condenser required on the last pan but also a good save-all that will completely strain the syrup from the vapor before it enters into the condenser. The writer has seen a "save-all" that reclaimed from 200 to 300 gal. of syrup in 24 hr. which would have been lost if permitted to pass into the condenser with the vapors and out into the hot well.

The standard type of vertical-tube submerged calandria is the most popular of all multiple evaporators. The vapor velocity is controlled, entrainment is prevented, perfect condensation is accomplished, air pockets are done away with and the condensed waters are properly drained from the calandrias.

In dealing with the problem of heat transmission, various means are advocated by different manufacturers. Some use baffle plates and different arrangements to control the direction of the steam, some leave out tubes, and others both. The trouble to overcome in large bodies is that some parts get more heat than others, causing unequal boiling and dead spaces in the calandrias.

In the earlier apparatus steam was admitted around a circular steam jacket on the outside of the calandria, and the drains were also on the outside directly under the steam entrance. The consequence was that all the ebullition or boiling took place on the periphery of the calandria, where the heat remained and a dead space formed in the center. This ebullition at the periphery was mistaken for circulation by someone who conceived the idea that if there were a central downtake to aid the circulation of the liquid from the center to the periphery, it would aid the evaporation; and on experiment the scheme apparently proved a success and the liquid actually appeared to work down through the downtakes and up through the tubes from center to circumference. The fact of the matter, however, was that the downtake filled the void and dead space in the center of the calandria, and it was still found that there were air and gas pockets in the center of the calandria which retarded evaporation. Something accordingly had to be done to relieve this and make the vapor, or steam, find its way to the center, which brought about the use of baffle plates, or deflectors, so arranged as to admit the vapor at one side and deflect it around in two directions to the other side, and thence to the center, as shown in Fig. 8 of the paper.

This was tried years ago at the American Sugar Refining Co.'s Plant, New Orleans, where one of its standard triple evaporators was equipped with deflecting plates, exactly as in Fig. 8. The plates deflected when the steam was first turned on, but after the entire calandria was filled the pressure became equal and the plates of no further use. In connection with these plates, however, the drain, gas and air pipes were connected as near to the center of the calandria as possible, where a partial vacuum was formed, causing the removal of the pocket, or dead space, and the steam to be drawn to the center, thus utilizing heating surface that had previously been idle.

Large 16-ft.-diameter effects, built last year in Cuba, without the plates, but with the drains to sweet-water pump, and with gas and air pipes all connected in the center of the calandria, have proved by good results that plates are not necessary.

Professor Kerr has quoted tests on most, if not all, of the various types of evaporators now on the market. It is hard to form a comparative conclusion, since conditions alter and govern each test, as proved by the fact that the same man on the same type of machine obtains different results at different centrals. The rating of any evaporator is computed by the number of pounds of water evaporation per square foot of heating surface as a basis. A multiple of four, or a quadruple effect, is now conceded to be the most economical and is being adopted by most large houses. The following list shows how the tests of different experts vary in a quadruple effect:

Hausbrand	5.2 lb. per sq. ft.
Kerr	5.5 lb. per sq. ft.
Jones & Scard	6.0 lb. per sq. ft.

Wallis & Taylor .... 7.5 lb. per sq. ft.
Hind .... 8.0 lb. per sq. ft.
Deerr .... 9.0 lb. per sq. ft.

We are indebted to the Louisiana State University for the interest it is taking, through Professor Kerr, in carrying out such tests as described in the paper, which are of so much value and interest to all concerned.

George A. Orrok (written). Professor Kerr has undertaken the investigation of one of the most complicated problems in the heat-transmission field, and the results show the difficulties en-

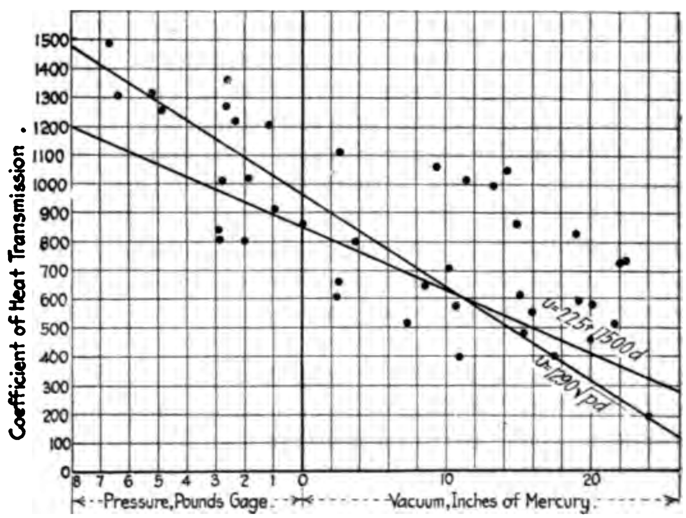

Fig. 20 Comparison of Author's and Parr's Formulæ for Coefficient of Heat Transmission

countered. Theoretically, the case is that of the transfer of latent heat, introducing two changes of state, in consequence of which both upper and lower temperatures are constant and fixed by the pressures employed. The question is complicated by the presence of both air and water films on the gas side, with approximately similar conditions on the liquid side of the tube walls. Moreover, there is lack of control of film thickness because the circulation is negative or self-induced, and an increase of certain film thickness (i.e., gas on the water side and water on the gas side) with an increase of circulation. When we add to this the scale and oil troubles on the gas side of the tube and the gumming of the surface from impure juice on the liquid side, it seems remarkable that so consistent results should have been obtained. The author has plotted in Fig. 12 the results of all his laboratory tests given in his former paper. I have plotted (Fig. 20) across these points the line $U = 225 + 17{,}500\,d$ which he has given as best representing them. I have also plotted a line $U = 1290\ \sqrt{pd}$, Parr's[1] formula, in which the coefficient was obtained from the author's figures. These two lines cross at approximately 700 B.t.u. and 8 in. of vacuum. It would appear that variations as wide as 800 to 1370 B.t.u. at about 3 lb. per sq. in. pressure and 400 to 1070 B.t.u. at 10 in. of vacuum could hardly be expressed by either curve, and therefore the use of either as a correction formula would be inadmissible. It is curious that all the performance curves in the paper cut the X-axis to the left of 22 in. of vacuum.

Clark D. Libermuth.[2] The fouling of the heating surface by scaling is one of the most important of the practical considerations of the sugar apparatus.

Although it has been known that pressures above the atmosphere could be used in the first body of the evaporator, the high pressure has been brought into use only recently. At present in Cuba there are many evaporators in which steam under pressure is used in the first body. There being no sulphuration and little liming, no difficulty is met with. These higher pressures are now being used in Louisiana. More scale will be precipitated in the first body due to the higher temperature, which may result in fouling the first body and cutting down the steam supply. The author is asked to investigate the effect of this condition in the first body with the Louisiana juices of high salt content.

Last year a rather peculiar situation presented itself over the State as a whole. The juices were of a higher purity than normal, yet the boiling took a longer time. It was found, however, that the proportion of salts was much higher than usual, resulting in a high boiling point. This suggests the investigation of the relative effects of dissociable salts and organic compounds upon the elevation of the boiling point.

[1] On the Theory of Multiple Evaporators, P. H. Parr, The Engineer, London, Feb. 18, 1916.

[2] Lauderdale P. O., La.

W. H. P. Creighton.[1] The evaporation of sugar juices is an extremely important problem to us in Louisiana. Ten cubic feet of juice in the multiple effect will be evaporated to two cubic feet, and in the vacuum pan will be evaporated to one cubic foot. This cubic foot is a half-and-half mixture of molasses and sugar, and the apparatus must be designed to boil this mass. A 1000-ton house will have about 900 tons of water to evaporate because maceration water and milk of lime will be added to the juice to be evaporated. To evaporate this water economically is a problem. There are temperature limitations; too high a temperature will discolor the sugar, while the low temperature is limited by the amount of vacuum obtainable. The *apparent* range is about 103 deg. But this is only apparent. The difference in mean temperatures in the first body added to that in the second and in the third will not total 103 deg., because sugar solutions and water do not boil at the same temperature under the same pressure. Then there is the loss due to hydrostatic head, the difference in juice height, which increases the pressure on the juice at the bottom of an effect. Hence, in the triple effect, 2 or 3 deg. may be lost in the first body, 4 or 6 in the second, and 10 or 12 deg. in the third. These added together and increased by unknown viscosity losses reduce the apparent temperature range materially.

Personally, I am much interested in the evaporator in Fig. 8, as it was due to a suggestion of mine that Mr. Weber designed it. The entering steam was to sweep over tubes through a gradually diminishing area to maintain its velocity and keep the air moving with it to the very end where it was taken out by an air pump. In the ordinary standard evaporator, there is no way of showing that the air is being removed. There is violent evaporation in spots, indicating inefficiency, as the heat transfer is confined practically to such spots and they need high temperature differences to be effective; whereas, if the air is removed and the steam sweeps through a fixed channel, a less temperature difference will be necessary in the evaporator. This is illustrated in Experiment 35, Table 3. Here the pressure in the first effect is only 6 lb. per sq. in. absolute, and in the last is 3 lb. per sq. in. absolute, a total pressure difference of only 3 lb. per sq. in. This means that if the amount of steam admitted to the first effect had been large enough, the pressure would have been greater and hence there would have been a greater capacity.

The author should say that capacity depends upon *three* factors:

[1] 1445 Henry Clay Ave., New Orleans, La.

quantity of heat supplied to the first effect, coefficient of heat transmission, and temperature fall. Scientific research in this field, considering each of the problems separately and as they affect the whole, is going to be of great value to the manufacturer and should result in the saving of an enormous amount of copper in the construction of evaporators.

E. H. Rousseau.[1] I had the pleasure of making experiments with A. L. Weber on a small experimental evaporator. Heat transmission depends upon a difference in temperature. For an effective temperature drop from one side of the tube to the other, there must be a rapid replacement of the hot particles of juice by colder particles, so that the more rapidly cold juice can be brought to the heating surface, the more rapid will be evaporation. Refer to evaporator D for the juice side of the evaporator and to evaporator E for the steam side. These I wish to combine in evaporator A, which is our standard evaporator.

Viscosity increases with the concentration of a solution. As the viscosity increases, the size of bubbles formed in ebullition becomes greater in proportion to their weight and they are carried along more rapidly with the vapor currents formed. In the first body of a quadruple-effect evaporator will be found conditions of high temperature and low density of juice. This is favorable for heat transmission. In the last body the conditions are low temperature and high density of juice. This is not favorable for heat transmission, but as the viscosity of the juice is high, the upward velocity of its bubbles is high and the temperature differences between the sides of the tubes are great, equalizing the rate of evaporation in the last with that of the first body.

In making these experiments there was used an evaporator with a steam belt 2 ft. in diameter and 3 ft. high, with 2-in., No. 17 gage, standard copper tubes. Steam was under 10 lb. per sq. in. pressure and juice under atmospheric pressure. Steam distribution was as given in type E. With each new density the level of the juice was varied from 0 to 36 in., and it was found that at 6 deg. Brix the highest rate of evaporation was when the liquor was at $\frac{1}{3}$ the height of the tube. At 55 deg. Brix the highest rate was with the liquor at $\frac{1}{4}$ the height of the tube, showing that with equal conditions of temperature the density and viscosity of the juice increasing gave better results with a low height of liquor.

[1] Dibert, Bancroft & Ross Co., New Orleans, La.

In these tests steam was used. The air eliminator was closed; the heat transmission was 250 B.t.u. Opening the air eliminator and drawing out all of the air, the heat transmission was increased to 1000 B.t.u., an increase of 400 per cent. Uniform steam pressure can be maintained by high velocity of steam, by means of baffles, or by removing some of the tubes at the entrance, allowing the steam to enter the bulk of the heating surface, a pocket being provided from which to draw the air.

Leo Loeb (written). Another application of the multiple-effect evaporators is for producing fresh water for drinking or boiler-feed purposes from water containing a large amount of soluble salts. Compared with the sugar evaporators, the coil-pressures in the first stages are generally higher, the heating surface is seldom, if ever, totally submerged, and the removal of the concentrated brine is governed by considerations of capacity and fouling of the heating surface and shell in contact with the brine.

The usual installation on shipboard consists of duplicate sets of three evaporators, so piped as to be operated in triple effect or single effect, depending upon the immediate demand for fresh water.

Of the various factors mentioned in the paper as affecting the capacity of evaporators, the most important in the design and operation of salt-water evaporators are:

5 Presence of condensation on the heating surface
8 Cleanliness of heating surface (liquor side)
14 Density of liquor being boiled.

Air and other incondensable gases have less effect in the salt-water evaporator, where higher steam pressures are used than in the sugar evaporators under vacuum, providing they are not permitted to blank off a portion of the heating surface.

The great loss in capacity due to condensation on the heating surface was clearly illustrated by recent tests on two horizontal-tube evaporators. One consisted of 1½-in. brass-tube heating surface arranged in eight horizontal rows and connected by return bends. Thus the condensate traveled the entire length of the eight tubes before being vented, with the result that the coefficients of heat transmission were low, ranging from 115 to 323 B.t.u. per sq. ft. per hr. per degree difference in temperature between steam at entrance to coils and vapor in shell. When operating in single effect, the temperature of condensate at the vent was from 32 to 176 deg.

below the temperature of saturation of steam entering the coils, and the evaporating capacity of the heating surface was from 7.7 to 19.9 lb. of vapor per sq. ft. with brine at 180 deg. fahr.

A modified design having nine rows was provided with a header, so that the condensate from any tube could reach the vent by traveling through only one cooling coil. With the same steam and vapor pressures as in the first type, the coefficients of heat transmission were from 387 to 719 B.t.u. per sq. ft. per hr. per degree temperature difference between steam at entrance to coils and vapor in shell. The amount of sub-cooling of brine never exceeded 90 deg., and the evaporating capacity of the heating surface was from 17.2 to 81.4 lb. of vapor per sq. ft. per hour with brine at 180 deg. fahr.

The reduction in capacity of the salt-water evaporator when the concentration of the brine is increased from twice to three times that of normal sea water is about 10 per cent, amounting to fully 15 per cent if the concentration be further increased to four times that of sea water.

It would be interesting to know how Professor Kerr calculated temperature difference. In salt-water evaporators the temperature gradients on the two sides of the heating surface are complex. A portion of the surface in contact with the heating medium destroys the superheat which may result from throttling at the inlet valve, a great portion of the area is effective condensing surface and the final portion is water-cooling surface. On the brine side these states are reversed, a part of the surface acting as feed-heating surface, the major portion as evaporating surface and a small portion at times as superheating surface. Under these circumstances it has been the practice to consider temperature difference as indicated above to be the difference between the saturation of steam to coils and saturation temperature of the vapor in the evaporator body.

The Author. I wish to call attention to the preliminary statement I made when this paper was presented — that it should be looked upon as a survey rather than as a research. The tests recorded were made on full-size sugar-house evaporators operating under regular conditions, therefore there was little opportunity to control conditions, as would be required for research work in the fullest sense. A former paper[1] by the author giving the results of a series of experiments on a laboratory evaporator is better entitled to consideration as research work, and I would respectfully direct the attention

[1] Trans.Am.Soc.M.E., vol. 35, p. 731.

of Mr. Creighton, who mentioned in his discussion the need of research along these lines, to this paper. The tests recorded in the present paper give data only on the subject of rates of heat transmission, nothing having been said relative to the cost of heating surface, the cost of operation, maintenance, etc. The proper selection of an evaporator for a specific installation should of course include a consideration of all of these points.

Mr. De Pass calls attention to the loss of sugar due to entrainment. This bears on the present paper only indirectly because of the fact that, in order to separate the entrained juice, save-alls must be installed and the friction of the vapors passing through them may result in pressure losses which in turn reduce capacity by reducing temperature fall. In elaborate tests made by the author on a laboratory evaporator, data were obtained which seemed to show that entrainment is principally affected by the height of boiling and by the rate of boiling. In other words, the higher the boiling and the greater the rate of boiling, at least in the standard type, the greater the amount of entrainment. Practice seems to show that it is difficult to design save-alls that are efficient as separators that do not offer excessive friction.

Several references were made by those discussing the paper to type E, which makes use of a set of baffle plates in the calandrias to give a long and gradually narrowing steam path. This type was originally developed in Europe many years ago, but was recently introduced into this country by A. L. Weber.

It will be seen in the paper that the average coefficient of heat transmission for tests 33, 34 and 35 of type E was 248, whereas the average for the standard evaporator, type A, was 197, which seems to show that the higher steam velocity and possibly the better removal of incondensable gases due to the baffle plates have resulted beneficially. It may be well to state that this is only one of a considerable number of different arrangements of baffle plates that have been used or suggested for distributing steam in calandrias. The increase of steam velocity in such an arrangement will naturally result in pressure drop, which must be overcome by means of a pump. The power required to operate such a pump, however, is not so important in connection with an evaporator as in the case of a surface condenser, for the reason that the exhaust steam from the evaporator pump can be utilized.

Mr. Orrok calls attention to the fact that all the performance curves in the paper cut the X-axis to the left of 22 in. of vacuum.

I will call his attention to the fact that the coefficients plotted were *apparent*, that is, they were determined by dividing the heat transmitted by the apparent temperature fall. In other words, the temperature fall was obtained by subtracting the saturation temperature of the steam corresponding to the pressure in a body, from the saturation temperature corresponding to the pressure in the preceding body. Due to the effect of hydrostatic head, viscosity of juice, the presence of air in steam, etc., the actual temperature fall is less than the apparent fall. As is well known, also, the difference between *actual* and *apparent* fall is greater in the last bodies. Consequently a curve representing actual temperature falls would be nearer horizontal than those given in the paper. The above will answer Mr. Loeb's question regarding the method of determining temperature fall.

The effect of juice scale, as Mr. Libermuth points out, also the effect of juice viscosity — discussed by Mr. Rousseau, are important, and while some work has been done along these lines by the author, we hope to make experiments in the future that will enable us to form more definite conclusions regarding these points than we now have.

No. 1534

THE EVOLUTION OF LOW-LIFT PUMPING PLANTS IN THE GULF COAST COUNTRY

By W. B. Gregory, New Orleans, La.
Member of the Society

It is the object of this paper to show the part that pumping machinery has had in the development of the city of New Orleans and in the reclamation of the fertile wet prairie lands of Louisiana and Texas; also to show the importance of pumping plants to the rice industry of the Southwest. To this end the history of pumping in this section will be briefly related and examples of typical pumping plants will be given.

2 Pumping plants used for drainage and those used for irrigation may both be classed as low-lift, although generally irrigation pumps elevate water through a greater lift.

3 Some parts of the world appear to have been left in a nearly perfect condition in the process of evolution, which has been working out, through countless æons, the destiny of this planet. Other sections are not so fully finished and consequently require the coöperation of man in the work of creation, in order to complete the task. Fortunately it often happens that the spots where there is most to do offer the best assurances of reward when the task is completed.

4 Tredgold, in 1827, defined Engineering as "the art of directing the great sources of power in nature for the use and convenience of man." The modern engineer who devotes his efforts to land reclamation must deal with an ever improving art in which applied science is generously mixed in the form of Mechanical, Electrical, Civil and Hydraulic Engineering. In the same way that the low lands of Holland, the Fens of England and similarly situated lands in other parts of Europe are dependent on their dikes and pumps, so

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the reclaimed lands of Louisiana and Texas must depend on their levee and pumps. These lands in their natural state produce nothing mor valuable than water grass, but when reclaimed they are classed wit the most fertile and productive to be found in the world.

5 Since the time, ages ago, when an arm of the Gulf of Mexic extended as far North as Cairo, Illinois, the mighty Mississippi an its tributaries have been building up the alluvial deposits of the delt through which it flows. Wells drilled at various points in the alluvia country have shown that the depth of the deposits is more than 300 ft. in some instances. The deposit is very fertile as it was formed fro the best of surface soil from the various states above. Usually th surface of the wet prairies, when first reclaimed, is covered with muc or humus, the decayed vegetation of hundreds of years, to a dept varying from a few inches to several feet. This material is rich i nitrogen and is a valuable fertilizer. The Southern half of Louisian contains much wet prairie land, the Eastern portion of which alluvial, while the Western part and the wet lands of the coast of Tex are the lower and poorly drained portions of the coastal plain, elevate from the bed of the Gulf during some mighty upheaval in a geologic epoch long ago. This soil is usually a silt loam with an impervio clay subsoil. It has been found especially well adapted to the cult vation of rice.

6 As the alluvial soil was deposited by the rivers, the highes land is found near the banks and there is a gradual slope from th rivers and bayous back to the swamp. Previous to the last decad the only reclaimed agricultural lands in Louisiana were in the rear o sugar plantations. The early planters cultivated the narrow strip o land along the streams, which could be drained by gravity. Th width varied greatly, but usually the distance from the levees bac to the swamp was from half a mile to two miles. The cultivation o sugar cane created a demand for more land and the demand was me by continuing the plantations in the direction of the swamps an removing the water by means of pumps from lands too low to drain b gravity.

7 About ten years ago a movement was started by Edwar Wisner to reclaim the wet prairie lands lying at or near Gulf level The work is now being carried out on an extensive scale. More tha a quarter of a million acres of agricultural lands have been reclaime or are at present in the process of reclamation in the State of Louisian

8 The work of land reclamation in this section has been of suc recent development that there is no adequate record of the magnitud

of the pumping plants, such as was furnished by the special census of 1910 for irrigation.

9 In this section it was only natural and normal that pumps used successfully in drainage work would be employed in the irrigation of rice when that industry developed. Often the lift for irrigation did not exceed that in common use in drainage plants and especially in relift irrigation pumping plants. Drainage by pumps was an accomplished fact before the irrigation of rice assumed proportions that made the industry of economic importance. Later on drainage

FIG. 1 MISSISSIPPI RIVER LEVEE IN LOUISIANA

received a fresh impetus from the reclamation of the wet prairie lands while the rice industry had attained maturity. The story of pumping plants in this section at times must therefore deal with drainage and at times with irrigation.

10 New Orleans was founded by Bienville in 1718. Looking back over nearly two centuries to the conditions that are known to have existed at that time, one is forced to wonder if the founder of the city failed to see the difficulties involved in building a city in a subtropical climate, on a narrow strip of land, along a river that was liable to overflow any year, with a swamp in the rear; or whether he had a prophetic vision of the future engineering and sanitary achievements that were to improve conditions and make the spot selected a desirable and beautiful habitation for man. At any rate, the de-

velopment of modern pumping machinery has had much to do with the improvement of conditions in this city.

11 As a matter of fact, we know that the site was chosen because on one side Bayou St. John gave an opportunity for transportation, with the small craft of that day, from the rear of the city into Lake Pontchartrain, and out through the Rigolets to the Mississippi Coast; while on the other side of the river, opposite the city, Bayou Barataria offered transportation facilities to the South through a network of bayous to the Gulf. By both these routes the city could be approached without stemming the current of the river. One of the first acts of the early colonists was to build a small levee in front of the city. The height has been raised from time to time because of increased flood heights, due largely to the confining of the upper river between levees instead of allowing it to overflow large areas.

12 In the city of New Orleans, a large portion of the surface is at or near Gulf level and an artificial water level must be maintained several feet below that of Lake Pontchartrain, which opens into the Gulf and normally is at the same elevation. Levels along the Mississippi are usually referred to "Cairo Datum," abbreviated C.D., an arbitrary system for reading elevations, so chosen that the elevations, even at the Gulf, are positive quantities. Mean Gulf level, according to the latest determination, is Elevation 20.43 C.D. In front of the city, the extreme high water of 1912 gave a gage reading of 40.13 C.D. Water is held in drainage canals of the city at Elevation 5.00 C.D., during fair weather.

13 The city is not only protected from the Mississippi on the river front by a levee, but is entirely surrounded by levees. There is one near the shores of Lake Pontchartrain, while protection levees extend from the river to the Lake both above and below the city to guard against the possible danger due to crevasses in the levees of the river above or below the city. All the run-off must therefore be pumped and some of the water is elevated two or three times before it is finally discharged into Lake Pontchartrain or through Bayou Bienvenu into Lake Borgne. (Fig. 15.)

14 The drainage of agricultural lands is quite a different problem from the drainage of a city. While it is undesirable to have ditches and canals overflow their banks on farm lands, because of the possible injury to crops, flooding at rare intervals, provided the water may be removed before serious losses result, may be justified as good engineering; while in a city all floods are to be prevented in so far as possible. In any case, the first cost of the works required to eliminate

the danger, the interest on the investment, and the depreciation and the upkeep are to be considered and balanced against the possible damage from floods and the damage which is dependent on the average period between the occurrence of unusual storms.

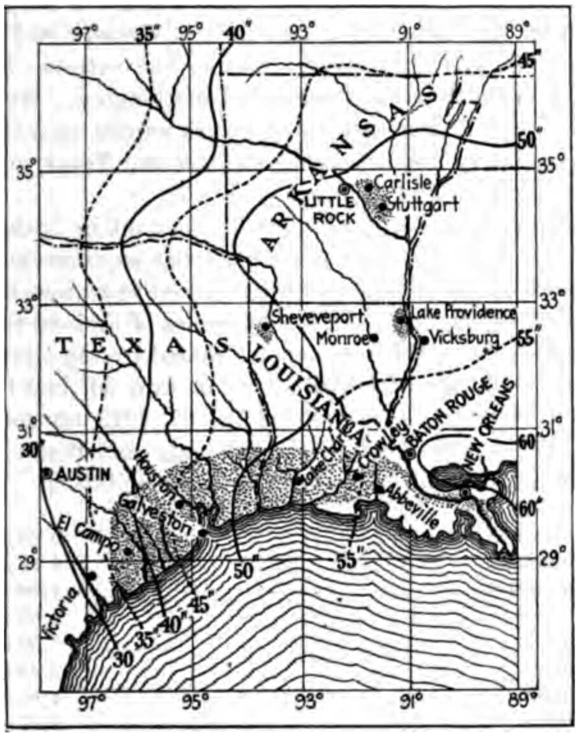


FIG. 2 DISTRIBUTION OF MEAN ANNUAL RAINFALL IN RICE COUNTRY OF LOUISIANA, TEXAS AND ARKANSAS

15 The capacity of pumping plants varies with many factors, among which may be mentioned: (*a*) rate and distribution of rainfall; (*b*) general design of collecting ditches and canals, including their storage capacity, cross-section, slope and arrangement; (*c*) nature of the territory drained. The run-off will be much greater for a city with paved streets and closely built structures than for agricultural lands. Again, the condition of agricultural lands or the kind of crop produced will affect the run-off.

RAINFALL IN LOUISIANA

16 The rainfall map of the United States shows that there is an area extending from the North shore of Lake Pontchartrain, east along the coast of the Gulf of Mexico, and far into Florida, where the rainfall is 60 in. or more per annum. This amount of annual rainfall is exceeded in the mountain region at the Western end of North Carolina and in East Tennessee; also in the extreme Northwest Pacific coast, in the States of Oregon and Washington. West of New Orleans the rainfall decreases to 50 in. per annum near the Sabine River, the boundary line between Louisiana and Texas, and falls to 30 in. in the Eastern half of Texas. (Fig. 2.)

17 Not only is the annual rainfall abundant in Louisiana, but the rate at which precipitation takes place has an important bearing on the capacity of the pumps that are required to remove the run-off from the nearly level lands. A few examples of unusual storms will make clear that the problem is one not shared by the other cities of this country. On April 25, 1907, 7 in. of rain fell in 5 hours and 8.59 in. in 12 hours. On March 22 and 23, 1912, unusual rainfall occurred. The records kept by the Sewerage and Water Board of New Orleans showed the following rates of precipitation:

5 minutes	0.68 in.
8 minutes	1.09 in.
10 minutes	1.38 in.
15 minutes	1.69 in.
20 minutes	2.00 in.
1 hour	2.48 in.
2 hours	4.66 in.
2¼ hours	5.07 in.
Total, 2 days	7.23 in.

18 The United States Weather Bureau records at the New Orleans station show that there have been 57 storms in the past 26 years, during which the precipitation in 24 hours exceeded 3 in. These storms are classified as to their intensity as follows:

57 rains exceeding 3 in. in 24 hours.
25 rains exceeding 4 in. in 24 hours.
10 rains exceeding 5 in. in 24 hours.
5 rains exceeding 6 in. in 24 hours.
3 rains exceeding 7 in. in 24 hours.
3 rains exceeding 8 in. in 24 hours.
0 rains exceeding 9 in. in 24 hours.

The maximum rainfall in 24 consecutive hours recorded at New Orleans was 9.22 in. The storm occurred April 7 and 8, 1883.

19 While the normal rainfall for New Orleans is less than 60 in., the rainfall that sometimes occurs in an unusual season is well illustrated by the observations of the year 1912 when the rainfall amounted to 81.50 in., as reported by the U. S. Weather Bureau. Fig. 3 shows the normal rainfall by months at the end of 1912 and the rainfall for

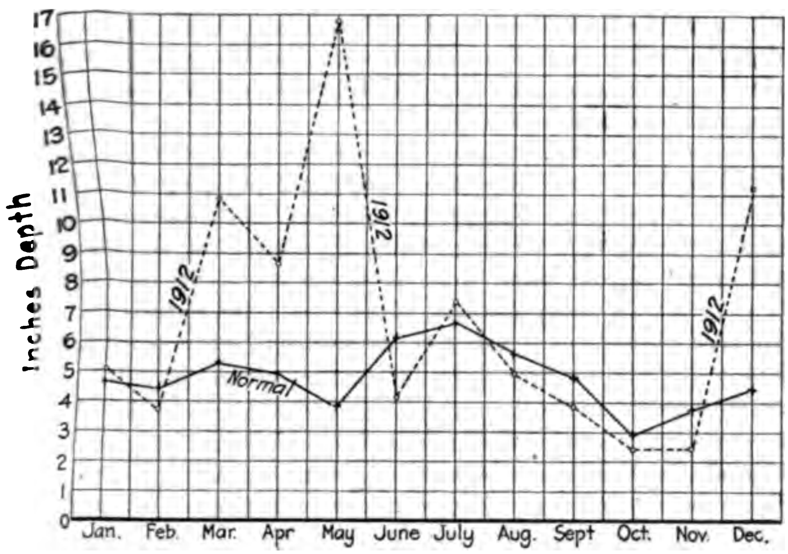


FIG. 3 RAINFALL AT NEW ORLEANS

that year. It will be noted that the normal rainfall, which is typical of that for a large portion of the State and especially for the reclaimed lands to the south and southwest of New Orleans, is remarkably well distributed and that the growing season of June, July and August receives the maximum rainfall.

DRAINAGE COEFFICIENT

20 Run-off is usually stated in terms of a drainage coefficient, or the depth in inches of water, if spread over the entire area to be drained, removed in 24 hours.

21 In England and Holland the annual rainfall is less than in the Gulf Coast country of the United States and what is more important in deciding the capacity of pumping plants for drainage, the intensity

of rainfall is much less in unit time in the European countries. For the Fens of Eastern England, with the annual rainfall 22 to 27 in., capacity in pumping plants equivalent to a drainage coefficient of $\frac{1}{4}$ in. has proved satisfactory, while in Western England where the annual precipitation is approximately 50 in. the drainage coefficient is increased to $\frac{3}{4}$ in. In Holland, Haarlem Lake is drained by pumps having a drainage capacity of $\frac{3}{8}$ in.[1]

22 In the Upper Mississippi valley and along the Illinois River, a drainage coefficient of $\frac{1}{4}$ in. has been found satisfactory. In the later plants, however, there has been a tendency towards greater capacity and pumping plants recently installed have been designed for a drainage coefficient of $\frac{3}{10}$ in.

23 In Louisiana, it is only in recent years that a drainage coefficient has been worked out. The older pumping plants varied widely in capacity according to the ideas of the parties installing the machinery. In 1909 Drainage Investigations, U. S. Department of Agriculture started some work in Louisiana along the following lines.[2] "(1) To study the soil, climate, and the natural condition with special reference to the drainage problems encountered and the value of the land for agricultural purposes when successfully drained. (2) To collect such technical data and to examine such details of present practice as will afford information of value to land owners and especially to engineers interested in the reclamation of such lands. (3) To disseminate in available form the results of the investigations and to encourage land drainage by emphasizing the benefits to be derived from bringing such lands under cultivation."

24 This work, inaugurated by Arthur M. Shaw, Drainage Engineer, and the writer, has been continued since 1909 by Chas. W. Okey, Drainage Engineer, and much information of value has been collected and made available.

25 The area of the individual drainage districts varies from a few hundred acres in the smaller districts to several thousand acres in the larger ones. The pumping capacity in 24 hours was found to vary from less than 0.9 to 1.51 in. These figures represent average practice in Louisiana and are being used as a guide in deciding the capacity of drainage pumping plants for agricultural lands.

26 Mr. Okey sums up the records of rainfall and run-off from four drainage districts of Louisiana, as follows: "In the year 1914

[1] Engineering for Land Drainage, Elliott.

[2] The Wet Lands of Southern Louisiana and Their Drainage, U.S. Dept. of Agr., Bulletin 71, by Charles W. Okey.

the results from the Poydras, Des Allemands, Jefferson, and Raceland plants may be said to be typical. In addition, the results from the last-named district for 1913 are also typical. An average of the records for 1914 for each of the districts and including the records for 1913 from the Raceland district, gives the following results:

Rainfall	50.85 in.
Run-off	22.59 in.
Days per year on which operation occurred	104
Number of times per year boilers were fired	97
Number of 24-hour days per year at full capacity	21.1
Average number of hours run at full capacity per fire-up	5.2

27 "While the above rainfall is slightly below normal for this section of the State, it is believed that the percentage of rainfall appearing in run-off in the above average will not be far from normal. With the mean annual rainfall of 56 in. and the same percentage, the amount to be pumped would be 25 in. per year. Therefore a plant with a 24-hour capacity of 1.25 in. in depth of water over the area drained would have to operate an average of about 20 days of 24 hours each, per year. Where such factors as leaky levees enter in, the amount of pumping would of course be greatly increased. Where the capacity of the plant is much less than specified above, the amount pumped would be somewhat less, due to evaporation prior to pumping."

COST OF OPERATING DRAINAGE PLANTS

28 It must not be thought that the large rainfall and run-off impose too great a burden in the cost of pumping water from the reclaimed wet prairie lands.

29 Information has been collected and compiled by C. W. Okey in regard to the cost of operating several drainage pumping plants for periods varying from two to eight years and for a variety of types in equipment. The tabulated costs include fuel, labor, repairs and incidentals. The figures indicate that the cost per acre per year will range from $0.50 to $1.25, depending on the sort of machinery used, the cost of fuel and the peculiarities of the season. For those using steam plants he finds the cost of fuel per acre-foot, lifted one foot, less with Corliss engines than with slide-valve engines.

30 For average seasons, in each case, the depth of water to be removed from land in drainage is less than the depth of water pumped

onto the land in rice irrigation and the average lift in drainage is much less than that in irrigation.

THE DRAINAGE WHEEL

31 The first pumps to be used in the Gulf Coast country for artificial drainage were undoubtedly of the "drainage wheel" or "scoop wheel" type. The latter term is misleading, for it would seem to imply that the paddles were made in the form of scoops, which is not the case. The paddles are usually straight boards, inclined back at the periphery of the wheel. This type of pump has long been used in Europe, and according to W. H. Wheeler[1] was used by the Romans in the South of Spain nearly two thousand years ago.

FIG. 4 SMALL DRAINAGE WHEEL

32 Starling states that the windmill has been used in Holland since 1308.[2] It was greatly improved in 1573 when the discovery was made that it was not necessary to turn the whole mill, but only the top. The windmills furnished power to operate scoop wheels and, at a later date, to drive screws made on the principle discovered by Archimedes. In some cases in Holland where scoop wheels are used, two or three successive lifts are necessary. As far as the writer

[1] The Drainage of Fens and Lowlands by Gravitation and Steam Power, 1888.
[2] Drainage Pumps of Holland, Trans. Am.Soc.C.E., vol. 26, 1892, p. 622.

knows the screw of Archimedes has never been used in this country, nor has the windmill been used to any extent as a source of power for drainage or irrigation pumps. However, small drainage wheels have been operated by horse or mule power, one of which is shown in Fig. 4.

33 There are many of the steam-driven drainage wheels yet in use in Louisiana. Some are of venerable age, while others date from a comparatively recent period. Many of the old drainage wheels of the sugar plantations worked between brick piers that were substantially constructed. As one travels about through the lowlands of Louisiana today the foundations and flumes of these old wheels are frequently seen. Some of the later installations used foundations of wood, resting on long piles. The cost of foundations and the difficulty of holding them rigidly in most localities has increased the cost of the drainage wheels to such an extent that they have been practically eliminated from competition with other cheaper forms of pumping plant.

34 Another point against the drainage wheel is the difficulty involved in adjusting its height. Once set, the depth to which the water may be lowered is definitely fixed. As a rule these wheels are expected to pump against a maximum head equal to one-fourth the diameter. The large steam-driven wheels usually range from 28 to 32 ft. in diameter with a width of from 4 to 7 ft. The humus of the reclaimed land in time disappears as the land is cultivated and the level of the land seems to fall. The amount of shrinkage varies with the conditions as the depth of humus is not uniform. As a result, after a few years it has been found desirable to pump to a lower level, and with a drainage wheel it requires either a lengthening of paddles or the lowering of the foundations and power plant.

35 Even with the high cost of the drainage wheel and the difficulty of lowering it as the land subsided, it has been a favorite with many sugar planters because of its reliability and ease of operation. Usually the engines to drive drainage wheels are of the old, long-stroke type, of ample power even with low steam pressures. The boilers in some cases are those rejected from the sugar houses and only capable of carrying a very moderate pressure. The attendant is often a dusky laborer with no great claim to skill.

TEST OF A DRAINAGE WHEEL

36 The writer made a test of a drainage wheel in 1905 used to drain the sugar plantation of the South Side Planting Co., on the right

bank of the Mississippi River, opposite New Orleans. The plantation contained 1700 acres, 1600 of which were under cultivation in 1905. Fig. 5 is from a photograph of this wheel.

37 Open ditches were used exclusively. The smaller ones were brought together successively and terminated in a large canal leading to the drainage wheel. The water drained away from the river. The flow was obtained by deepening the ditches as they approached the

FIG. 5 DRAINAGE WHEEL, SOUTH SIDE PLANTING COMPANY

pumping plant, as well as from the natural slope of the land. Here the water was elevated and then flowed back into the swamps. The plantation was protected from backwater by means of a levee.

38 The general design of the wheel and method of bracing are clearly shown in Fig. 6. It is a type of its class, but has some distinct features in the double gearing and in the number of paddles. Care

was exercised so to design the wheel that the water would not be lifted unnecessarily. The backward flow through the pump wheels when at rest was prevented by the swinging door shown in Fig. 6.

39 The diameter of the wheel is 28 ft., and the width 6 ft. It is driven by a simple non-condensing engine of the slide-valve type, having a cylinder 16 in. in diameter and a stroke of 24 in. The steam pressure was 40 lb. or less and wood was ordinarily used as fuel.

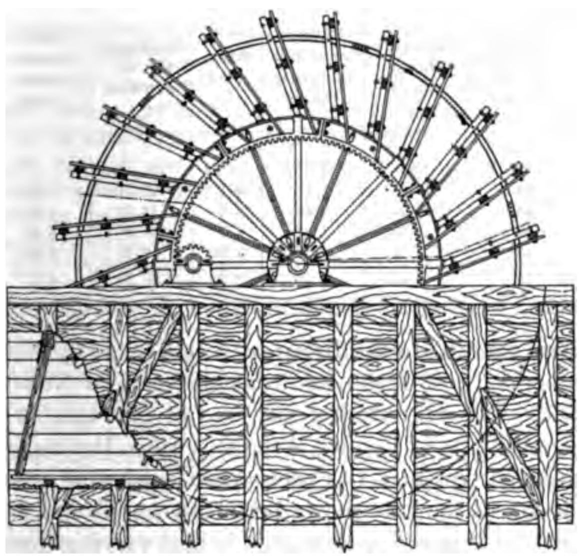


Fig. 6 Drainage Wheel, South Side Planting Company

40 The test was made while pumping out the canal system and was necessarily short, lasting only about an hour. A boiler test, under such conditions, would have been worthless and was not attempted.

41 The method of testing consisted in traversing the discharge flume with a current meter and taking indicator cards and other observations as quickly as possible after traverse was finished. By this means the indicated horsepower was a little less than the mean corresponding to the water measurement, but as the latter required only about ten minutes, the error was not great. This method was rendered necessary by lack of observers.

42 The results given in Table 1 are very satisfactory, as they show an efficiency of engine, transmission gears, and pump in every case exceeding 38 per cent, and in two cases considerably above that figure, while the actual lift of the pump varied from 2.4 ft. to 2.86 ft. During the last observation the paddles dipped into the water to a depth of approximately 1 ft., and the slip or backward flow was quite large. The clearance on the sides of paddles was about ¾ in.

TABLE 1 ENGINE AND PUMP TEST, SOUTH SIDE PLANTING COMPANY DRAINAGE WHEEL

Steam Pressure, Pounds	Revolutions per Minute of Engine	Indicated Horsepower	Revolutions per Minute of Wheel	Head, Feet	Discharge, Cu. Ft. per Sec.	Useful Water Horsepower	Efficiency, Per Cent
40	61	13.43	2.00	......	....	....	....
40	66	12.61	2.17	2.4	20.71	5.59	44.3
38	68	10.37	2.34	2.8	17.20	5.41	52.2
36	67.5	8.80	2.22	2.7	11.23	3.41	38.8
37	68	6.95	2.24	2.86	8.21	2.66	38.3
Mean 38.2	66.1	9.68	2.22	2.69	14.34	4.27	43.4

43 These results are confirmed by a test made by W. M. White, Mem.Am.Soc.M.E., of a similar drainage wheel in New Orleans in August, 1900. The wheel tested was used at that time in one of the city drainage stations at London Avenue. Since the inauguration of the new drainage system it has been taken down and removed.

44 The log of the test shows that between 50 and 60 cu. ft. of water per sec. were pumped through a height varying from 4 to 5 ft. The efficiency of engine, gearing and pump ranged from 45 to 50 per cent. The duty per 100 lb. of coal was approximately 13,000,000 ft-lb.; the water rate of the engine 50.5 lb. per i.h.p-hr. The engine was of the type used in Mississippi River steamboats; diameter of cylinder 18 in., length of stroke 54 in. During the test the engine made about 35 r.p.m.

45 An installation of drainage machines of this type in Holland is fully described and the results of tests given in another publication.[1]

46 A bulletin[2] of the University of Wisconsin gives the results of researches in the hydraulic laboratory in testing a "flash" wheel 8 ft. in diameter and 2 ft. in width. Efficiencies of wheel were found to be more than 70 per cent in some instances.

[1] Engineering News (1910), No. 20, p. 581.

[2] Bulletin of the University of Wisconsin, No. 598, 1914.

EARLY DRAINAGE EQUIPMENT OF NEW ORLEANS

47 The drainage wheel was used extensively in this city previous to 1900. The drainage equipment of New Orleans before the inauguration of the present system was described[1] by the Advisory Board, the engineering members of which were B. M. Harrod, Henry B. Richardson, and Rudolph Hering, Mem.Am.Soc.M.E. The City Engineer was L. W. Brown. The following paragraphs are quoted from the Report:

48 The existing system of drainage is composed, as stated in the body of the report, of open canals receiving the water delivered from the higher portions of the city by the small street gutters, and conveyed by these open canals to the draining machines, which deliver the same into Lake Pontchartrain. These draining machines are four in number, and are located in the bottom of the basin between the river and the Metairie and Gentilly Ridges, excepting the London draining machine, which is located on Gentilly Ridge.

49 The Dublin draining machine is located at the intersection of 14th and Dublin streets, and has a maximum capacity to deliver 480 cu. ft. of water per sec., with a lift of 5 ft. This machine is composed of two wheels, one being 34 ft. in diameter and 6 ft. face and the other being 34 ft. 4 in. diameter, with a face of 5 ft. 9 in.

50 The Melpomene draining machine is located at the intersection of Claiborne and Melpomene streets, and has a maximum capacity to deliver 150 cu. ft. of water per sec., with a lift of 5 ft. This machine has only one wheel, which is 35 ft. in diameter, 4 ft. 6 in. face.

51 The Bienville draining machine is located at the intersection of Hagan Avenue and Toulouse Street, and has a maximum capacity to deliver 240 cu. ft. of water per sec., with a lift of 5 ft. This machine consists of two draining wheels, one being 28 ft. 6 in. in diameter, 4 ft. 4 in. face, and the other 34 ft. in diameter and 7 ft. face.

52 The London draining machine is located at the intersection of London Avenue and Gentilly Ridge, and has a maximum capacity to deliver 300 cu. ft. of water per sec., with a lift of 5 ft. This machine consists of two draining wheels, each of which is 35 ft. in diameter and 4 ft. 10 in. face.

53 There is located at the intersection of Bayou St. John and Orleans Streets a centrifugal pump delivering into the lake through the same tail race as the Bienville machine. This pump has a maximum capacity of 44 cu. ft. of water per sec., with a lift of 5 ft.

54 The draining machinery, with the exception of the Orleans pump, has been in service for upwards of forty (40) years, and is consequently primitive and not economical in its operation.

55 Considering the matter from every point, our present situation as to drainage in conjunction with the topographical and hydrographical conditions, renders the formulating of a thoroughly efficient and comprehensive system of drainage for the city a unique and intricate problem, and, perhaps, is unparalleled

[1] Report on the Drainage of the City of New Orleans, 1895.

in this country or Europe, and the solving of the problem renders absolutely necessary a most careful and rigid investigation into all of the conditions bearing on the subject.

56 The combined capacity of the drainage wheels was 1170 cu. ft. per sec. with a lift of 5 ft. The one centrifugal pump then in use brought the total amount up to 1214 cubic feet per second. However, the accuracy of the capacity rating of these wheels is in doubt. While a capacity of 300 cu. ft. per sec. with a lift of 5 ft. was claimed for the two London Avenue machines or 150 cu. ft. per sec. for each, the test made by W. M. White in 1900 gave between 50 and 60 cu. ft. per sec. with a lift of approximately 5 ft. However, it is possible that the machinery might have been speeded up and the capacity considerably increased.

THE CENTRIFUGAL PUMP

57 So far as the writer has been able to learn, the first centrifugal pumps manufactured in the State of Louisiana were made by John Clark, a prominent foundryman, in the early fifties. It was similar in design and construction to the Palmer pump which had been used with varying success about that time in drainage work. The origin of the Palmer pump is uncertain.

58 Both pumps were decidedly crude as there was very little machine work done on them, and all joints were made with red lead excepting that between the shells, which was composed of sal ammoniac and cast-iron borings. The vertical suction extended through the bed plate and great difficulty was experienced in making a satisfactory joint with the suction pipe.

59 The centrifugal pump in some of its various forms has proved to be a favorite for the work of irrigation and drainage where large volumes of water must be elevated through only a few feet lift. In Fig. 7 are diagrams of two of the early types of centrifugal pumps.

60 There are many reasons for the popularity of the centrifugal pump, among which may be mentioned low first cost, reliability of operation, and simplicity of construction. If properly designed for the conditions under which it is to operate, it is efficient. Twenty-five years ago efficiencies of 50 to 60 per cent were considered good practice where the lift was moderate. Today many different centrifugal pumps may be found in irrigation work that have shown efficiencies of more than 80 per cent under carefully conducted tests. Drainage pumps of the centrifugal or screw type or a combination of

both have developed efficiencies on lifts less than 10 ft. that were thought impossible a few years ago.

61 It has also been realized that the design of the pipes to conduct the water from the suction basin to the pumps and from the pumps to the discharge basin or flume is well worthy of careful consideration. A few years ago it was a common practice to use straight suction and discharge pipes having the same diameter throughout. In recent years examples of this kind of ignorance of the fundamental laws of hydraulics have been uncommon in the Gulf Coast country, although they are found in parts of the Mississippi valley and occasionally descriptions of such plants find their way into the technical press.

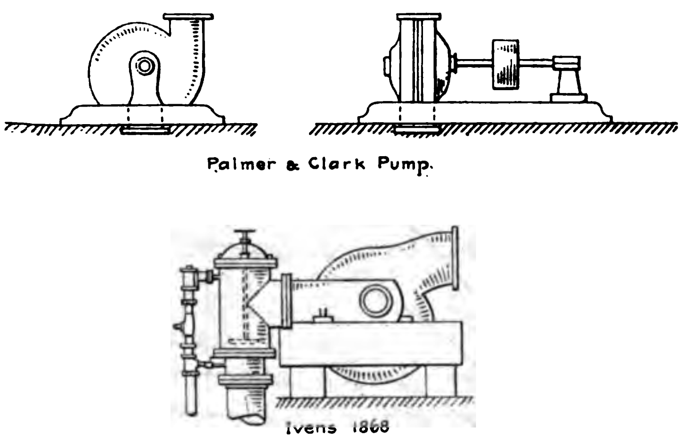


FIG. 7 TYPES OF CENTRIFUGAL PUMPS

62 In drainage installations where the lift is usually between 4 ft. and 10 ft., the losses at the entrance of suction pipes and the kinetic energy that may be thrown away at the end of the discharge pipe together make up a large percentage of the energy required. The entrance loss in feet of water is usually estimated at 0.93 of the velocity head at the entrance. The discharge loss is equal to the velocity head at the end of the discharge pipe. If the pipes are round and the diameter is doubled at suction and discharge end the area is multiplied by four, and if pumping a constant quantity of water there will be entrance and discharge velocities one-fourth as great as with straight pipes. Losses vary as the square of the velocity so they will

be reduced to one-sixteenth of the loss of a straight pipe, if the diameter of the ends of suction and discharge pipe are gradually enlarged to twice the diameter of the straight pipes, or if not round in cross-section, the area is increased to four times that of the original pipe.

63 The centrifugal or screw pump as ordinarily constructed with suction and discharge pipes may be placed at a level above the

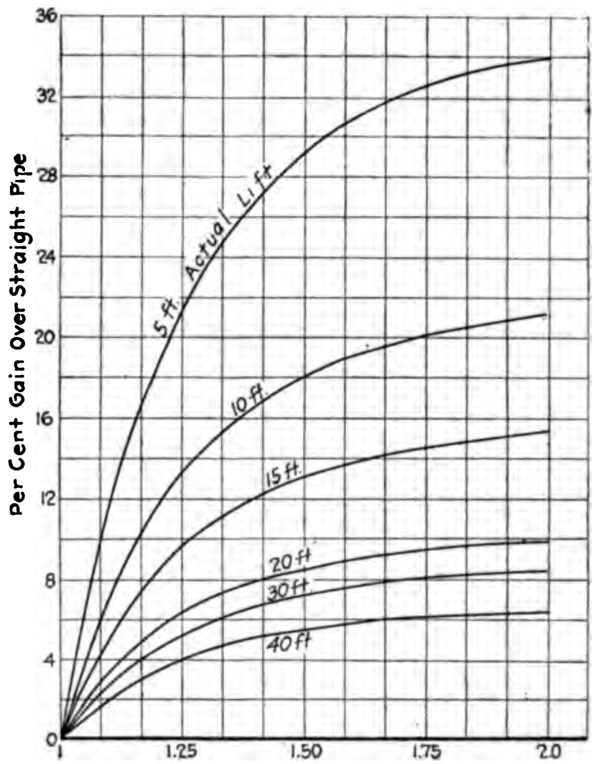


FIG. 8 GAIN FROM EXPANDING SUCTION AND DISCHARGE PIPE

discharge so that it is easy to install and to repair. The suction pipe and the discharge pipe, together with the pump, form a siphon. Variations in level in suction and discharge sides do not affect the pump and the lift is always equal to actual difference of level, while the head the pump must develop is the lift plus the various friction losses in the pump and piping.

64 The importance of the problem is illustrated by the accompanying curves, Fig. 8. It is assumed that the size of pump flange

is that for a 2-ft. diameter of discharge and that the mean velocity in a 2-ft. pipe is 10 ft. per sec. Up to 15 ft. actual lift, the length of straight pipe is taken as 15 ft., while for lifts greater than 15 ft. the length of straight pipe is equal to the actual lift. The following table is a sample of the calculations from which the curves are constructed:

TABLE 2 CENTRIFUGAL-PUMP CALCULATIONS FOR 5-FT. LIFT

Kind of Pipe	Loss of Head at Entrance $0.93\ V_s^2/2g$	Loss of Head at Discharge $V_d^2/2g$	Friction Loss in Straight Pipe $fV^2/d\,2g$	Total Head to be Produced by Pump	Gain	
					Ft.	Per Cent
Straight, 2 ft. diam.	1.44	1.55	0.24	8.23	0	0
Expanded to 2 ft. 6 in.	0.59	0.63	0.24	6.46	1.77	21.5
Expanded to 3 ft. 0 in.	0.28	0.31	0.24	5.83	2.40	29.1
Expanded to 4 ft. 0 in.	0.09	0.10	0.24	5.43	2.80	34.0

THE IVENS PUMP

65 In the late sixties the Ivens pump was invented by Edmund M. Ivens of New Orleans, grandfather of E. M. Ivens, Junior Member of this Society. It was mounted on a heavy wooden frame, and was readily installed over the intake or suction water and discharged through a flume bolted to the discharge.

66 The pump was fitted with a surface valve to which was connected a steam jet. The jet discharged through a check valve, thus priming the pump. During the process of priming, the connecting valve on the lower side of jet was opened slightly and vacuum established between valve and suction water. When in operation the valve rode the suction column, thus permitting the full area of suction opening. The pump was properly finished and on account of the results obtained is prominently identified with the drainage work of that period, in this section. Later on it was used extensively in rice irrigation.

67 The Ivens pump was made in a variety of forms and sizes. Vertical-shaft iron pumps had double suction openings. The smaller sizes had a single suction pipe but cored passages around the outside of the volute of the pump so that water entered the impeller from both sides. In the largest size independent suction pipes were used. The writer tested several Ivens pumps during the irrigating season of 1905 and the results from two different plants here follow.

TEST OF ABBOTT-DUSON CANAL, FIRST RELIFT[1]

68 This test was made by the writer of the pumping plant forming the first relift of the Abbott-Duson canal system. The plant is located at Egan, La., about 2½ miles east of the main pumping plant.

69 There are two Ivens pumps rated as 36-in., but the lift is so small that the pumps each discharge through a rectangular opening into a separate flume, having gradually expanding cross-sections. The two flumes are brought together at a distance of about 50 ft. from the pumps into a larger flume, which discharges into the canal beyond the plant. Fig. 10 shows this installation.

70 The boiler equipment consists of three horizontal return-tubular boilers, 72 in. in diameter by 18 ft. long, each containing seventy-two 4-in. tubes. Crude petroleum is used as fuel.

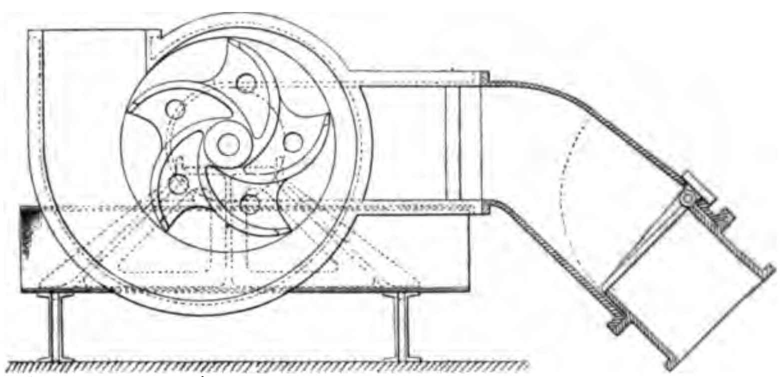


FIG. 9 IVENS CENTRIFUGAL PUMP, HORIZONTAL SHAFT

71 The two pumps are driven through the medium of a rope drive by a simple condensing Corliss engine, 24-in. diameter of cylinder and 48-in. stroke. The flywheel is 16 ft. in diameter and has 18 rope grooves. There are 1850 ft. of 1¼-in. rope required for the drive.

72 Under ordinary conditions an open heater is used. A direct-acting steam pump furnishes water to the heater, and a similar pump takes the water from the heater and delivers it to the boilers. The heater receives the exhaust from these two pumps and also from the condenser pump. During the test the heater was not used, as the water had to be measured. The piping was changed so that one of the pumps furnished water to fill two calibrated barrels, so placed that

[1] Bulletin 183, O.E.S., U.S. Dept. of Agri., 1907.

they could be emptied through a 2-in. valve into a lower barrel. The suction of the second pump was attached directly to the lower barrel, and this pump was used to feed the boilers.

73 The measurements of the water discharged by the pumps were made by the pitot tube. The discharge flume in which the measurements were made was 18.75 ft. wide. The depth of water

FIG. 10 VIEW OF PUMP AND FLUME, ABBOTT-DUSON FIRST RELIFT

varied from about 1.5 ft. to a little over 1.6 ft. The cross-section was divided into 20 rectangles of equal size and the mean velocity obtained at the center of each rectangle, or, in other words, the velocity was observed at ten different stations across the flume and at two different depths at each station.

74 The crude petroleum used for fuel during the test was measured in a calibrated barrel, the amount per hour being 712 lb. The following day fuel oil was measured for 1 hr. and 57 min., the feedwater heater being in use. It was found that the consumption of oil per hour was 603 lb. The temperature of water entering boiler was 200 deg. fahr. instead of 92 deg. fahr., as found when heater was not used. The theoretical gain by using the feedwater heater is about 11 per cent, while the actual difference in fuel used amounted to nearly 20 per cent. The discrepancy is accounted for by the difference in the amount of water present in the fuel oil. For example, during the test

on July 31 the amount of water in the fuel oil was sufficient to put out the fires momentarily on two occasions, and to require careful oversight of the oil burners to prevent irregularities in the amount of combustion and, consequently, in steam pressure. On the second day the oil in the supply tank had become quite thoroughly separated from the water, the latter having settled to the bottom, and the result was that there was no trouble with the burners.

FIG. 11 IVENS PUMP AT ACADIA CANAL RELIFT

75 During the test, indicator cards were taken at 15-min. intervals, and observations were made of steam pressure, vacuum gage, revolutions of engine and pump, and of head pumped against. At intervals of a half-hour water measurements were taken in the flume, and the temperature of water, oil and air was noted. The amount of water and fuel oil used was also carefully noted.

76 The efficiency of engine, transmission and pump is excellent — in fact, the best of any of the plants tested in 1905 in which centrifugal pumps are used. This efficiency had an average value of 64.2 per cent. If the efficiency of the rope drive is assumed to be 95 per cent and the mechanical efficiency of the engine as 90 per cent, the efficiency of the pump is found to be about 75 per cent.

77 The results of the tests are as follows:

Boiler Test No. 3, First Relift, Abbott-Duson Canal, July 31, 1905

Duration of test, 4 hours.
Total fuel oil, 2886 lb.
Average steam pressure by gage, 65.6 lb. per sq. in.
Average temperature of feed water, 92 deg. fahr.
Factor of evaporation, 1.157.
Total weight of water fed to boiler, 28,629 lb.
Equivalent water evaporated from and at 212 deg. fahr., 33,112 lb.
Boiler horsepower, 240.
Average temperature of fuel oil, 171 deg. fahr.
Average air temperature, 92 deg. fahr.
Water apparently evaporated per pound of oil, 9.92 lb.
Equivalent evaporation from and at 212 deg. fahr. (not corrected for quality of steam), 11.47 lb.
Total feed water (including steam used by auxiliaries) per indicated horsepower-hour, 31.2 lb.

TABLE 8 ENGINE AND PUMP TEST, ABBOTT-DUSON FIRST RELIFT

Steam Pressure, Pounds	Vacuum Gage, Inches	Revolutions per Minute of Engine	Indicated Horsepower	Revolutions per Minute of Pump	Head, Feet	Discharge, Cu. Ft. per Sec.	Useful Water Horsepower	Efficiency, Per Cent
52	22	66.0	224.8	117	11.06	....	....	....
68	21	66.0	240.7	117	11.04	....	....	....
65	21	66.5	223.4	118	11.07	....	....	....
66	22	67.0	247.8	119	11.12	....	....	....
68	23	67.0	240.3	119	11.17	118	149	62.0
67	23	67.0	233.5	119	11.19	....	....	....
65	23	67.0	234.1	119	11.22	116	147	62.9
65	23	66.5	221.6	118	11.25	....	....	....
67	23	66.5	228.2	118	11.25	114	145	63.6
65	23	67.0	225.5	119	11.24	....	....	....
68	23	67.0	227.4	119	11.31	114	146	64.2
67	23	67.0	236.1	119	11.36	....	....	....
68	23	66.8	228.5	118	11.33	115	147	64.4
68	23	67.0	225.6	119	11.34	....	....	....
67	23	66.5	227.0	118	11.33	117	150	66 1
67	23	67.0	223.3	119	11.30	....	....	....
67	23	67.0	219.4	119	11.36	114	146	66 5
Mean 65.6	....	66.8	229 8	118.5	11.24	115.4	147.1	64.2

Duration of test, 4 hours.

TEST OF ACADIA RELIFT[1]

78 The Acadia relift is located on the main Acadia Canal, about a mile north of Iota, La., and was tested by the writer in August, 1905. The pump is nominally an Ivens 36-in. pump, similar to those at the Abbott-Duson first relift, and having two suction pipes 24 in. in diameter. (Fig. 11.)

79 The equipment of this plant includes two horizontal return-tubular boilers 72 in. in diameter by 18 ft. in length, each containing seventy-two 4-in. tubes. Fuel oil is used. The boilers are fed by means of two direct-acting steam pumps. One pump takes water from the canal and furnishes it to the open heater. Another pump takes the water from the heater and pumps it into the boiler. Both pumps and the engine exhaust into the heater.

80 The engine is a simple non-condensing Corliss, cylinder diameter 22 in., stroke 42 in. The pump is driven by rope drive. There are 10 grooves in the engine flywheel and 958 ft. of 1½-in. rope are used.

81 The discharge from the pump was measured in the flume about 50 ft. from the pump. At this place the flume had a uniform cross-section 9.27 ft. wide and about 1.8 ft. deep. A current meter was used and the cross-section slowly traversed at three different depths to obtain the mean velocity in all but two observations, when the pitot tube was used. With the latter instrument the velocity was observed at ten different stations across the flume and at three different depths in the first observation and at two different depths in the second.

82 The arrangement of the plant is similar to that of the Abbott-Duson first relift, except that there is only one engine and one pump instead of one engine and two pumps, as in the plant referred to. There was one engine and a rope drive in each case. Both the engine and the rope drive were larger in the case where two pumps were used, but the loss due to friction in the two cases probably was not very different. The height through which the water was lifted was a little greater with the two pumps than with the single pump of the Acadia relift, and this probably had some effect on efficiency.

83 It was practically impossible to make a complete boiler test because the necessary changes in the piping could not be made for measuring the feed water. Fuel oil, however, was measured by means of a calibrated barrel, and a partial boiler test of 4.45 hours'

[1] Bulletin 183, O.E.S., U.S. Dept. of Agri., 1907.

duration was run, during which time the average steam pressure by gage was 79.4 lb. per sq. in and 1567 lb. of fuel oil were consumed. The data of the engine and pump test are given in Table 4.

TABLE 4 ENGINE AND PUMP TEST, ACADIA RELIFT

Steam Pressure, Pounds	Revolutions per Minute of Engine	Indicated Horsepower	Revolutions per Minute of Pump	Head, Feet	Discharge Cu. Ft. per Sec.	Useful Horsepower	Efficiency, Per Cent
80	68.5	136.0	117	....	....	....	....
80	69.5	144.8	119	9.53	70.6	76.1	52.6
79	68.7	139.4	117	9.55	71.1	76.7	55.0
78	68	128.6	115	9.48	69.6	74.6	58.0
80	69	138.4	118	9.50	73.7	79.1	57.1
80	68.5	136.8	118	9.51	67.8	72.8	53.2
79	68.7	139.4	117	9.45	75.0	80.1	57.5
78	68	132.7	116	9.46	73.0	78.0	58.7
81	70	143.3	118	9.45	70.2	74.9	52.3
Mean 79.4	68.8	137.7	117.2	9.49	71.4	76.5	55.6

Duration of test, 4 hours.

84 The Ivens pump was made in a special form for low lifts. A wooden box was used for the body of the pump. Into the box the impeller discharged its water through two or four expanding nozzles. The impeller, the vertical shaft and the parts surrounding the impeller, including the cast-iron nozzles, were made of metal, while the remainder was constructed of wood. Figs. 12 and 13 show some of the details of this pump, which was especially adapted to drainage where low lifts prevail.

THE MENGE PUMP

85 The Menge centrifugal pump was brought out by the late Joseph Menge, to whom patents were issued in 1888 and 1891. S. L. Menge, son of Joseph Menge, patented in 1910 an improvement in the number and area of passages by which the water enters the pump.

86 As ordinarily constructed, the body of the pump is of wood, although concrete has occasionally been used. The shaft is vertical and the impeller is submerged in the water to be pumped. The openings in the bottom of the pump box were arranged to admit water both above and below the impeller. A ball bearing at the top of the shaft takes the weight of the moving parts. Aside from the shaft,

impeller and bearings the pump is entirely of wood and can be easily and cheaply erected. As the water pours over the top, the quantity of water pumped is impressive.

87 The pump is made with impellers varying in size from 8 in. diameter by 4 in. high to 60 in. diameter by 20 in. high, and in rated capacities from 833 gal. per min. to 60,000 gal. per min. Either a belt or rope drive is used to connect the pump with the source of power. Sectional views of the Menge pump are shown in Fig. 14.

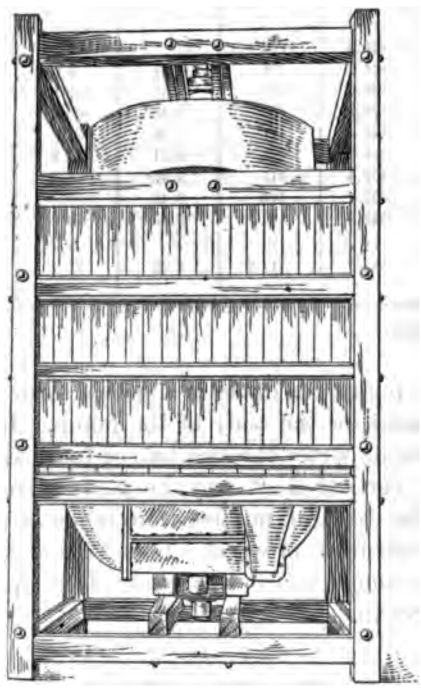


Fig. 12 Ivens Quadruple Discharge Pump, Box Type

88 A single impeller is used for low lifts, while for higher lifts two or three impellers are arranged at intervals on the same shaft. The pump was justly popular from the time of its inception because of its cheapness and its ability to handle large quantities of water at good efficiencies, especially at low lifts. In the drainage of sugar plantations this pump found a particularly favorable field and it has been used quite extensively in rice irrigation.

89 A model pump exhibited at the Chicago Exposition in 1893

attracted a great deal of attention and received a medal. The wording of the award is given below:

90 For centrifugal pump is of very simple design and construction, adapted especially to raising large volumes of water for irrigating, flooding rice fields, and various other useful purposes.

91 The wheel is disposed horizontally on the lower end of a shaft below water line and is supplied with double suctions, one from underneath center of wheel and the other from top, so that the suctions balance and prevent downward pressure on wheel. The top cover of the upper suction prevents the vertical column of water from having any downward pressure on wheel.

92 The wheel is set in a large square wooden box, which serves as the discharge pipe and the water is thrown out radially from the wheel where it has ample room to ascend.

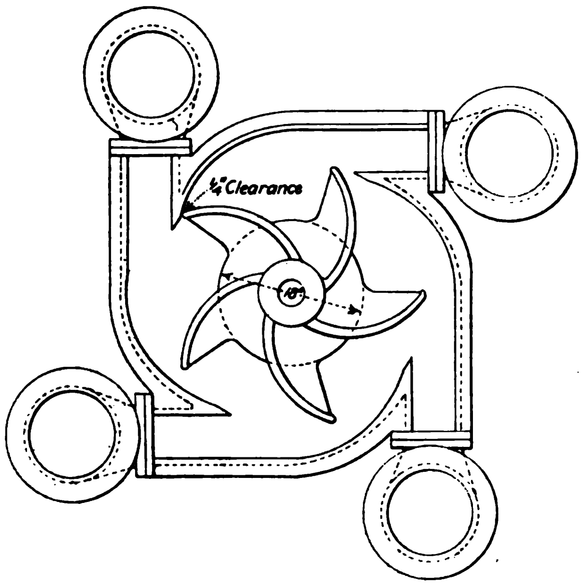


FIG. 13 DIAGRAM OF CASING AND IMPELLER, IVENS QUADRUPLE PUMP

93 The weight or resistance of the ascending column is horizontal in all directions against the outer side of the blades, the weight or force on one side being counteracted by that on the other side without causing weight or friction on wheel or bearings. The pump has neither suction nor discharge valves, and being set below the water line is always primed.

94 The working parts, consisting of water wheel, shaft and pulley, are of iron, the frame and walls of wood. The top bearing carries the weight of wheel and can be adjusted so as to set the wheel midway between the suctions. In operation it is a free wheel in a square box, between the top and bottom suction ports, with free and continuous delivery of water from each and all of the blades of

the wheel into the box. The balance suction relieves the journals of strain and friction that would otherwise be due to the force of suction. For centrifugal lift pump of good design and extreme simplicity of construction, which insures high efficiency and economy of power.

95 The Menge pump is still extensively used in this territory and several thousand are in operation in various parts of the United States and in foreign countries.

96 The writer has made several efficiency tests of Menge pump installations, both for drainage and irrigation. Where the pumps were favorably located and the plants in good condition, the efficiencies

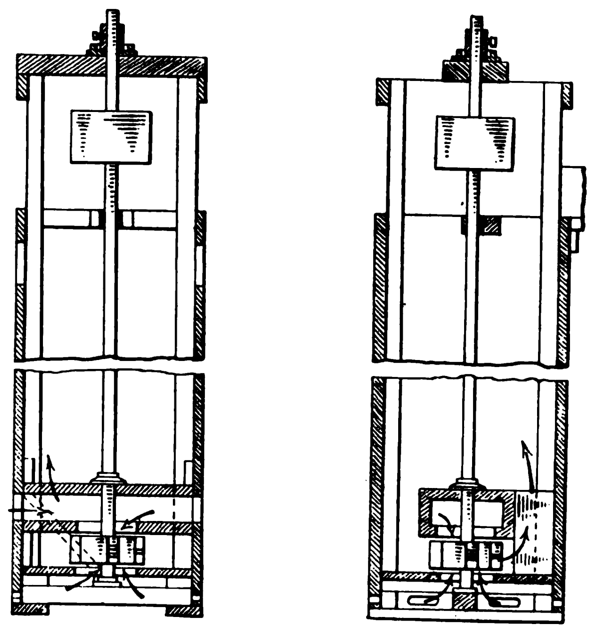


FIG. 14 MENGE PUMP

were excellent. In a few cases where the pumps were set to lift the water higher than was necessary, or the wooden box was not kept in repair, the results were not so good.

97 The results given in Table 5 were obtained from a test of a drainage plant made at Paradis, Louisiana, Sept. 16, 1909. If the mechanical efficiency of the engine be assumed as 90 per cent and the efficiency of transmission 95 per cent, the efficiency of the pump will be approximately 55 per cent with a 5-ft. lift.

98 The plant consisted of a 48-in. by 18-in. Menge pump run by means of a rope drive from an Atlas steam engine, diameter of cylinder 14 in., stroke 20 in. Various speeds of rotation were employed for the purpose of finding the best efficiency. The results were excellent and have been confirmed by tests of other plants.

TABLE 5 TEST OF MENGE PUMP, PARADIS, LA.

September 16, 1909

R.P.M. Engine	R.P.M. Pump	Boiler Gage Pressure, Lb.	Indicated Horsepower	Head on Weir, Feet	Discharge		Gages			Water H.P.	Eff. Engine, Transmission and Pump, Per Cent
					Cu. Ft. per Sec.	Gallons per Min.	Reservoir	Flume	Difference		
67	78	76	11.4	0.09	1.28	575	10.00	12.67	2.67	0.39	3.4
92	98	77	39.6	0.78	39.6	17,800	9.90	13.45	3.55	15.9	40.2
92	99	80	40.1	0.78	39.6	17,800	9.80	13.45	3.65	16.3	40.6
107	117	75	59.6	0.96	54.5	24,500	9.56	13.80	4.24	26.1	43.8
80	87	77	22.1	0.52	20.8	9,360	9.75	13.17	3.42	8.0	36.2
93	101	79	37.7	0.73	35.5	15,970	9.50	13.40	3.90	15.6	41.4
107	116	69	59.3	0.95	53.7	24,150	9.40	13.70	4.30	26.1	44.0
97	104	70	41.7	0.76	37.9	17,050	9.30	13.45	4.15	17.8	42.7
81	88	84	22.0	0.43	15.2	6,840	9.35	13.10	3.75	6.4	29.1
111	121	68	65.8	0.93	52.0	23,400	9.00	13.80	4.80	28.2	42.8
100	111	55	48.4	0.82	42.8	19,300	9.00	13.55	4.55	22.0	45.5
114	123	55	71.0	0.97	55.5	25,000	8.80	13.80	5.00	31.4	44.2
110	122	50	65.3	0.96	54.5	24,500	8.80	13.80	5.00	30.8	47.1
109	118	48	60.6	0.92	51.2	23,000	8.75	13.65	4.90	28.4	46.8
116	128	70	76.5	0.98	56.3	25,350	8.50	13.80	5.30	33.7	44.1

Duration of test, 3 hours 13 minutes.

THE STREAM PUMP

99 The E.L. Stream centrifugal pump was manufactured in 1891 by H. Dudley Coleman & Brother, of New Orleans. It was a side-inlet pump, the main feature of which was the method of attaching the suction elbow by a threaded nipple with a jamb ring. By this means the suction could be pointed at any desired angle to meet the condition of a falling river, as the angle of the suction could be changed to conform to the surface line of the batture without dismantling the pump.

RICE CULTIVATION

100 Louisiana has cultivated rice for many years. Directly after the Civil War the abandoned sugar plantations were planted to rice and this grain became one of the staple crops of the state. For several

years planters were allowed to place flumes in the levees for the purpose of irrigating their rice fields. The flumes became a menace during high water and several crevasses were directly due to leaks around flumes in high water. This led to the passage of the law prohibiting their use. Since that time siphons have been used over the levees. In years of extreme low water, it is necessary to pump the water from the river to shallow pools that supply the siphons. The pumping plants are usually very crude.

101 "River rice" is harvested by means of sickles, as the soil is too soft to permit the use of a self-binder. The crop is usually planted and harvested early. The grain is of good quality and commands a high price in the early market. The rice grown along the river has been but a small part of the total crop of the state in recent years.

102 By far the greater part of the rice crop is now raised on the upland prairies. The increase in the amount of rice produced in the Gulf States has been due chiefly to the utilization of these lands. The water for irrigation must be pumped, the lift varying from a few feet to 55 ft. or more.

IRRIGATION PUMPS FOR RICE

103 The rapid increase in the rice industry, beginning in the early nineties, created a demand for pumping machinery in the prairie country of Southwest Louisiana and many different types were installed. Among the first of the larger pumping plants, erected near Crowley, La., was one which consisted of two cast-iron vessels into which steam at a moderate pressure was introduced alternately. On condensing the steam by means of a spray of water, a vacuum was formed and water entered through suction pipes, partially filling the vacuum. The vessels were placed sufficiently high to allow the water to flow from them to the land to be irrigated. Valves were opened automatically to accomplish the various steps referred to above.

104 The process of pumping was slow and the capacity small when compared to plants of similar size but operating on more modern systems. Worse than this, the pumps failed to act at a time when water was most needed, because the water used for condensing had become too warm, and consequently the vacuum formed was not sufficient to raise the water to the height of the vessels. Mechanical engineers will recognize this pump as very similar to the one used by the Marquis of Worcester and described by him in 1663.

105 Among the other freak designs for pumps in the early days of rice irrigation by pumping, there was a pump that scooped up water at the periphery of a large wheel and discharged it near the axis. The water passages were spirals. The wheel was about 25 ft. in diameter with a width of about 6 ft. It was driven by gearing in a manner similar to the drainage wheels. This type might, with propriety, be called a "scoop" wheel. At least one of these wheels was built for irrigation and another for drainage. The type has now disappeared.

106 The Hollingworth Water Elevator was another curiosity. It was built with a shallow inclined flume up which water was carried by slats attached near their ends to sprocket chains. The head and tail shafts were fitted with sprocket wheels, over which the chains traveled. The outfit was carried by a wooden frame and means were provided to raise or lower the upper end. The tail end was submerged several feet in suction water and by the rapid motion of the slats, the water was elevated and discharged over the upper end. The head under which this pump worked was from 1 ft. to 10 or 12 ft.

CAPACITY OF PUMPING PLANTS FOR RICE IRRIGATION

107 It was about 22 years ago that the first large irrigation pump was installed to irrigate rice on the prairie lands of Southwest Louisiana. At that time there was little definite knowledge regarding the capacity for a given acreage. The rainfall was fairly abundant and usually well distributed, so much so that "Providence" rice had often been successfully raised. However, the uncertainty of rain at the most critical period of the growth of the crop and the small amount of land favorably situated to receive the run-off of higher lands, soon led to the installation of pumping plants. The judgment shown in the selection of the capacity was, of course, widely variable.

108 Profiting by early mistakes and guided by the results of measurements made on canal systems and rice fields by the U. S. Dept. of Agriculture, the capacity became standardized. On the prairie lands where the soil and subsoil contains a large amount of clay so that the seepage is practically nil, the amount should be from 7½ to 8 gal. per min. for each acre irrigated, depending somewhat on the character of the soil and the distance from pumping plant to the field where the water is to be used. In some localities where the nature of the soil is such that it will allow more seepage, as much as 10 gal. per min. should be provided for each acre irrigated, and in land

having a loose subsoil where the seepage is large, the capacity needed may be as much as 30 to 40 gal. Only in isolated cases covering small areas are these large capacities required.

109 The total quantity of water to be pumped in a season into the large canals varies from less than 2 acre-feet to 3 acre-feet per acre of rice irrigated, with the average approximately $2\frac{1}{2}$ acre-feet per acre. The actual plants with their canal systems vary in size from that of the Neches Canal Co., near Beaumont, Texas, which irrigated 26,853 acres in 1914, down to the individual well plant that irrigates only from one hundred to two hundred acres. About 1900 small well pumping plants have been installed in the rice country. In Louisiana the favorite source of power is the steam engine. In Texas, while steam is used to some extent, gasoline engines are largely employed. In Arkansas both the gasoline and steam engine are used.

110 The earlier well pumps were usually vertical shaft centrifugal type, from 4 in. to 8 in. in diameter and placed in pits about 6 ft. square, dug by hand to a depth of from 30 to 50 ft. Trouble was experienced in keeping the shaft in line and this was overcome in a measure by means of the "Munger Attachment," an arrangement for rigidly fastening the bearings to the discharge pipe.

111 Another design of a well pump for an open pit by Stamm, Scheele & Co., of Rayne, La., had two discharge pipes on opposite sides and the bearings for the shaft were rigidly supported between the two pipes. The wood-lined pits gave considerable trouble and in the early nineties the steel pit centrifugal pump appeared. This is the only type of pump installed for wells at present.

STATISTICS ON RICE IRRIGATION

112 The Census of 1910 showed a total acreage of rice harvested in the United States of 610,175. The acreage in the Gulf Coast States was as follows:

	Acres	Per Cent of Total in U. S.
Arkansas	27,419	4.5
Louisiana	317,518	52.0
Texas	237,586	38.9
	582,523	95.4

113 Since the census was taken, rice has been successfully raised in California, where it is a rapidly growing industry. There has also been an increase in the acreage in Arkansas, but Louisiana and Texas

together continue to produce over 80 per cent of the rice grown in this country.

114 The rice crop of the United States for 1915, as reported in the Monthly Crop Report of the U. S. Dept. of Agriculture for December 30, 1915, was as follows:

State	Acres	Per Cent of Total
North Carolina	200	
South Carolina	3,700	0.5
Georgia	900	0.1
Florida	500	
Missouri	200	
Alabama	300	
Mississippi	1,800	0.2
Texas	260,000	32.4
Louisiana	401,000	50.0
Arkansas	100,000	12.4
California	34,000	4.2
Total in U. S.	802,600	

The same publication gives the yield of rice in the United States for 1915 as 28,947,000 bushels and the farm value as $26,212,000.

115 The relative importance of pumping plants in the rice territory of the Gulf States is well shown by the irrigation census of 1910, from which the figures in Table 6 are taken. The census of 1910 showed that the water used on 88.7 per cent of the acreage in Louisiana was pumped, in Texas, 98.8 per cent and in Arkansas, 90.8 per cent. The water that was not pumped was carried to the fields by gravity from flowing wells, from lakes and streams and through siphons along the lower Mississippi River, where the water during the irrigation season is higher than the land behind the levees.

TABLE 6 IRRIGATION PUMPING PLANTS OF THE UNITED STATES

State	Number of Plants	Capacity of Plant, H.P.	Capacity of Plant — Gal. per Min.	Per Cent of Water Pumped from Wells
Arkansas	315	12,440	436,402	87.9
Louisiana	1,007	57,426	5,064,173	28.8
Texas	575	48,179	3,907,380	16.7
Rice District, Total	1,897	118,045	9,407,955	....
Total in United States	15,803	361,480	19,355,864	....
Total excluding Rice District	13,906	243,435	9,947,909	....

NEW ORLEANS MODERN DRAINAGE SYSTEM

116 The drainage of New Orleans and the sanitary sewers be to separate systems. Work was begun on the modern drainage sy in 1896 and it went into operation in 1900. The reports of the Di age Board and later of the Sewerage and Water Board give details of the various installations. Table 7 is from the Tw Fourth Semi-Annual Report of the Sewerage and Water Bc

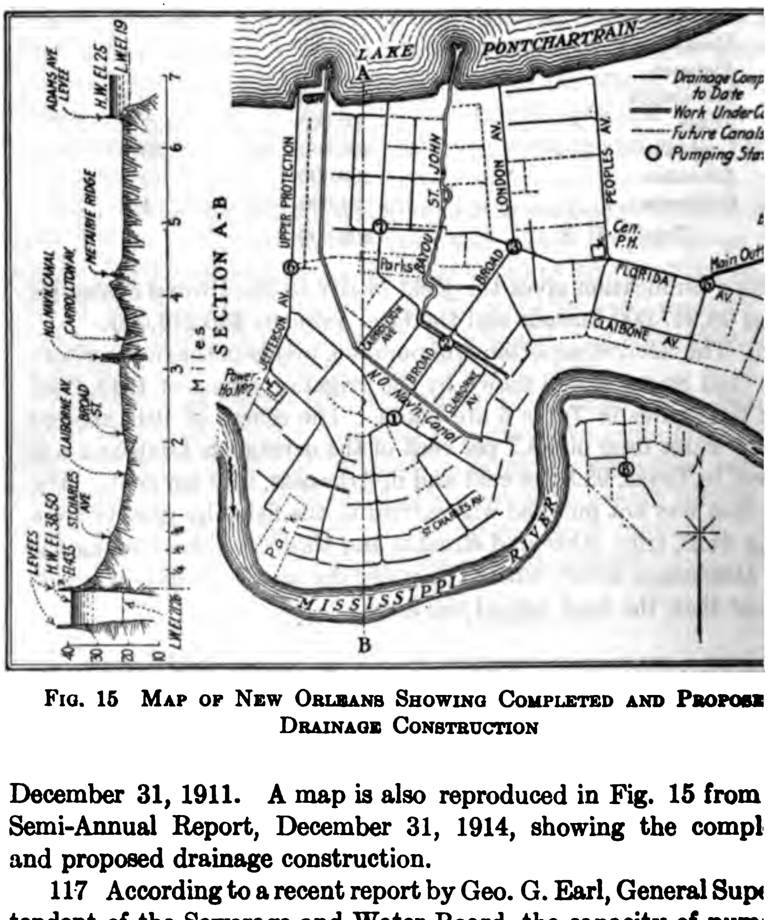


FIG. 15 MAP OF NEW ORLEANS SHOWING COMPLETED AND PROPOSE DRAINAGE CONSTRUCTION

December 31, 1911. A map is also reproduced in Fig. 15 from Semi-Annual Report, December 31, 1914, showing the compl and proposed drainage construction.

117 According to a recent report by Geo. G. Earl, General Sup tendent of the Sewerage and Water Board, the capacity of pum plants for drainage will eventually be equal to 7.33 in. of ru removed in 24 hours. The present capacity is about 3.5 in. The drained amounts to 25,000 acres, or a little more than 39 square m

It is believed that no other city in the world has such enormous volumes of drainage water to dispose of.

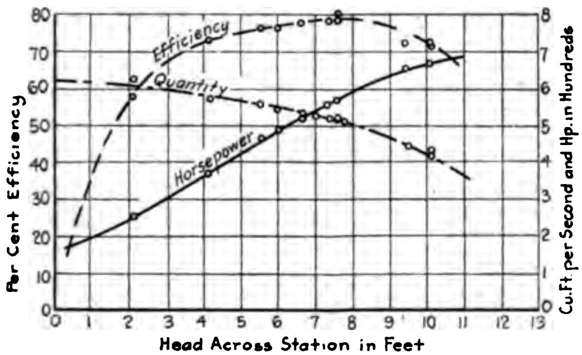


FIG. 16 TESTS OF 12-FT. SCREW PUMP

118 The pumping capacity for the drainage of the city is being increased by the installation of eleven 12-ft. screw pumps to be used in

FIG. 17 12-FT. SCREW PUMPS, DRAINAGE STATION NO. 1

Drainage Pumping Plants Nos. 1, 2, 3, 5, 6 and 7. The test of one of these pumps reported in Engineering Record, Jan. 8, 1916, by Geo. G. Earl, and in Engineering News of Jan. 13, 1916, by Prof. W. H. P.

TABLE 7 DATA ON PUMPING UNITS

Drainage Operating Department, Sewerage and Water Board of New Orleans.

Pumping Station Number	Number of Units	Date of Installation	Screw-Type Pump = S Centrif.-Type Pump = C	Vertical Shaft = V Horizontal Shaft = H	Impeller Enclosed or Open	Diameter of Impeller	Diameter of Suction Pipe	Diameter of Discharge Pipe	Combined Length of Suction and Discharge Pipes	Capacity, Cu. Ft. per Second	Revolutions per Min.	Nominal Head	Horse Power of Motor or Engine	Guaranteed Efficiency of Pump	Average Efficiency of Pumps at Nominal Head	Driving Power
						In.	In.	In.	Feet			Feet		%	%	
1	3	1903	S	V	......	108	96	96	55	250	88.3	5	266	60	54	3-phase synchronous motors
1	1	1903	C	V	Enc.	49.5	42	42	55	40	167	10	149	60	58	3-phase synchronous motor
1	1	1913	S	H	......	28	30	30	30	24.6	360	6.9	30	70	80	3-phase induction motor
1	2	1915	S	H	......	144	144	144	45	550	75	5	600	70	76	3-phase synchronous motors
2	2	1902	S	V	......	108	96	96	60	250	88.3	5	266	60	56.5	3-phase synchronous motors
2	1	1902	C	V	Enc.	49.5	42	42	60	40	167	10	149	60	60	3-phase synchronous motor
2	2	1916	S	H	......	144	144	144	55	590	75	5	600	70	......	3-phase synchronous motors[1]
3	2	1903	C	H	Enc.	84	84	84	70	250	82	8	400	60	63	3-phase synchronous motors
3	1	1903	C	H	Enc.	81	36	36	70	50	100	12	120	60	58.3	3-phase synchronous motor
3	2	1916	S	H	......	144	144	144	60	590	83.3	10	1200	70	......	3-phase synchronous motors[1]
5	1	1896	C	H	Open	70	36	52	71	150	125	12	400	65	No test	Triple-expansion engine
5	1	1896	C	H	Open	70	36	52	71	150	125	12	333	65	No test	3-phase synchronous motor
New 5	2	1916	S	H	......	144	144	144	41	550	83.3	10	1200	70	......	3-phase synchronous motors[1]
6	4	1900	C	V	Enc.	114	96	96	60	250	62.5	10	466	65	66.6	3-phase synchronous motors
6	1	1906	C	V	Enc.	37	30	30	60	30	280	13.5	100	60	65.9	3-phase induction motor
6	2	1916	S	H	......	144	144	144	86	590	83.3	10	1200	70	......	3-phase synchronous motors[1]
7	3	1900	C	V	Enc.	114	96	96	60	250	62.5	10	466	65	No test	3-phase synchronous motors
7	1	1906	C	V	Enc.	37	30	30	60	30	280	13.5	100	60	65.9	3-phase induction motor
7	1	1916	S	H	......	144	144	144	40	590	83.3	10	1200	70	......	3-phase synchronous motor[1]
8	1	1901	C	H	Open	70	36	52	71	150	125	12	400	65	No test	Triple-expansion engine

Creighton, shows efficiencies above 70 per cent for lifts between 3.5 and 10.5 feet. The efficiency curve rose to nearly 80 per cent at approximately 7.5 ft. lift. (Fig. 16.) The quantity of water pumped varied from 600 cu. ft. per sec. at a 3-ft. lift to 400 cu. ft. at a lift of 10.5 ft.

119 These pumps were built by the Nordberg Manufacturing Co., of Milwaukee, from designs by A. B. Wood, Mechanical and Electrical Engineer of the Sewerage and Water Board. In Fig. 17 is a view of the large pumps in drainage station No. 1 of the New Orleans system and in Fig. 18 a view showing the discharge from one of these pumps.

Fig. 18 Discharge from One Screw Pump at Station No. 1

120 Chamber wheel pumps made by the Connersville Blower Company, or the P. H. & F. M. Roots Co., of Connersville, Indiana, have been used in irrigation and drainage work in this section. One of these is shown in Fig. 19.

The largest rice irrigation enterprise in all the Gulf Coast country, that of the Neches Canal near Beaumont, Texas, uses Connersville pumps at the first lift to elevate water from 30 to 35 ft. There are six units, having a total capacity of approximately 200,000 gal. per min.

121 The water is carried for about two miles through a canal, having levees 150 ft. between crowns, where it is again elevated about 7.5 ft. by two Connersville units and one centrifugal pump built by

TABLE 8 TEST OF MORRIS CENTRIFUGAL-PUMP UNIT, SECOND LIFT, NECHES CANAL

July 15–16, 1909, Beaumont, Tex.

July 15, 1909

R.P.M.	Indicated H.P.		Cubic Feet per Sec.	Gallons per Minute	Actual Lift in Feet	Total Head on Pump, Ft.	Useful Water H.P.	Efficiencies: Engine, Pump and Piping	Efficiencies: Engine and Pump	Efficiencies: Pump and Piping	Efficiencies: Pump	Fuel Oil, Lb.
112	259.8	....	....	......	8.18	9.99	......	......	......	......	....	....
112	260.2	....	151.8	68,300	8.12	9.99	139.4	53.5	66.0	57.0	70.2	232
112	262.4	....	154.1	69,300	8.14	10.05	141.8	54.1	66.9	57.5	71.2	273
112	260.8	....	153.9	69,250	8.10	10.11	140.7	53.9	67.3	57.3	71.6	241
112	263.0	....	154.4	69,500	8.08	10.18	141.0	53.6	67.2	57.0	71.5	259
112	262.0	....	153.4	69,100	8.07	10.16	139.9	53.4	67.2	56.8	71.5	254
112	261.9	....	154 8	69,700	8.07	10.18	141.2	53.8	68.1	57.3	72.5	269
112	263.7	....	154.8	69,700	8.06	10.21	141.1	53.6	67.7	57.0	72.0	250
112	260.6	....	153.4	68,100	8.03	10.23	139.2	53.9	68.1	56.8	72.5	260
Mean	261.6	....	153.8	69,300	....	10.12	140.5	53.7	......	57.09	71.62	....
											Total	2038

261.6 – 16.2 = 245.4 245.4 + 261.6 = 94%

July 16, 1909

R.P.M.	Indicated H.P.		Cubic Feet per Sec.	Gallons per Minute	Actual Lift in Feet	Total Head on Pump, Ft.	Useful Water H.P.	Efficiencies: Engine, Pump and Piping	Efficiencies: Engine and Pump	Efficiencies: Pump and Piping	Efficiencies: Pump	Fuel Oil, Lb.
102.5	152.0	....	92.5	41,700	8.33	9.45	87.1	57.3	65.0	62.8	71.2	....
106	191.3	....	118.5	53,300	8.30	9.88	111.1	58.1	60.1	62.8	74.6	....
109 108.5	230.5 218.1	219.3	134.0	60,300	8.31	10.12	126.0	57.5	69.9	61.7	75.0	....
116 115	294.7 287.1	291.4	161.5	72,700	8.35	11.04	152.3	52.3	69.2	55.7	73.6	....

TABLE 9 ROOTS ROTARY PUMP, WILLSWOOD PLANTATION

June 15, 1909

Boiler Pressure	Indicated Horsepower	R.P.M. Engine	R.P.M. Pump	Cu. Ft. Per Sec.	Gallons Per Min.	Actual Lift, Ft.	Head on Pump	Water H.P. On Actual Lift 1	Water H.P. On Head on Pump 2	Efficiency Engine, Transmission and Pump 1	Efficiency Engine, Transmission and Pump 2
80	153.5	108.5	89.0	78.2	35,200	5.2	10.0	46.0	88.4	29.9	57.5
89	163 0	115.0	94.5	83.0	37,300	5.3	10.3	49.6	96.5	30.4	59.1
89	155.8	112.5	92.0	80.8	36,300	5.4	10.3	49.3	94.0	31.6	60.3
90	166.8	116 5	95.5	83.9	37,700	5.5	10.5	52.1	99.5	31.3	59.8
92	166.8	117.5	96.0	84.3	37,900	5 5	10.5	52.3	100.0	31.4	60.0
87	152 2	109.0	89.5	78.7	35,400	5.5	10.5	48.8	93.3	32.1	61.3
86	145.6	106 0	87.0	76.4	34,400	5.6	10.5	48.4	90 8	33.2	62.3
											60.04

Duration of test, 1 hour, 30 minutes.

Morris Machine Works, Baldwinsville, New York. Table 8 gives test data for the latter unit. The boiler pressure varied between 156 and 157 lb., and the vacuum between 26.1 and 26.2 in.

122 A test of the main pumping plant of the Neches Canal was made in 1906 and reported in Vol. 28, Transactions of this Society under the title, "Tests of a Rotary Pump." The test made showed a remarkable efficiency for the pumps and the results were confirmed by a test at the same plant made in August, 1915, when almost identical results were obtained. No adjustments of the pumps in the two units tested has been necessary during the twelve years they have been used.

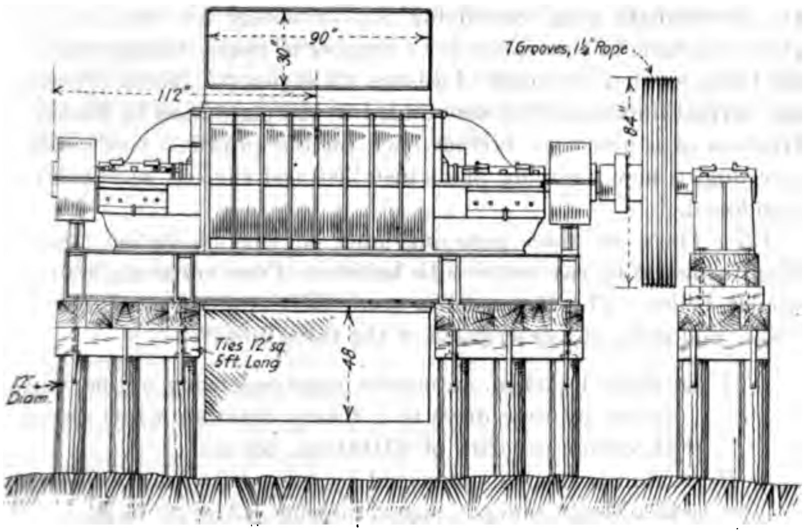


Fig. 19 Roots 30-in. by 90-in. Rotary Pump

123 The installations of rotary chamber wheel pumps for rice irrigation have not been an unqualified success; there have been many troubles with some of the plants, partly because of the lack of proper talent in their operation. However, the writer has for years visited pumping plants of this type in which trouble was conspicuous by its absence, largely because of the high-grade men in charge.

124 The intermittent acceleration and retardation of the water pumped make it necessary to have air chambers near the pump. Siphons have been used as on a centrifugal pump, but the latter pump with its constant velocity of discharge has considerable advantage over the former.

125 Drainage pumps, when built in large units such as are ordinarily required in drainage work, lift the water higher than is necessary, and while they are efficient if credited with the higher lift, they lose their efficiency on low lifts when actual difference of levels is considered.

126 This point is well illustrated by the test, Table 9, made at the drainage pumping plant of Willswood Plantation. This is an old river-front plantation of 2400 acres, on the Southern Pacific Railway, ten miles above New Orleans. About one third of this tract is sloping and has good natural drainage, though practically all of the water drains into the main canal system and is handled by the pumps. An intermediate area, containing approximately one third of the whole, has been in cultivation for a number of years, drainage originally being secured by means of a large wheel pump. About 19 years ago, several hundred acres were added to the plantation by the construction of a new levee farther back on the prairie. New canals were dug, a new pumping plant installed and the old wheel pump abandoned.

127 There are three pumping units on this plantation, steam being furnished by two water-tube boilers and one horizontal return-tubular boiler. The fuel used is crude oil. A feedwater heater is used. Following is a description of the three units:

1 A 16-in. by 24-in. automatic non-condensing engine, connected by rope drive to a rotary chamber wheel pump. Maximum capacity of 40,000 gal. per min.
2 A similar engine, connected by rope drive and bevel gear to a Menge pump. Size of impeller, 42 in. by 16 in.
3 A double vertical engine, direct-connected to a centrifugal pump. Diameter of discharge pipe, 36 in.

128 Pumps 1 and 2 discharge into open flumes at an average head on pump of about ten feet, which is about five feet greater than is necessary. The bottom of the discharge flume was placed at the elevation of the top of the back levee, which normally was about 4 ft. higher than the water of the swamp behind the levee.

129 When the pump was credited with the head through which the water was elevated at the pump, the average efficiency of engine, transmission and pump was found to be 60 per cent. Assuming the mechanical efficiency of the engine as 90 per cent and the efficiency of transmission as 95 per cent, the pump efficiency is a little more than 70 per cent.

130 It should be added that the pump had been installed about ten years before this test was made and had gone through a fire that destroyed the building in which it was housed. The amount of damage to the pump is unknown but it is certain that it was not improved by the experience.

TEST OF NECHES CANAL RELIFT PLANT

131 The test was made August 10, 1906. The same methods were employed and the same instruments used as in the test of the Neches Main Pumping Plant above referred to.

132 The water measurements were made in the flume of Unit No. 2, using current meter and pitot tube. The average slip was found to be about 0.2 per cent. This slip was applied in computing the discharge from Unit No. 1, from the observed revolutions per minute. The results are given in Table 10.

FUELS

133 Irrigation and drainage plants used wood or coal for fuel previous to the opening years of this century.

134 The Lucas oil well was brought in Jan. 10, 1901, at Spindletop, near Beaumont, Texas. Soon after, oil was discovered at other fields, many of which were in or near the rice irrigation country, notably Jennings, Welsh, Vinton and Sulphur in Louisiana, and Dayton, Sour Lake, Saratoga, Batson and Humble in Texas.

135 At first the production of crude oil was far in excess of the demand of the refineries and crude oil sold for as low as ten cents a barrel of 42 gal. at the fields. Later, as refineries were erected and means worked out to handle the output, the price of crude oil steadily advanced to from 75 cents to $1.50 per barrel and even more in some instances. The cost at any particular plant in the Gulf Coast country depends upon the transportation costs of delivery and the time of purchasing. Many of the nearby pumping plants were supplied by pipe lines from the oil fields. The problem of pumping both for irrigation and for drainage plants has been greatly simplified by crude oil as a fuel.

136 During the year 1915 the petroleum production of the United States as estimated by the U. S. Geological Survey was 267,400,000 barrels. Of this amount, Texas produced 26,000,000 barrels or 9.72 per cent and Louisiana 18,500,000 barrels or 6.92 per cent.

TABLE 10 TEST OF RELIFT PLANT, NECHES CANAL COMPANY, BEAUMONT, TEX., (SUMMARY OF MAIN ITEMS ONLY). AUGUST 10, 1906

R.P.M. Engines		Indicated Horsepower			Pump Discharge, Cu. Ft. per Sec.	Displacement, Cu. Ft. per Sec.		Slip, Per Cent	Height Through Which Water Was Raised, Ft.		Useful Water Horsepower		Efficiencies: Pump and Engine		Efficiencies: Pump		Pressures, Boilers		Vacuum, In.	Fuel Oil Lb.
No. 1	No. 2	Engine No. 1	Engine No. 2	Total		No. 1	No. 2		No. 1	No. 2	No. 1	No. 2	No. 1	No. 2	No. 1	No. 2	No. 1	No. 2		
52.8	49.7	229.7	217.8	447.5	133.2 134.0	142.3	134.0	+0.6	10.1	10.05	162.5	152.2	70.8	70.0	75.5	74.5	151.5	151.0	23.6	 343
53.8	49.5	231.3	214.8	446.1	131.4 128.9	145.1	133.5	+1.6	10.1	10.05	165.7	151.7	71.7	70.5	76.3	75.1	148.5	149.0	23.6	 686
53.6	48.6	230.4	210.0	440.4	131.2 131.8	144.5	131.0	−0.2	10.1	10.05	165.0	148.8	71.7	70.9	76.3	75.5	150.5	150.0	23.5	1030
53.6	48.7	230.2	213.0	443.2	132.5 131.9	144.5	131.3	−0.9	10.1	10.05	165.0	149.2	71.8	70.0	76.4	74.5	150.5	150.0	23.5	1373 1716
53.8	48.0	232.6	207.0	439.6	130.4 129.7	145.1	129.4	−0.8	10.05	10	164.8	146.3	70.9	70.7	76.6	75.3	150.5	150.0	23.4	 2059
53.8	48.4	234.7	205.2	439.9	129.5 128.8	145.1	130.5	+0.8	10.05	10	164.8	147.5	70.2	71.9	74.8	76.6	151.5	151.0	23.4	 2402
53.6	48.4	232.5	207.2	439.9	131.0 133.1	144.5	130.5	−0.4	10.05	9.95	164.6	146.8	70.8	70.8	75.4	75.4	149.5	150.0	23.4	 2746
53.8	48.4	233.3	205.5	438.8	129.6 129.2	145.1	130.5	+0.7	10.05	9.95	164.8	146.8	70.7	71.4	75.3	76.1	150.5	150.0	23.4	 3089
53.6	48.8	230.0	204.1	434.1	 Mean:	144.5 144.5	131 6 131.4		10.05	9.90	163.4 164.5	147 3 148.5	71.1	72.2	75.7 75.8	76.8 75.5	147.5	146.5	23.4	3432

APPENDIX

TYPICAL DRAINAGE AND IRRIGATION PLANTS. DESCRIPTION OF EQUIPMENT AND RESULTS OF TESTS

THE JEFFERSON-PLAQUEMINES DRAINAGE DISTRICT

This district embraces about 38,000 acres of land located in the parishes of Jefferson, Plaquemines and Orleans, on the right bank of the Mississippi. The tract extends from a point nearly opposite the center of the city in a southeasterly direction. The western boundaries are Harvey Canal and Bayou Barataria. The land along the river has been in cultivation for many years, but a large part forming the interior of the district was wet prairie. It is the largest drainage district in the State where water is pumped.

TABLE 11 DATA UPON JEFFERSON-PLAQUEMINES PUMPS

76-in. Pump

Difference of Levels	Gallons per Minute	Duty in Millions of Foot-pounds per 1000 Lb. Dry Steam
1	168,000	35
3	156,000	78
5	135,000	90
7	130,000	90
10	11,000	....
13	9,000	....
	48-in. Pump	
3	47,500	60
5	45,000	80
7	42,000	75
8	40,000	70

When the pumping plant is completed the drainage coefficient will be $\frac{4}{10}$ in. removed in 24 hours. At present the equipment consists of two 76-in. pumps, each driven by a cross-compound engine, 16½ and 35 by 36 in., and one 48-in. pump driven by a cross-compound engine, 9 and 18 by 36 in. All the pumps are of the horizontal-shaft centrifugal type, designed to operate against differences of level varying from 1 to 13 ft. The engines are operated condensing, using surface condensers guaranteed for a vacuum, with water at 85 deg. fahr., of 26 in. of mercury. Three Heine boilers are used, one with 1380 sq. ft. and two

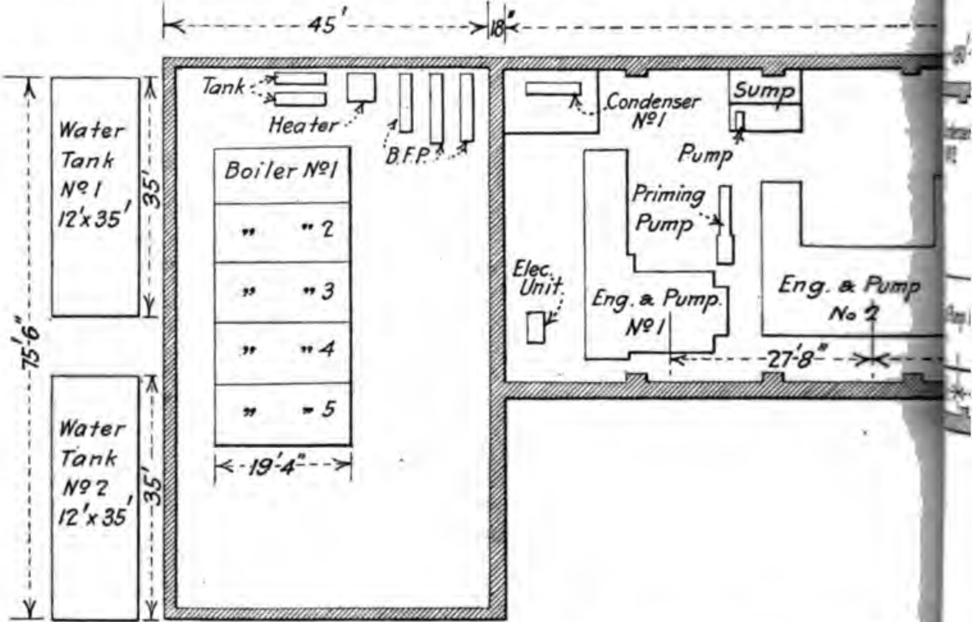


FIG. 20 GENERAL ARRANGEMENT OF PUMPING STAT

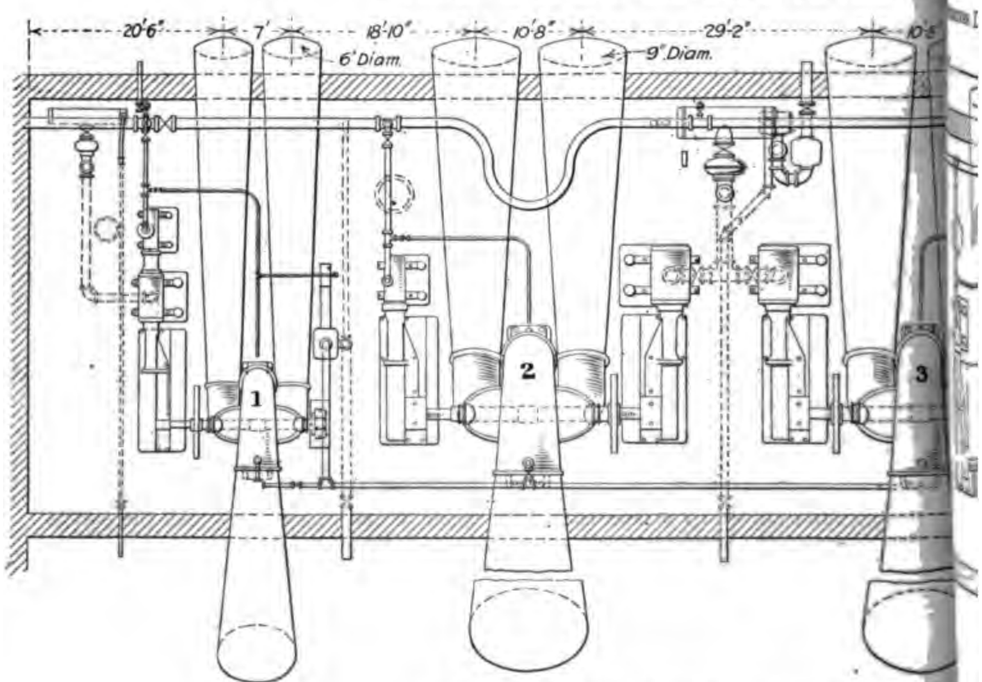
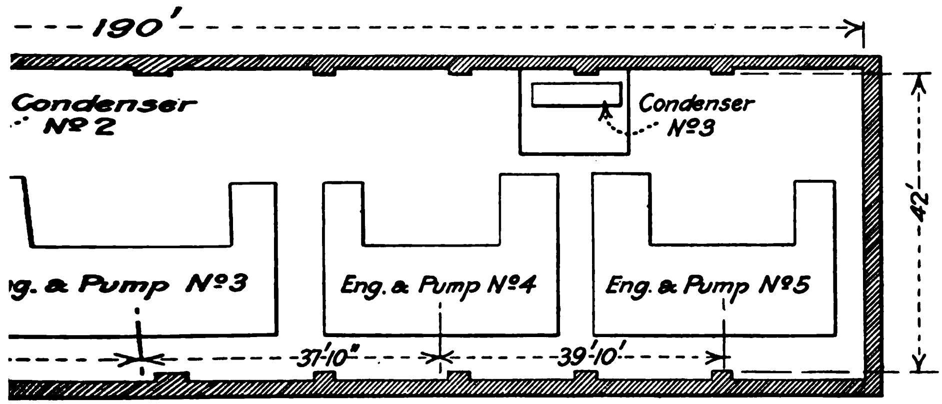
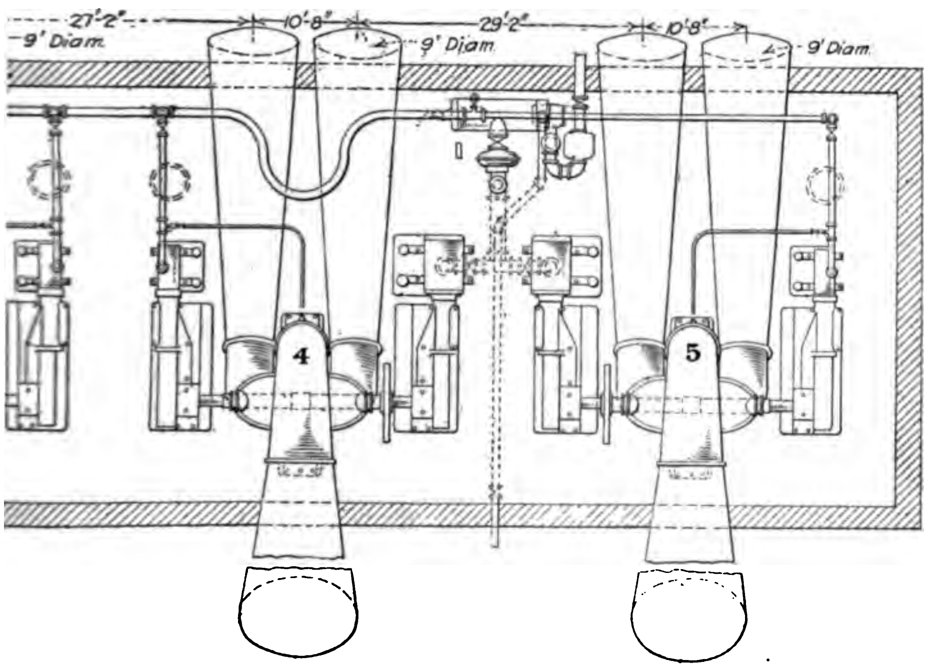


FIG. 21 GENERAL ARRANGEMENT OF ENGINES AN



ON-PLAQUEMINES DRAINAGE DISTRICT



3. JEFFERSON-PLAQUEMINES DRAINAGE DISTRICT

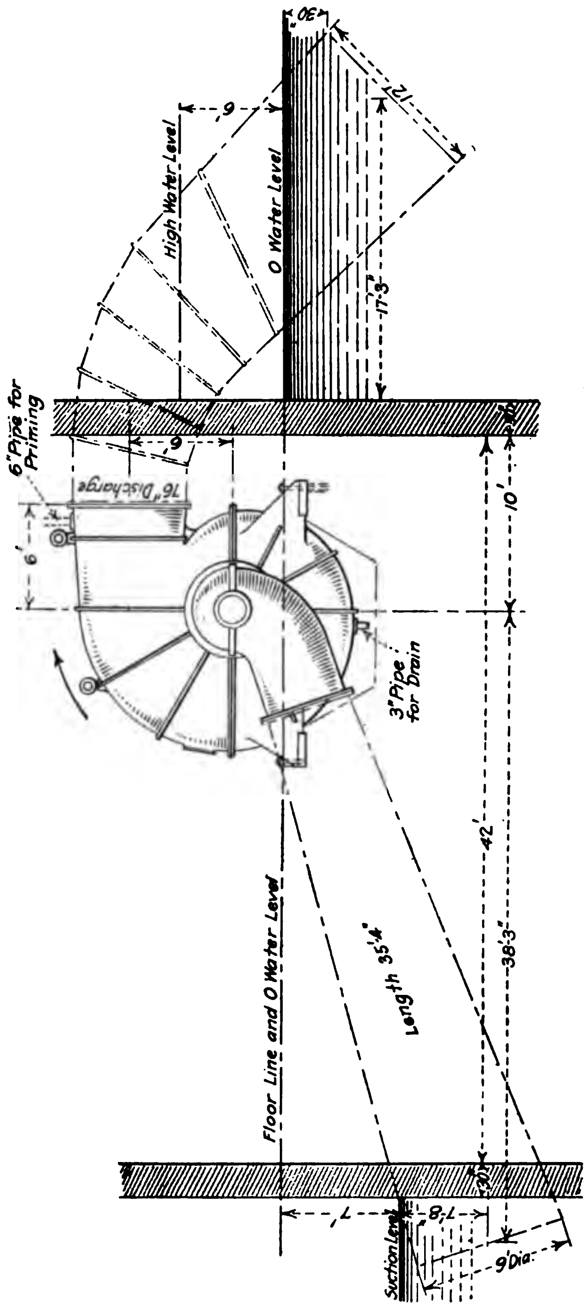


FIG. 22 ELEVATION OF 76-IN. JEFFERSON-PLAQUEMINES PUMP

with 2220 sq. ft. of heating surface each. The steam piping is extra heavy and the whole plant gives the impression of excellent design and good materials and work-

FIG. 23 ONE UNIT OF JEFFERSON-PLAQUEMINES PLANT

manship. The engines were built by the Nordberg Manufacturing Company, Milwaukee, and the pumps by the Southwark Foundry and Machine Company,

FIG. 24 76-IN. JEFFERSON-PLAQUEMINES PUMP

Philadelphia. J. F. Coleman was the engineer for this project and the mechanical engineering work was in charge of Walter Castanedo, Mem.Am.Soc.M.E. The guarantees for capacity and duty under given conditions are given in Table 11.

This plant has not been tested because a great deal of vegetation is pumped, of a nature that would interfere with pitot tube readings.

PUMPING PLANT AT CITRUS, LA.

The New Orleans Lake Shore Land Company has reclaimed an area of 6943 acres of wet prairie land, located on Lake Pontchartrain in Orleans Parish, and therefore in the city of New Orleans. A large portion of the land has been planted to citrus fruit and it is the intention of the company eventually to convert the entire tract into orange groves.

FIG. 25 76-IN. PUMP DISCHARGE, JEFFERSON-PLAQUEMINES STATION

The pumping plant first erected on this tract, near Little Woods, La., contained a 48-in. Morris centrifugal pump with double suction pipes 38 in. in diameter. The pump is driven by a 16-in. by 36-in. Corliss engine direct-connected. Steam is furnished by a Heine boiler. When first installed, in 1908, the suction pipes were 105 ft. long and each contained a 90 degree elbow and was enlarged to 60 in. at the end. The discharge pipe was 31 ft. long and not enlarged. A test made by the writer in April, 1909, showed the overall efficiency of the engine, pump and piping to be 22 per cent, with a lift of 2.3 ft. Since the test was conducted the suction pipes have been shortened and other changes made.

FIG. 26 PUMPING PLANT, CITRUS, LA., SHOWING DISCHARGE SIDE FACING LAKE PONTCHARTRAIN

This original plant has now been superseded for regular pumping by a new plant erected in 1913 at Citrus, La., about 10½ miles from the center of New

FIG. 27 PUMPING PLANT, CITRUS, LA., SHOWING SUCTION PIPES AND SUCTION BASIN

37¾ in. in each case. Two pitot tubes were used to determine mean velocity in the suction pipes, the points of observation being so chosen that the arithmetical mean of the velocities at ten different points in each suction pipe gave the mean velocity of the water in that pipe. Traverses were made simultaneously in the two suction pipes at half-hour intervals.

The test of June 15 was made at 112 r.p.m. for engine and pump, and continued for four hours. During that time the fuel oil consumed was determined by the fall of level in a tank, and from the readings, the quantity of oil used was accurately computed. The horsepower developed varied but little, averaging 261.6 h.p., and delivery of the pump averaged 69,300 gal. per min. The total head on pump, including velocity head, ranged from 9.99 to 10.23 ft., with an average value of 10.12 ft. The corresponding efficiencies of the pump ranged from 70.2 to 72.5 per cent, with an average efficiency of 71.6 per cent.

On June 16, observations were taken from engine and pump at variable speeds, and friction cards were taken from the engine. Whereas on the 15th, the average quantity of water pumped was 69,300 gal. per min. at a mean efficiency for the pump of 71.6 per cent, on June 16, with exactly the same head on pump, with a speed of 108¾ r.p.m., the efficiency of the pump was 75 per cent, while the quantity of water pumped was 60,300 gal. per min. During the latter test, the speed was at one period increased to 116 r.p.m., at which the delivery was raised to 72,700 gal. per min., with an accompanying efficiency of 73.6 per cent.

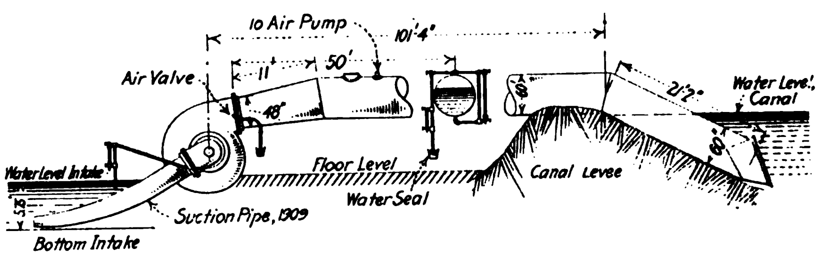


FIG. 30 48-IN. MORRIS PUMP, NECHES RELIFT STATION

The efficiencies for the test of June 16 are plotted in Fig. 31.

In the curve showing variation of pump efficiencies with revolutions per minute, it will be noted that the efficiency of pump rises to 75 per cent at 109 r.p.m. and then falls off for higher speeds.

From the curve showing the efficiency of pump and piping, on the basis of useful work, it is evident that the most economical speed is from 102 to 107 r.p.m., at which revolutions the heads ranged from 9 to 9.6 ft. and the quantity of water pumped from 40,000 to 57,000 gal. per min. The efficiency based on useful work is of interest in operating the plant.

On June 15, with revolutions at 112 per min. and with an average head of 10 12, the efficiency of pump showed a marked improvement as the head increased from 9.99 ft. to 10.23 ft. Now to find the probable head for 112 r.p.m. in the test of June 16, it will be necessary to interpolate between the total head at 108¾ r.p.m. and that at 115½ r.p.m., (Fig. 31), which respectively were 10.12 and 11.04 ft., giving 10.58 ft. It is reasonable to suppose that the efficiency for this head would be about sufficient to make the results of the first day's test agree with the second. If it had been impossible to uncouple the engine and pump, a mechanical efficiency of 90 per cent would have been assumed, as usual for engines under corresponding conditions, which would have given an efficiency for the pump on June 15 of 74.7 per cent.

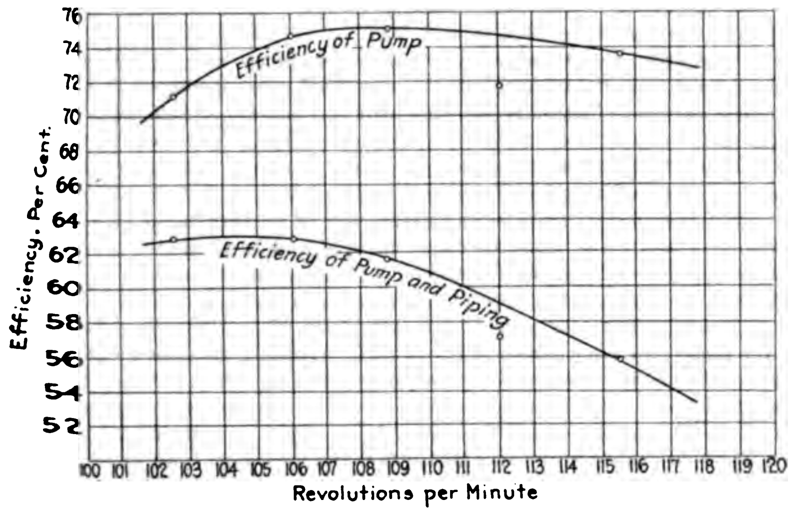


FIG. 31 EFFICIENCY CURVES, MORRIS PUMP, NECHES RELIFT STATION

TEST OF GRAND CANAL PUMPING PLANT[1]

Between the pumping seasons of 1905 and 1906, extensive changes were made in the equipment of the pumping plant of the Grand Canal. There were installed a new Atlas water-tube boiler, a tandem-compound Hamilton Corliss engine, 20 by 40 by 42 in., with jet condenser having a pump 14 by 20 by 24 in.; and a 36-in. horisontal-shaft centrifugal pump made by the Lawrence Machine Company, Lawrence, Mass.

The new equipment was tested on September 21, 1906. It was not possible to run a long test, since the demand for water for irrigation is small at this season and the pumps had to be stopped when the canal was filled to the danger line. Fuel consumption during the test was extremely regular, however, the water level of the boiler fairly constant, and all conditions favorable for accurate results.

The fuel oil was measured in a calibrated barrel and its heat value, determined by means of a Parr calorimeter, was found to be 17,834 B.t.u. per lb., the lowest heat value the writer has ever found in an oil from the Jennings field. No water

[1] Bul. 183 O.E.S., U.S. Dept. of Agri., 1907.

was present in the oil. Boiler feed was measured by a 6-in. Cipolletti weir, so arranged that the heater could be used during the test.

Water measurements of the pump discharge were made with a current meter, in the flume which conducts the water from the discharge to the canal. The current meter was slowly moved across the flume at three different depths, the direction of movement then reversed, and the path retraced in an opposite direction. On account of the unusual width of flume (19.2 ft.) it was found necessary to correct the current meter readings for the component of motion at right angles to the axis of the flume in each case.

The average mechanical efficiency of engine, pump and rope drive was 69 per cent for observations where the proper speed was maintained. If the mechanical efficiency of engine be assumed to be 92.5 per cent and efficiency of transmission 95 per cent, the efficiency of pump would be 78.5 per cent.

The centrifugal pump and piping show remarkably high efficiency for a pump of this type, due primarily to good design, although one other cause is worthy of note. The double suction pipes enlarge from 24 in. near pump to 34 in. at a distance of about 4 ft. from the flange of pump; again at the lower end of the suction pipes there is a conical frustum 9 ft. long, with a diameter of 42 in. at intake. The vertical discharge pipes in each case are enlarged to 42 in. at a short distance above the pumps, and just below where they enter the bottom of the flume they are enlarged in the last 5 ft., changing the cross section from a circular section 42 in. in diameter to a section 51 in. square at entrance to flume. Enlargement of suction pipe reduces the velocity of the entering water and reduces the entrance loss, while the enlarged discharge pipe reduces the velocity of the water discharged and consequently the "velocity head" lost at entrance to flume.

The results of the test are as follows:

Boiler Test, Grand Canal, September 21, 1906

Duration of test, 3.717 hr.
Total fuel oil used, 2476 lb.
Average steam pressure by gage, 153.4 lb.
Average temperature of feedwater, 188.5 deg. fahr.
Factor of evaporation, 1.074.
Total weight of water fed to boiler, 28,944 lb.
Equivalent water evaporated from and at 212 deg. fahr., 31,086 lb.
Boiler horsepower, 242.2.
Water apparently evaporated per pound of oil, 11.69 lb.
Equivalent evaporation from and at 212 deg. fahr. (not corrected for quality of steam), 12.56 lb.
Total feedwater (including steam used by auxiliaries) per indicated horsepower hour, 17.7 lb.

DRAINAGE PUMPING PLANT, DALCOUR, LA.

This pumping plant was installed in 1913 by the Fidelity Land Company, Dalcour, La., about 22 miles below New Orleans, to drain 650 acres of citrus fruit land. It was designed and erected by A. M. Lockett & Co., Ltd., New Orleans. At present one unit is in place and it is the intention to install a duplicate unit later.

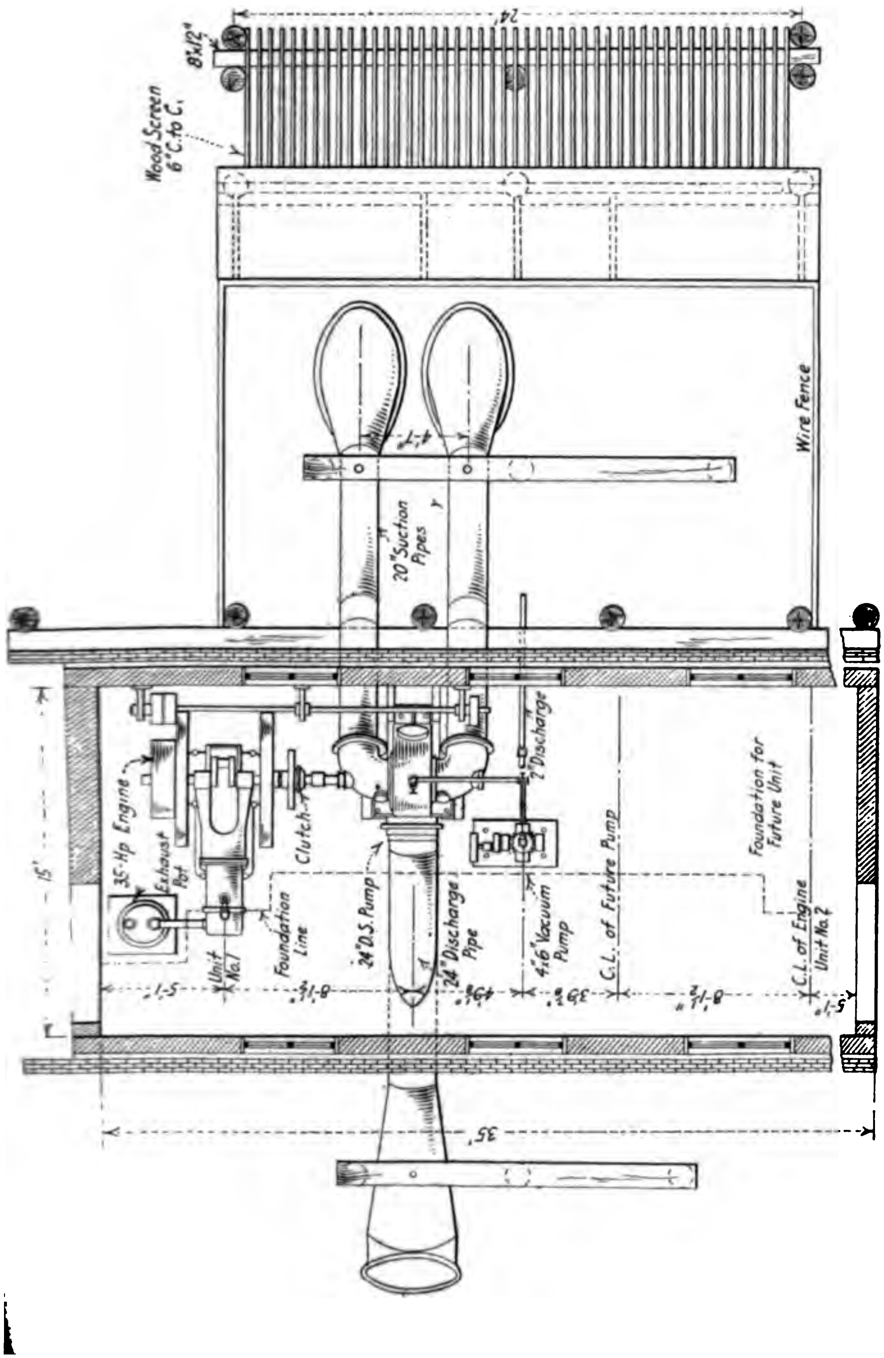
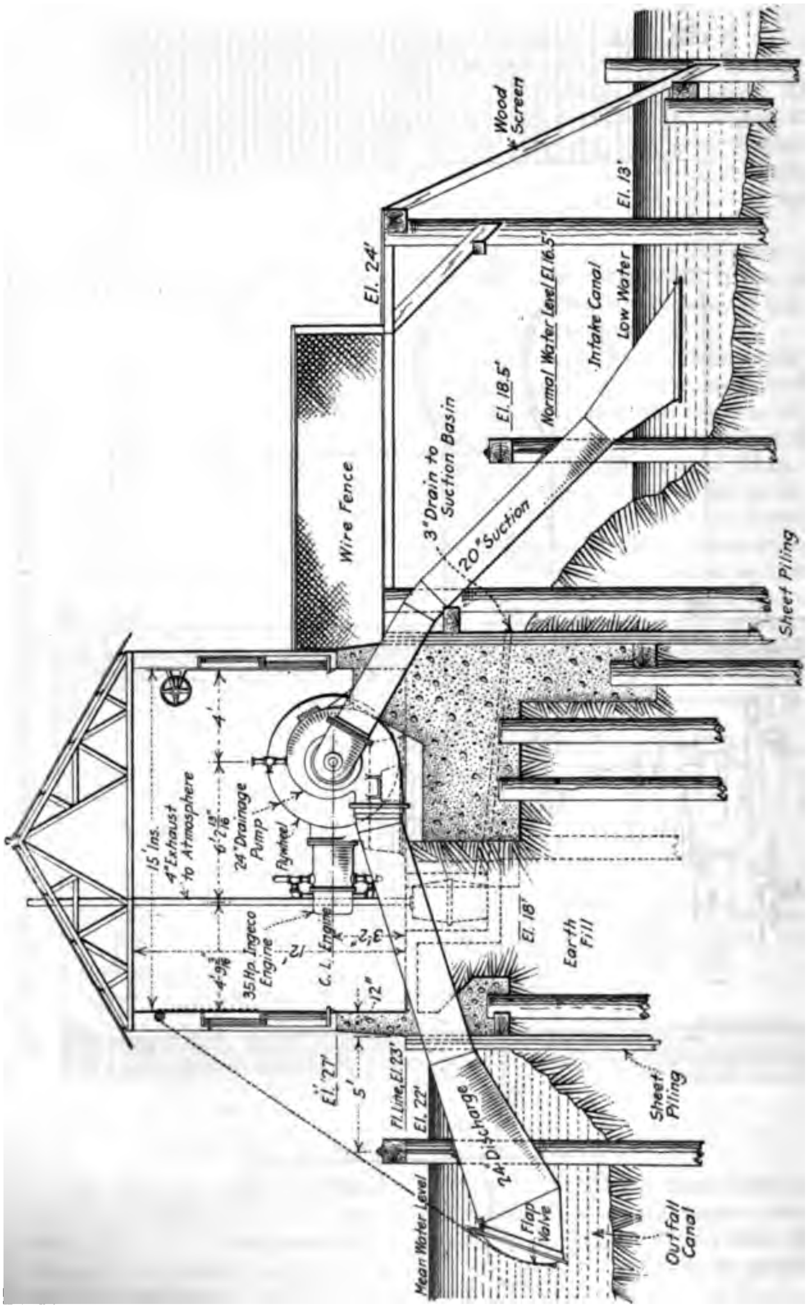


FIG 32 PLAN OF PUMPING PLANT, FIDELITY LAND COMPANY, DALCOUR, LA.


Wood Screen
El. 24'
El. 13'
Normal Water Level El. 16.5'
Intake Canal
Low Water
El. 18.5'
3" Drain to Suction Basin
20" Suction
Wire Fence
Sheet Piling
15' Ins.
4" Exhaust to Atmosphere
24" Drainage Pump
Flywheel
35 Hp. Ingeco Engine
C. L. Engine
3'-2"
12"
El. 18'
Earth Fill
El. 27'
5'
Fl. Line, El. 23'
El. 22'
24" Discharge
Mean Water Level
Flap Valve
Sheet Piling
Outfall Canal

A 24-in. Worthington drainage pump is used, driven through a friction clutch by a 35-h.p. "Ingeco" engine, using distillate. The following is quoted from the specifications for the plant, to contain 2 units when complete:

Equipment is desired for pumping 18,500 gal. of water per min. against a difference of level in suction and discharge basins of 5.5 ft. The outfit must be capable of pumping some water against a difference of level of 9 ft.

When pumping against less than 5.5 ft. difference of level, the pump or pumps should throw more water, but they must be so designed that they will not unduly overload the engine or engines, even with the difference of level reduced to zero.

Pumps. The pumps shall be of the centrifugal or screw type. They shall be direct-connected to the steam engine by means of a proper flange coupling, so that engine may be run without operating pump when coupling is disconnected.

Suction and Discharge Pipes. The suction and discharge pipes shall each be enlarged at the ends where they dip into the water to four times the area at the pump flanges, — the area to be that on a section at right angles to the axis of the pipe. Detail drawings of these pipes shall be submitted with each proposition.

Suction pipes shall be so designed and arranged that the water level may be reduced to elevation 12.0 C.D. without taking air. The discharge pipe must be so designed and arranged that it will be submerged when the water in outfall canal is at elevation 21.0 C.D. The suction and discharge pipes shall be constructed of ¼-in. steel.

RESULTS OF TEST

An acceptance test was run on May 26, 1914. The fuel used was distillate of 45 deg. Baumé at 86 deg. fahr., which reduced to 60 deg. fahr. is equivalent to 42 deg. Baumé; fuel weighing 6.80 lb. per gallon. The guarantee was to consume not more than 3.56 gal. per hour of kerosene or No. 2 Solar oil, with pump operating at a capacity of 9250 gal. per min. with difference of level of 5.5 ft.

It was impossible to run long enough to pump the water down to a lift of 5.5 ft., but it was agreed that the results obtained at 5 ft. lift would govern if satisfactory. On a 3-hour run with the pump operating at an average of 210 r.p.m., and an average lift of 4.56 ft., an average delivery of 10,190 gal. per min. was obtained at a fuel consumption of 3.98 gal. per hour.

On a later run, of about 2 hours, with the carburetor adjusted, the pumping unit, operating at practically the same speed as before and under an average lift of 5 ft., showed an average delivery of 9602 gal. per min., with a fuel consumption of but 2.99 gal. per hour.

DRAINAGE PLANT, WHITELAKE LAND COMPANY, FLORENCE, LA.

This pumping plant is a good example of a high-grade steam plant with simple Corliss non-condensing engines and efficient pumps. There are two 54-in. Worthington double-suction pumps which take suction from the principal drainage canal of the district and deliver it into a short outfall canal from which it overflows into the surrounding marsh outside of the tract. It is the intention later to dig this outfall canal to Whitelake, which will decrease the discharge head somewhat.

The pumps are direct-connected to two 16-in. by 36-in. Hamilton slow-speed Corliss engines, the pump shaft being solid with the engine shaft, thus doing away with flanged couplings. Part of the exhaust is utilized in a 400-h.p. Blake open feedwater heater which heats the feedwater to 210 deg.

The steam generating equipment consists of two 72-in. by 18-ft. Erie City return-tubular boilers rated at 150 h.p. each and designed for 125 lb. working pressure. The furnaces were arranged for oil burning by cutting down the bridge wall level with the gates and covering the grates with a checkerwork of fire brick. One Peabody burner is used under each boiler. Oil is fed to the burners by a Lockett fuel oil pumping outfit which takes the oil from a 15,000-gal. storage tank outside the building and heats and delivers the oil to the burners at a uniform pressure.

Each pumping unit has a capacity of 65,000 gal. per min. against 5 ft. difference in level between suction and discharge canals. Each unit has a 40 per cent overload capacity when pumping against 3-ft. head and is capable of pumping against a 10-ft. head at a reduced discharge, if necessary.

One unit of the plant was tested on October 1, 1912, by C. W. Okey of the U. S. Department of Agriculture and B. S. Nelson, Jun.Am.Soc.M.E. Its object was to find the actual operating efficiency at the time of the test. The plant was in charge of the usual operating engineer during the test and was run at a capacity which they had found suited the drainage needs at that time. The test was run simultaneously on the pumping unit and on the boiler and lasted six hours.

The quantity of discharge from the pump was obtained by a pitot tube in each suction of the pump. These suctions are tapering and the area of the suction pipes at the point of application of the pitot tube was carefully measured. The total head on the pump was obtained by mercury manometers attached to petcocks placed close to the suction and discharge flanges of the pump. The difference in elevation between the suction and discharge cocks was carefully measured as well as the area of the suction and discharge pipes at those points so that the proper correction for elevation and velocity head could be made in computing the total dynamic head. The static lift was gotten from the difference in elevation as shown on two gages which are installed in the suction and discharge canals close to the intake and discharge of the pumps. After the test the mechanical efficiency of the engine was obtained by breaking the vacuum and draining the pump and taking cards from the engine when running at the speed run during the test gave the friction load.

It was impossible to utilize the heater during the test as there was no way of delivering hot water to the weighing barrel. The tank pump was therefore piped to take water from the canal and deliver it into two barrels placed above a third barrel from which the feed pump took suction. The total amount of oil used during the test was recorded on a Worthington oil meter installed in the oil line to burners and at the end of the test this meter was calibrated by pumping oil through it at the same rate and pressure used during the test into a barrel placed on scales.

It will be noticed that the conditions were not the same as those for which the pump was built, the static head being less than 5 ft. and the discharge being considerably more than 65,000 gallons. This, no doubt, lowered the pump efficiency but it shows that the efficiency of the pump is very good over a wide range.

Condensed logs of the engine and pump tests are given in Tables 13 and 14 on the following page.

TABLE 13 TEST OF WHITE LAKE LAND CO.'S ENGINE AND PUMP UNIT NO. 2

October 1, 1912

R.P.M.	Indicated H.P.	Discharge in 100 Gal. per Min.	Static Head, Ft.		Total Dynamic Head, Ft.	Useful Water H.P.	Efficiency		
			Suction Gage	Discharge Gage			Pump, Engine and Piping	Pump and Engine	Pump
							%	%	Based on 91.1% Mechanical Efficiency of Engine and Pump from Friction Cards.
90	177.1	832	6.74	11.10	4.83	91 6	51.7	57.3	
89	171.4	845	6.60	11.10	4.79	96.2	56.2	59.8	
91	178.4	875	6.60	11.15	4.87	105.0	58.9	60.3	
90	181.6	855	6.60	11.14	5.21	98.4	54.2	62.2	
89	171.3	879	6.55	11.15	5.04	102.4	59.8	65.4	
94	201.1	892	6.50	11.20	4.87	106.0	52.8	54.7	
94	201.1	895	6.45	11.15	4.87	106.4	52.9	54.8	
91	196.6	875	6.40	11.20	5.15	106.2	54.1	58.0	
92	195.4	876	6.35	11.20	5.06	107.5	55.0	57.3	
92	197.3	853	6.30	11.20	....	....	....	....	
91	188.0	862	6.25	11.20	5.09	107.6	57.2	58.9	
91	188.0	830	6.25	11.20	5.23	103.9	55.3	58.4	
90	172.8	821	6.25	11.15	5.37	101.6	58.8	64.5	
91.1[1]	186.2[1]	860.77[1]	......	....	5.02[1]	102.7[1]	55.6[1]	59.9[1]	65.7[1]

[1] Mean value.

There are installed on each of these pumps a rate-of-flow meter which indicates the amount of water being pumped at any instant in thousands of gallons per minute. After the main test, each pump was run at various speeds varying from the slowest speed to the highest at which the pump will run and the indication on the flow meter noted from which figures a scale is made for each meter correct for that particular pump.

These meters work on the principle used in the venturi meter. Usually the enlarging discharge pipe is tapped at two points having quite different cross sections. When water is flowing the difference in the velocity heads at the two sections is read in feet of water.

TABLE 14 EFFICIENCIES AT VARIOUS CAPACITIES AND CONSTANT HEAD

R.P.M.	Indicated H.P.	Static Head, Ft.	Dynamic Head, Ft.	100 Gal. per Min.	Useful Water H.P.	Efficiency	
						Pump and Engine	Pump, Eng. and Piping
						%	%
71	83.8	....	....	....	....	....	....
61	122.2	5.15	5.81	619	80.5	74.3	65.9
82	132.7	4.85	5.24	696	85.3	69.5	64.3
85	154.5	4.95	5.08	767	96.0	63.8	62.2
89	183.4	4.95	5.08	812	101.5	57.0	55.3
98	225.9	4.95	5.08	900	112.6	51.1	49.9
101	249.7	5.10	5.55	951	122.6	53.3	49.1

The steam generating equipment consists of two 72-in. by 18-ft. Erie City return-tubular boilers rated at 150 h.p. each and designed for 125 lb. working pressure. The furnaces were arranged for oil burning by cutting down the bridge wall level with the gates and covering the grates with a checkerwork of fire brick. One Peabody burner is used under each boiler. Oil is fed to the burners by a Lockett fuel oil pumping outfit which takes the oil from a 15,000-gal. storage tank outside the building and heats and delivers the oil to the burners at a uniform pressure.

Each pumping unit has a capacity of 65,000 gal. per min. against 5 ft. difference in level between suction and discharge canals. Each unit has a 40 per cent overload capacity when pumping against 3-ft. head and is capable of pumping against a 10-ft. head at a reduced discharge, if necessary.

One unit of the plant was tested on October 1, 1912, by C. W. Okey of the U. S. Department of Agriculture and B. S. Nelson, Jun.Am.Soc.M.E. Its object was to find the actual operating efficiency at the time of the test. The plant was in charge of the usual operating engineer during the test and was run at a capacity which they had found suited the drainage needs at that time. The test was run simultaneously on the pumping unit and on the boiler and lasted six hours.

The quantity of discharge from the pump was obtained by a pitot tube in each suction of the pump. These suctions are tapering and the area of the suction pipes at the point of application of the pitot tube was carefully measured. The total head on the pump was obtained by mercury manometers attached to petcocks placed close to the suction and discharge flanges of the pump. The difference in elevation between the suction and discharge cocks was carefully measured as well as the area of the suction and discharge pipes at those points so that the proper correction for elevation and velocity head could be made in computing the total dynamic head. The static lift was gotten from the difference in elevation as shown on two gages which are installed in the suction and discharge canals close to the intake and discharge of the pumps. After the test the mechanical efficiency of the engine was obtained by breaking the vacuum and draining the pump and taking cards from the engine when running at the speed run during the test gave the friction load.

It was impossible to utilize the heater during the test as there was no way of delivering hot water to the weighing barrel. The tank pump was therefore piped to take water from the canal and deliver it into two barrels placed above a third barrel from which the feed pump took suction. The total amount of oil used during the test was recorded on a Worthington oil meter installed in the oil line to burners and at the end of the test this meter was calibrated by pumping oil through it at the same rate and pressure used during the test into a barrel placed on scales.

It will be noticed that the conditions were not the same as those for which the pump was built, the static head being less than 5 ft. and the discharge being considerably more than 65,000 gallons. This, no doubt, lowered the pump efficiency but it shows that the efficiency of the pump is very good over a wide range.

Condensed logs of the engine and pump tests are given in Tables 13 and 14 on the following page.

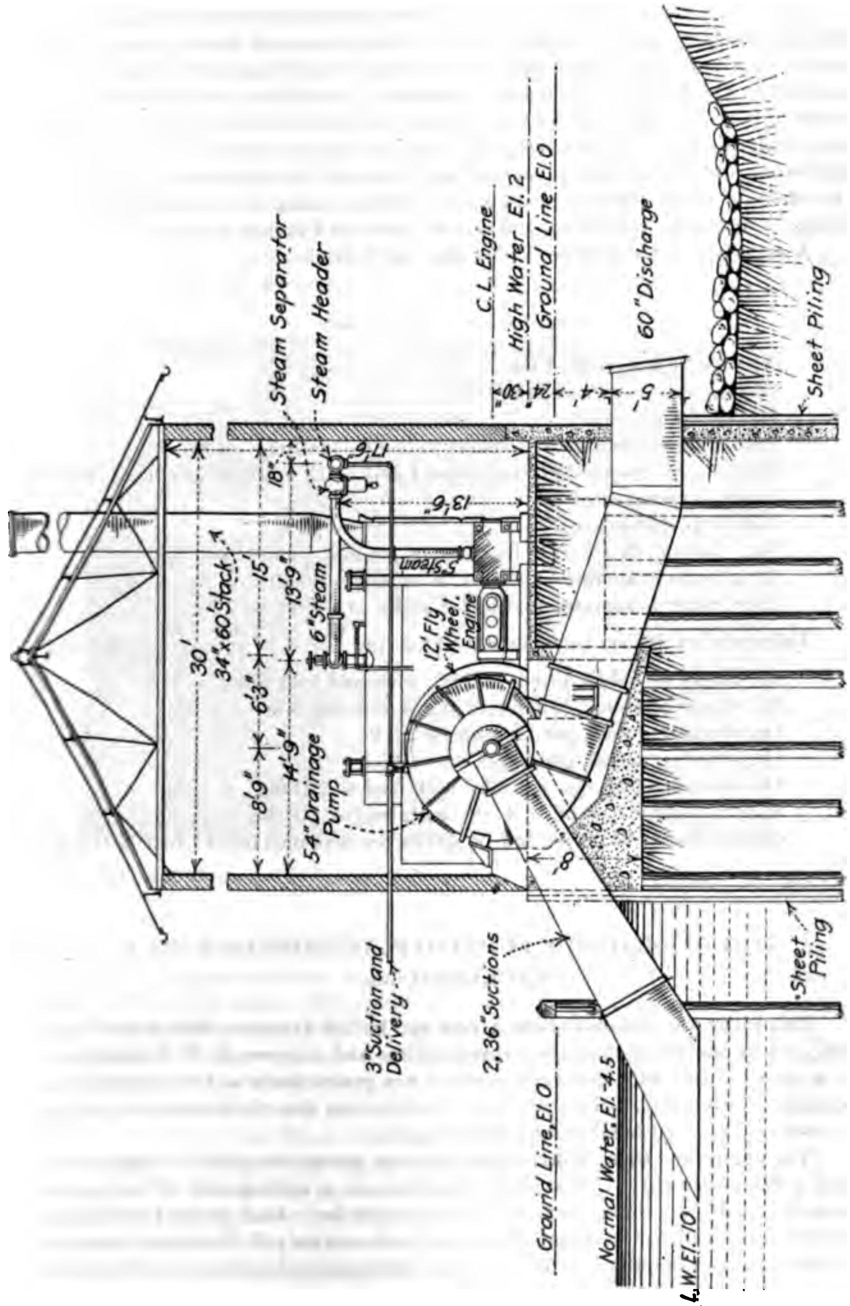


FIG. 35 ELEVATION OF WHITELAKE PUMPING PLANT

These meters are of great benefit in large drainage and irrigation plants as they furnish a means of keeping record of the amount of water pumped per minute, and if the total length of time the pump is run is known, the total water handled by the pump can be readily computed. The White Lake Land Co. intends to keep records of the amount of water pumped which, together with the area drained, should furnish valuable data for future drainage work. While calibrating the flow meters advantage was taken of this opportunity to obtain the efficiency of the pump at various capacities by taking a set of readings consisting of discharge, total head, static head, speed and indicator cards.

A summary of the results of the boiler test follows:

Results of Boiler Test

Duration of test, 6 hr. 4 min.
Average steam pressure, 120.0 lb.
Average feed temperature, 72.0 deg. fahr.
Total water evaporated, actual = 31,555 lb. per hr. = 5199
Total water evaporated, equivalent f and a 212 = 37,560 per hr. = 6189
Factor evaporation, 1.19
Boiler h.p. developed, 179.3
Total fuel oil, lb., = 2975 lb., = 8.18 per min. = 490.8 per hr.
Ratio water evaporated and fuel oil, actual = 10.61 lb.
Ratio water evaporated and fuel, f and a 212 = 12.63 lb.

Efficiency of boiler, basis 18,500 B.t.u. per lb. $= \frac{12.63 \times 966}{18,500} = 66.1\%$

Lb. oil per acre-inch, lifted 4.72 ft., with cold feed water = 2.58
Lb. oil per acre-inch, lifted 4.72 ft., at 210 deg. fahr. = 2.27
Lb. of water, actual, per i.h.p.-hr. = 27.9
Lb. of water, actual, per w.h.p.-hr. = 47.5
Lb. of oil per i.h.p.-hr. = 2.63; with feed at 210 deg. = 2.32
Lb. of oil per w.h.p.-hr. = 4.48; with feed at 210 deg. = 3.95
Cost of fuel at $1.00 per bbl. of 320 lb. per acre-inch lifted 1 foot = 0.15c.

TEST OF DRAINAGE PUMPING PLANT, DISTRICT NO. 4, RACELAND, LA.

This plant was installed about a year ago by the Louisiana Meadows Company, under the direction of their vice-president and engineer, A. T. Dusenbury. It is used to drain 4466 acres of reclaimed wet prairie lands and the machinery consists of two units, duplicates in every way, except that the pumps are driven respectively by right-hand and left-hand engines.

The pumps are 48-in. Worthington drainage pumps designed for high speed and a flat power curve. The pump impellers are a combination of the screw propeller and the ordinary centrifugal pump impeller. Each pump has double suction pipes attached to 48-in. elbows and both suction and discharge pipes are enlarged at their outer ends. Wooden flap valves are used to cover the discharge ends while the pumps are being primed.

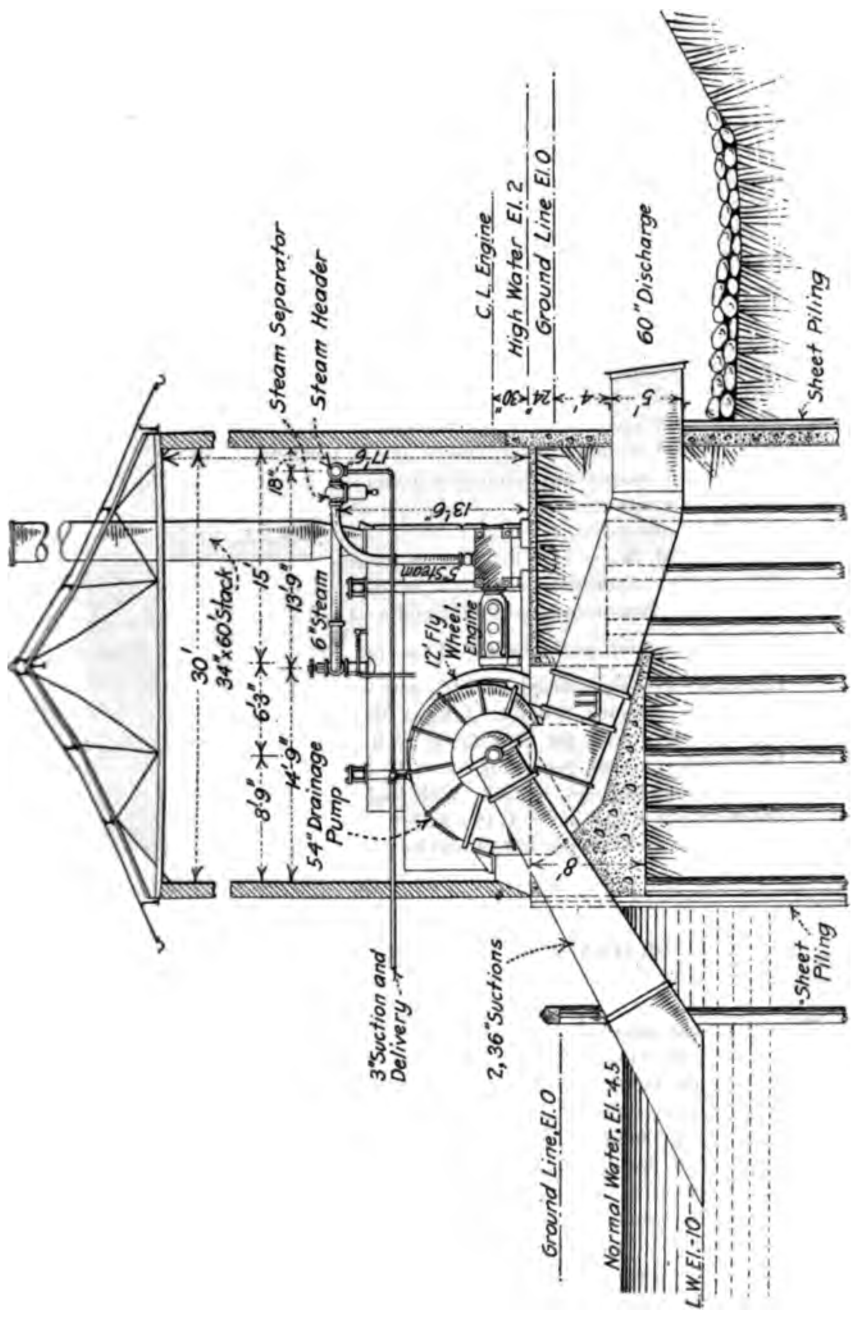


FIG. 35 ELEVATION OF WHITELAKE PUMPING PLANT

in the quantities of water measured in the two pipes as the right-hand pipe invariably carried more water. Without doubt, this condition had some effect on the efficiency of the pump and consequently reduced the over-all efficiency. Unequal depths of water below the suction pipes also had something to do with the formation of the eddy as well as the fact that a higher velocity is found in the suction canal normally near its center line.

The condition of the intake was known to be bad but it was not convenient for the owners to remedy defects at this time as they expect to have a dredge in the district at a later date to do the work at a lower cost than by hand labor.

The results are stated in pounds of oil per hour per useful water horsepower and per acre-foot of water elevated one foot. The economy of the plant can best be realized by a comparison with the steam pumping plant of the White Lake Land Co. near Gueydan, La., previously described. The equipment consists of high-grade simple non-condensing Corliss engines and Worthington volute pumps. When pumping against a head of approximately 5 ft. the fuel consumption of the steam plant was to that of the internal combustion engine plant as 4.28 to 1 for equal output of work.

Results of Tests

The test which extended over a period of nearly 6 hours, was run with a varying difference of levels in the canals, the lift increasing from 3.55 ft. at the beginning of the run to 4.96 ft. at the close. The speed of the pump was held uniform at 195 r.p.m., but the total water pumped varied with the change in actual lift, ranging from 60,050 gal. per min. at the lower lift (3.55 ft.) to 54,700 gal. per min. at the higher lift (4.96 ft.). The corresponding useful water horsepower varied from 53.8 h.p. at the lower lift to 68.5 h.p. at the higher lift, the consumption of fuel per useful water horsepower being 1.132 lb. in the former case and 0.887 lb. in the latter. The fuel consumption per acre-foot of water elevated one foot was 1.55 lb. of oil in the case of the lower lift and 1.21 lb. of oil for the higher lift.

ONISHI PUMPING PLANT, MACKAY, TEX.

On September 11, 1907, a test was made of the pumping plant of R. Onishi, constructed in that year on the Colorado River about 8 miles below Wharton, Tex. This plant waters 1200 acres of rice during the irrigating season, but has a capacity for 2000 acres. It was designed and installed by A. M. Lockett & Co., Ltd., and is an example of the application of machinery to special conditions at medium cost and with fair efficiency.

There is a great variation in stage on the banks of the river where the plant is located and in order to place the pump low enough to avoid an excessive suction head at low stage, and to protect the pump from overflow in case of high water, a circular reinforced concrete pit 22 ft. in diameter and 20 ft. deep was built, into which the pump and engine were placed on a foundation at the bottom. The suction pipe extends through the side of the pit and is 148 ft. long and 28 in. in diameter, except near the pump where it is reduced to 24 in. in diameter. The discharge is through a 24-in. vertical pipe, coming up from the pump within the pit and by means of a 90-deg. turn with a 5-ft. radius, ending in the flume where the area was increased to about twice the area of the pipe. A rectangular flap valve is provided in the flume to allow priming of the pump and

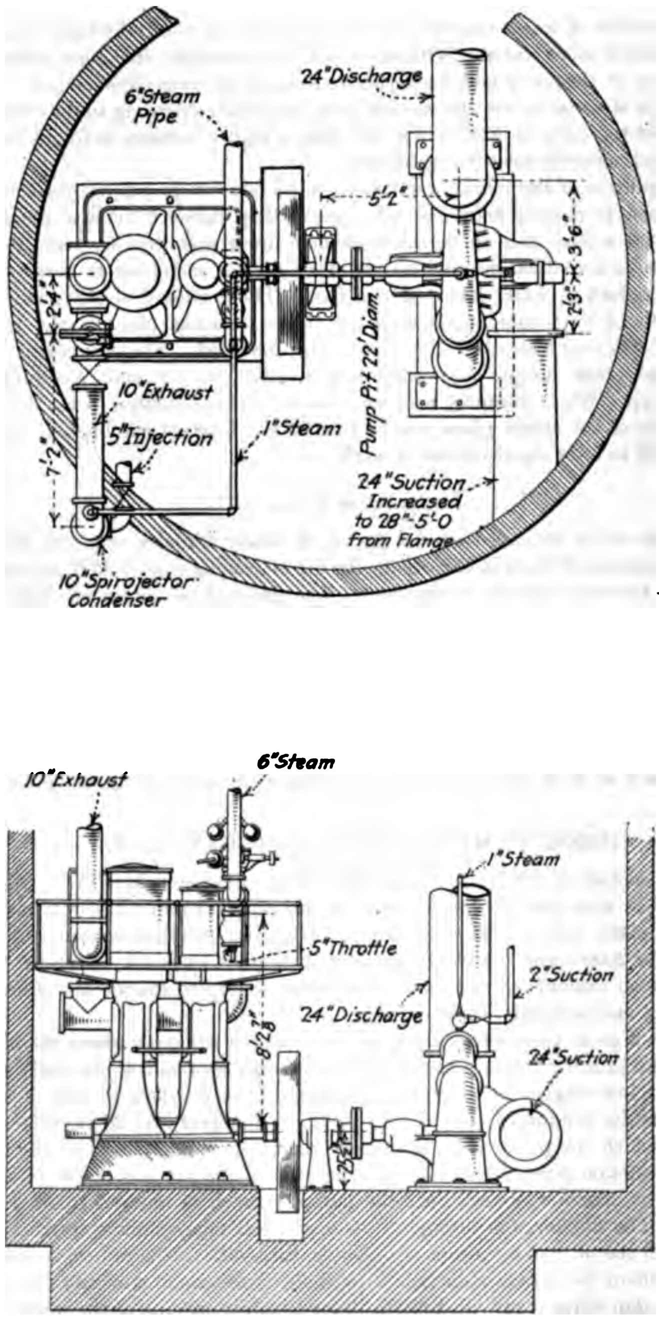


FIG. 38 ENGINE AND PUMP, ONISHI IRRIGATION PLANT, MACKAY, TEX.

to prevent the backward flow of water through the pump, as the discharge opening was under water.

The engine is a vertical, cross-compound Buckeye engine, 13 and 22 by 16-in., direct connected to the pump, which is a 24-in. Worthington volute single-suction centrifugal pump. The engine is provided with a governor which at ordinary speeds is inoperative, but which will prevent it from running away in case the pump loses its priming. The boiler is a Babcock & Wilcox water-tubular cross-drum boiler with 1550 sq. ft. of heating surface. The floor of the boiler room is a little above the level of the river bank.

A barometric condenser is used. The condensing water is taken in at the flume and after passing through the condenser is conducted by a pipe to the river. The end of the pipe is submerged and the difference of level between flume and river is so great that the water siphons over without the aid of a pump. A steam ejector is used to prime the condenser and start the water flowing. The boiler feed and oil pumps exhaust into an open heater. The fuel is crude oil, which cost $1.45 per barrel of 42 gal. at the plant.

While it was desirable that a complete boiler test be made, this was found impossible without cutting out the heater and pumping cold water into the boiler. For this reason it was omitted as it was thought important to measure the fuel used and the output of useful work from the pump, under ordinary conditions of running.

The test lasted for 6 hours, and is summarised below:

Summary of Results of Test on Onishi Plant

Mean revolutions per minute for engine and pump	271.7
Mean indicated horsepower	230.4
Actual lift, ft	37.23
Head on pump	39.43
Discharge, cu. ft. per sec	32.5
Useful water horsepower	137.2
Water horsepower, basis of head on pump	145.6
Efficiency of engine, pump and pipe, per cent	59.5
Efficiency of engine and pump, per cent	63.0
Efficiency of pump (estimated), per cent	70.0
Steam pressure, lb. per sq. in. at boiler	141.4
Temperature of feed water, deg. fahr	172.4
Temperature of injection water, deg. fahr	79.7
Temperature of discharge from condenser, deg. fahr	100.3
Mean vacuum, in. of mercury	23.2
Fuel oil per hour, barrels consumed	1.21
Fuel oil per hour, barrels guaranteed	1.37
Fuel oil per minute, lb	6.46
Fuel oil per i.h.p.-hour, pounds	1.68
Fuel oil per useful water h.p.-hour, lb	2.82
Fuel oil per pump horsepower-hour, lb	2.66

MOORE'S BLUFF PUMPING PLANT

This plant is located on the West bank of the Trinity River, about ei miles from Dayton, Tex. Although it contains no novel equipment, it is wort of comment as indicating what may be considered good practice in the des of a plant for rice irrigation. The only feature which may be considered out the ordinary is the high lift, it being designed to deliver 36,000 gal. per m against a maximum lift of 57 ft. measured between levels. The pumping pl was built by A. M. Lockett & Co., Ltd.

In the design of this plant it was realized that on account of the high lif was necessary to secure high economy in fuel, but it was also kept in mind t in an irrigation plant reliability and freedom from accident were of prime imp ance. It was therefore decided to install no special equipment, such as sup heaters and economizers, and to select main units of a type that would economical without adding to the amount of equipment to be cared for.

The equipment consists of a Worthington centrifugal pump, direct-connect to a cross-compound, condensing four-valve Hamilton engine, 18 and 36 by 30 i with no flywheel other than the pump impeller. There are three Stirling boil of 214 rated h.p. each, two set in one battery and the other with a separ setting. The condenser is of the jet type. The feed water is heated by a Webs open heater, from which the exhaust from all the auxiliaries is led. Fuel oil used.

The pump has double suction pipes 28 in. in diameter at the flanges and larging to 36 in. At the intake end the diameter is again increased to 5 ft. T horizontal discharge pipe is enlarged at the pump from 36 in. to 48 in. diamet About 9 ft. from the pump there is an elbow from which the pipe is carried u hill at an angle of approximately 45 deg., and at the top of the hill it again comes horizontal, enlarging as it enters the flume. The area at the discha end is about 12 sq. ft.

A four-hour test was run on the plant July 17, 1910, after it had been in op ation under practically constant conditions for several hours. The object of test was to determine the amount of fuel oil used per day, but incidentally efficiencies of the pump and piping were obtained. Two boilers only were u during the test.

The quantity of water pumped was determined by means of a Price curr meter of the tail type, used in the flume connecting the discharge pipe with main canal. The width of the flume is 8.61 ft. and the depth increased fr 3.84 to 4.03 ft. as the canal filled up. The height through which the water elevated was measured by means of two gages set in the suction intake and charge canal respectively. To determine the head the suction pipes were dril near the flange for ½ in. air valves to which a mercury column was connected rubber tubes.

A Worthington meter was used to measure the fuel oil, after calibrati The amount of fuel oil used to operate the electric light plant was computed fr the known load and the water rate of the engine, upon which point the previ test also furnished some information. No attempt was made to measure feed water.

The following tabulation gives a summary of results:

RESULTS OF TEST, MOORE'S BLUFF PLANT

Average i.h.p.	793.44
Average revolutions per minute	173.40
Average water pumped, cu. ft. per sec	88.28
Average water pumped, gal. per min	39,707.00
Actual lift, level to level, ft	55.88
Useful water horsepower	557.50
Efficiency engine, pump and piping, per cent	70.20
Efficiency pump and piping (basis eng. eff. = 92 per cent)	76.30
Head on pump, ft	57.51
Efficiency, engine and pump, per cent	72.30
Efficiency, pump	78.50
Pounds of fuel oil, used in 4 hours	4,375.00
Barrels of 320 pounds, used in 4 hours	13.671
Barrels of 320 pounds, used in 24 hours	82.026
Barrels of 320 pounds, used for electric lighting	0.85
B.t.u. per pound of oil	19,347.00
Total barrels of 320 lb., used in 24 hours	82.876
Pounds of fuel oil per i.h.p.-hr. (basis day run)	1.378
Pounds of fuel oil per useful water h.p.-hour	1.96

The showing made is exceptionally good. The oil used per i.h.p-hour (1.378) is the lowest the writer has found in a long experience in testing pumping plants of this class used for irrigation. The pounds of fuel oil per useful water horsepower-hour (1.96) is the lowest the writer has found in any pumping plant of its size and capacity, regardless of the type of pump.

THE GARWOOD IRRIGATION PLANT, GARWOOD, TEX.

The Garwood Irrigation Plant is equipped with a 36-in. Worthington pump, driven by a Hamilton high-speed cross-compound condensing engine, 18 by 36 by 30 in. Water is pumped from the Colorado River and as there is considerable difference of level at the pumping, the pump and engine were placed in a concrete pit in order to keep a reasonable suction lift at low water. The pit is high enough to prevent flooding in time of high water.

On August 15, 1913, a preliminary test was made by the writer, during which the plant was operated in practically the same manner as had been customary during the irrigation season. The boiler furnished steam for the oil pump, for atomizing the oil, for the boiler-feed pump and for the service pump. The last-named pump was run at a high rate of speed to prevent losing its priming and the supply tank was consequently overflowing continuously. As this pump is very wasteful of steam, it added considerably to the load on the boiler and to the fuel bill.

The quantity of water pumped was measured by means of two pitot tubes in the suction pipes. Piezometer openings in the piping near the pump were used in reading "head on pump" by means of mercury manometers. Gages were established in the river and in the flume, and from the readings of these gages the lift was determined.

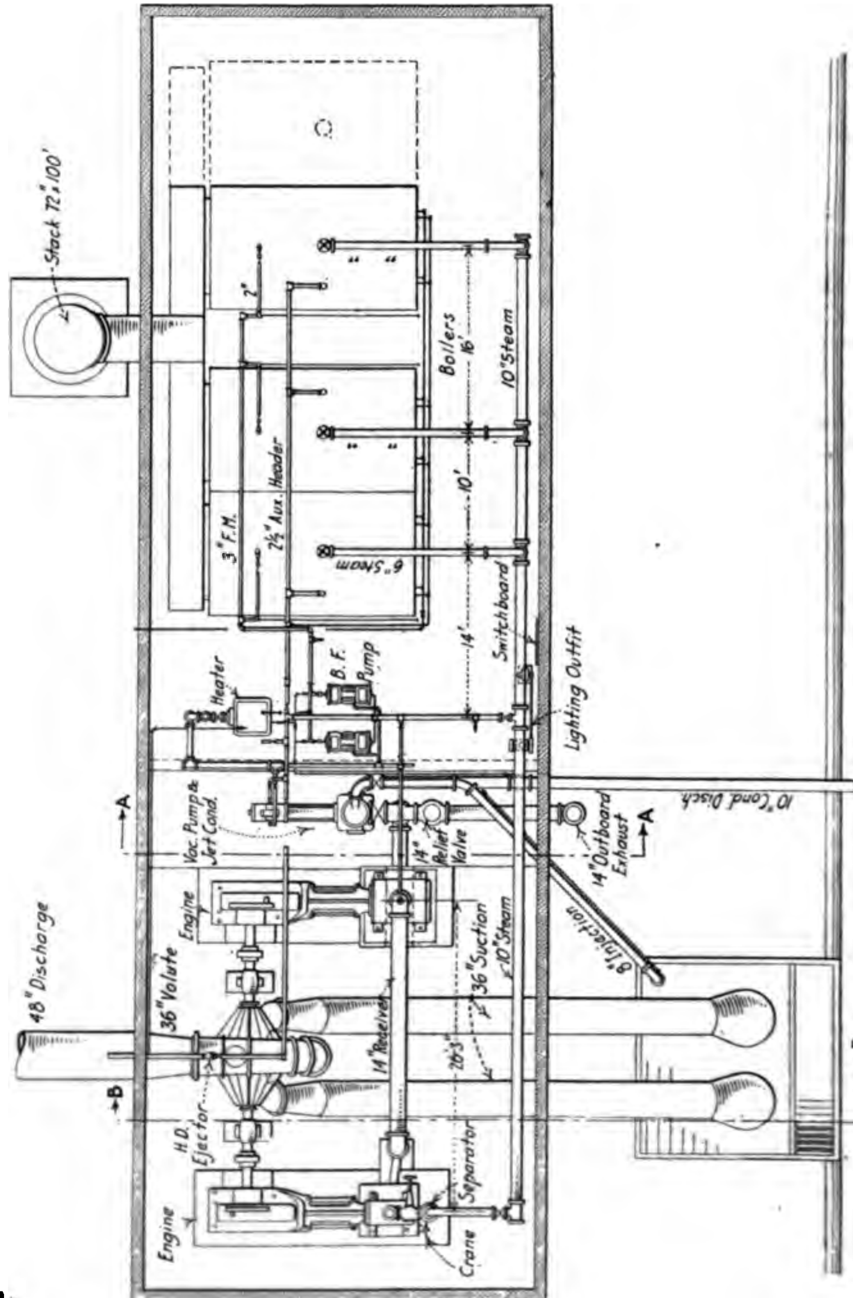


FIG. 39 GENERAL PLAN OF PUMPING STATION, MOORE'S BLUFF RICE COMPANY

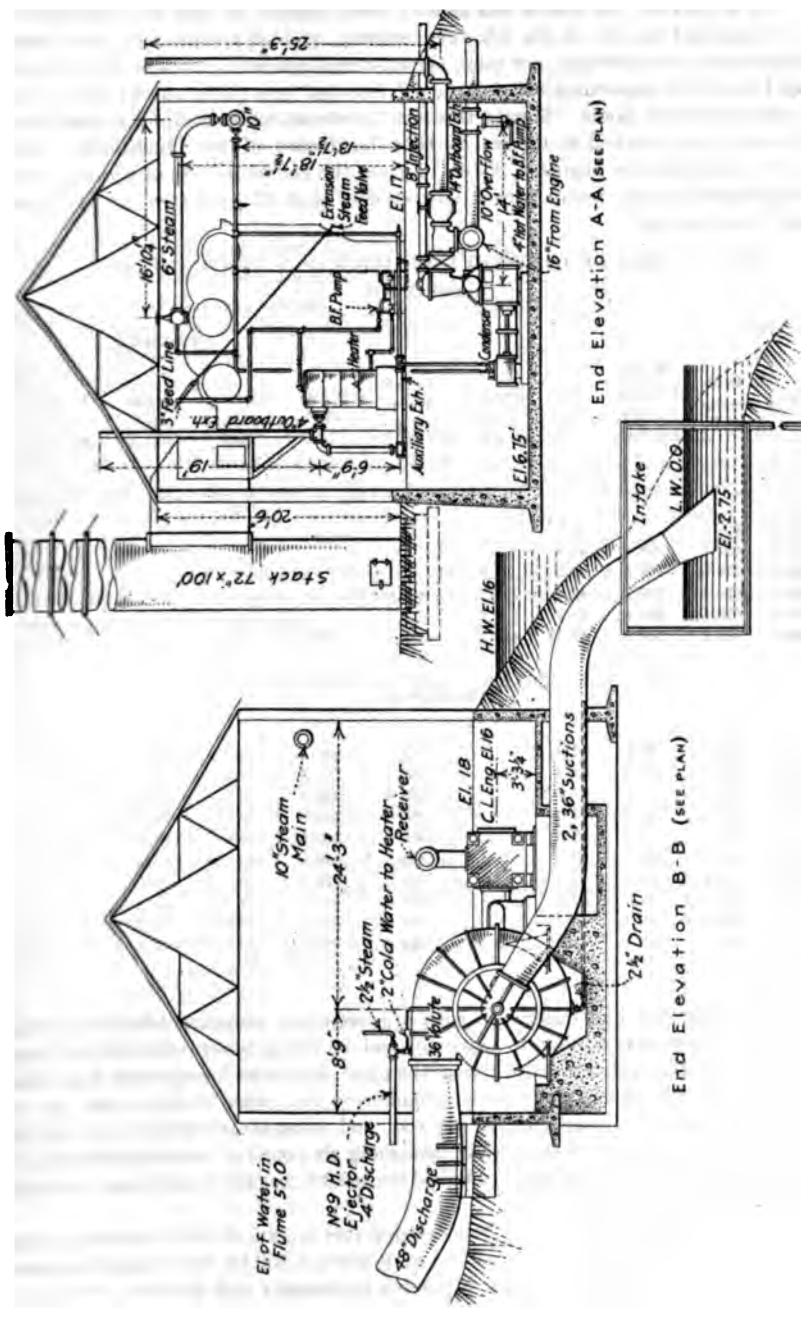


FIG. 40 ELEVATION, MOORE'S BLUFF PLANT

On August 16, the engine was slowed down slightly, so that the amount water pumped would, at the lift then existing, give approximately the sa useful work, as 40,000 gal. per min., elevated through 40 ft. When the prop speed had been approximately determined, the test was started at 11 a.m. a continued for four hours. During this test the steam service pump was not ru but water was supplied to the service tank by means of the "bulldozer," ru by the small gasoline engine. Table 13 gives the condensed log of these test The efficiency of the pump varied between 80.7 and 92.2 per cent, or 86.5 pe cent on an average.

TABLE 15 TEST OF GARWOOD IRRIGATION CO.'S PUMPING PLANT

R.P.M.	Indicated H.P.	Water Pumped, Cu. Ft. per Sec.	Ft. Lift	Head on Pump, Ft.	Useful Water H.P.	Pump H.P.	Efficiency: Engine, Pump and Piping	Efficiency: Engine and Pump	Efficiency: Pump. (Eng. 92.5%)
August 15, 1913									
							%	%	%
150.6	696.8	....	41.7	....	....	....	....	....	....
149.0	657.8	98.2	41.9	....	....	....	....	....	....
150.0	712.5	103.4	42.3	....	....	....	....	....	....
149.2	670.1	103.2	42.4	....	....	....	....	....	....
149.8	678.4	102.4	42.4	....	....	....	....	....	....
(Mean	683.1	101.5	42.14	439.4	488.0	510.0	71.4	74.7	80.7)
August 16, 1913									
146.8	489.9	80.9	42.0	....	385.0	399.0	78.4	81.3	87.0
146.4	492.3	75.0	42.0	....	356.5	369.5	72.3	75.0	81.1
146.4	489.2	74.7	42.0	....	355.0	368.0	72.5	75.2	81.2
146.8	504.7	81.6	42.0	....	388.0	402.0	76.9	79.7	86.1
146.8	521.8	84.5	42.0	....	402.0	416.5	77.0	79.8	86.3
147 0	531.9	90.5	42.0	....	430.5	446.0	80.7	83.7	90.5
146.8	539 5	87.1	42.0	....	414.0	429.0	76.7	79.6	86.1
146.8	514.5	89.0	42.0	....	423.0	438.5	82.3	85.3	92.2
146.6	518.4	85.3	42.0	....	406.0	420.5	78.8	81.7	88.3
(Mean	511.4	83.2	42.0	43.51	395.5	409.9	75.1	80.1	86.5)

The writer has had experience with a great many pumping plants having equipment somewhat similar to this one, and in which boiler tests have been made. From the probable water rate of the plant, the boiler horsepower has been computed for the two days. On the preliminary test, when doing work in excess of guarantee by more than 20 per cent and using steam wastefully in the service pump, the boiler was possibly furnishing an excess of capacity above its rating of from 30 to 35 per cent. During the second test the boiler was running at practically normal rating.

When it is remembered that boilers using fuel oil will operate continuously on overloads in excess of 50 per cent and even more, it will be seen that the plant may be crowded with ease if water is wanted in quantity and quickly, while for ordinary running, the boiler capacity is more than ample. There was no way to measure the steam used by the engine even if a boiler test could be made, as the steam used by auxiliaries could not be separated from that used by the engine.

The pump made the best record of any centrifugal pump ever tested by the writer. The lift was high and conditions favorable to a high efficiency. By way of comparison the following results are given:

Name of Plant	Pounds of Fuel Oil per Useful Water H. P-Hour
Neches Canal, First, Lift, Beaumont, Tex.	2.09
Jennings Canal, Jennings, La.	2.16
Sabine Canal, Vinton, La.	2.29
Pierce Estate Canal, Wharton, Tex.	2.84
Moore's Bluff	1.96
Garwood	1.82

From these figures it will be seen that the Moore's Bluff plant, which is the best of the list with the exception of the Garwood plant, uses more than 7 per cent more oil than the latter to do the same amount of useful work. The last two plants have shown the feasibility of pumping water to greater elevations than was thought commercially possible a few years ago. A properly designed steam driven pumping plant may be operated against a lift of 55 ft. at a cost that makes rice irrigation an attractive proposition, especially if the canal reaches a reasonably large acreage.

Summary of Results, Garwood Irrigation Plant

	1913 Aug. 15	1913 Aug. 16
Duration of test, hours	2	4
Mean h.p.	683.1	511.4
Worthington centrifugal pump 36 in. disk. noz.		
Amount of water pumped, cu. ft. per sec.	101.5	83.18
Amount of water pumped, gal. per min.	46,125	37,455
Difference of level between suction and discharge, ft.	42.14	42.00
Head on pump, ft.	43.94	43.51
Temperature of water pumped		85
Useful water horsepower	488.0	395.5
Pump horsepower	510.0	409.9
Efficiency, engine pump and piping, per cent	71.4	75.1
Efficiency, engine and pump, per cent	74.7	80.1
Efficiency, pump (eng. 92.5 per cent)	80.7	86.6
Quality of steam		99%
Total fuel oil burned	2273	2875
Barrels of fuel oil (320 lb.) per 24 hours	85.3	53.9
Barrels of fuel oil (320 lb.) per 24 hours to raise 40,000 gal. per min. 40 ft.		54.8
Pounds of fuel oil per h.p.-hour	1.66	1.41
Pounds of fuel oil per useful water h.p.-hr.	2.33	1.82
Cost to pump 2 acre-ft., oil at $1.30 .... $0.85		

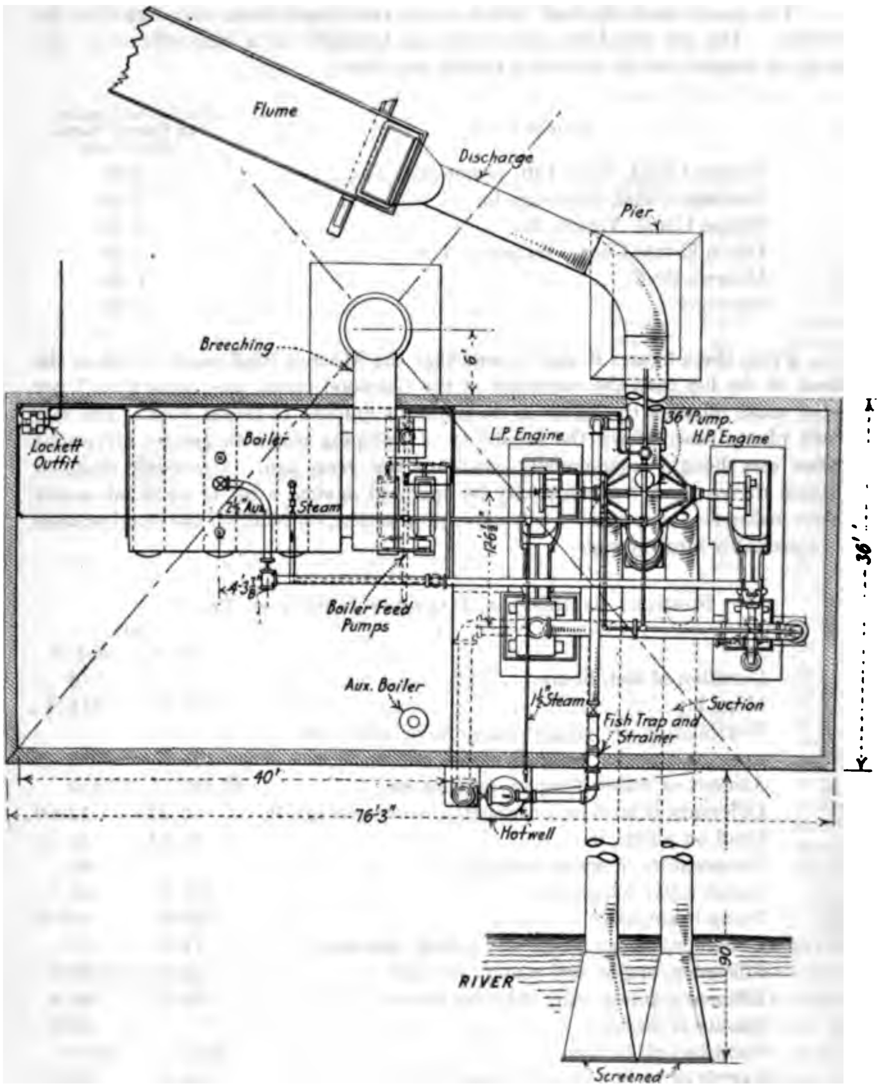


FIG. 41 PLAN OF PUMPING PLANT, GARWOOD IRRIGATION COMPANY, GARWOOD, TEX.

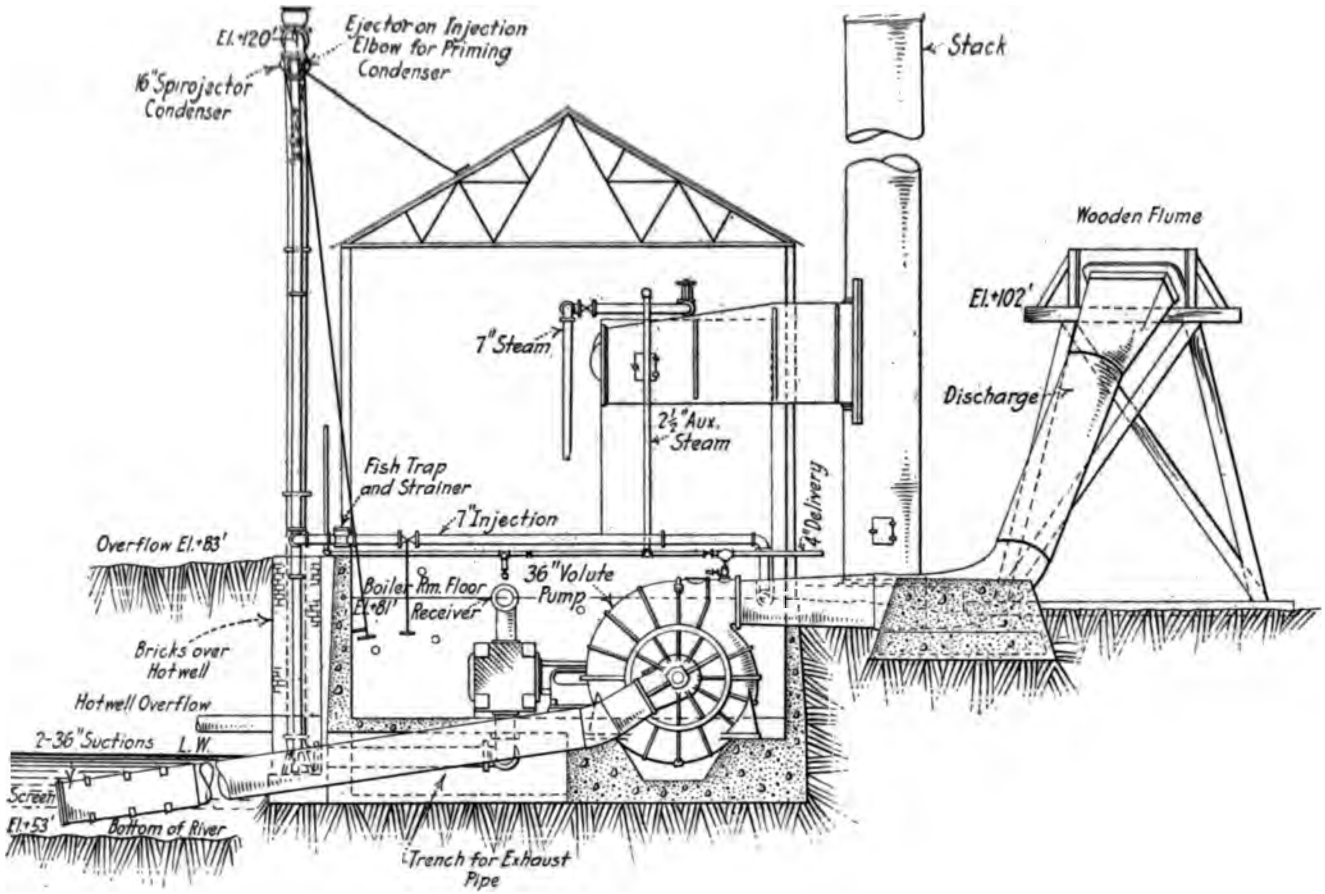


FIG. 42 END ELEVATION OF PUMPING PLANT, GARWOOD IRRIGATION COMPANY

DISCUSSION

JOHN J. HARMAN (written). In the Upper Mississippi Valley drainage pumps of other than the centrifugal type have hardly been used at all, although there has been quite a variety of different kinds of prime movers. The most predominant type of machinery used in the earliest plants was the low-head centrifugal pump, driven by a simple slide-valve engine, the steam being supplied at about 100 lb. pressure from a fire-tube boiler. The machinery in these plants was far from being high-grade in materials and workmanship, and often the pumps did not have sufficient capacity to handle the drainage water under wet-weather conditions. Another common fault was that the power units were not properly proportioned to the loads imposed upon them, so that in some cases where the pumps were abundantly large, the engines and boilers were too small to carry the load when the river was at a high stage. Add to this unstable foundations and poor management, and we have a fair conception of these earlier plants.

The quality of machinery installed has gradually improved. Centrifugal pumps especially designed to meet the peculiar requirements of this service have been developed, which show an efficiency of from 65 to 80 per cent, or even higher, throughout the working range. Careful designs of suction and discharge pipes have been made, reducing the total pipe-friction and velocity-head losses to a mere fraction of the losses in the earlier plants. The power-producing equipment has been vastly improved, both in quality of workmanship and materials and in economy of operation. Foundation designs of a highly permanent character and fireproof building construction have been introduced.

Although there is a rapidly growing trend toward high-grade pumping-plant construction, there are still a great many who willfully close their ears to the advantages of features such as the following: A reduction in the perpetual yearly operating cost of 50 per cent or more; permanent foundations having a life of 100 years or more, particularly the part below normal water level, which offers great construction difficulties; a substantial fireproof building having a life of 50 years or more; high-grade heavy-type machinery on stable foundations, having a life of 30 years or more; a substantial, reliable, confidence-breeding pumping plant, the heart of the reclamation project, which adds dollars to the value of every acre of land in the district. It is encouraging to note that during the past five or six years a number of this type of high-grade pumping plants have been constructed.

As typical of these high-grade plants, a brief description of three plants is given below, together with a statement of the economy guarantees and the results of the acceptance tests.

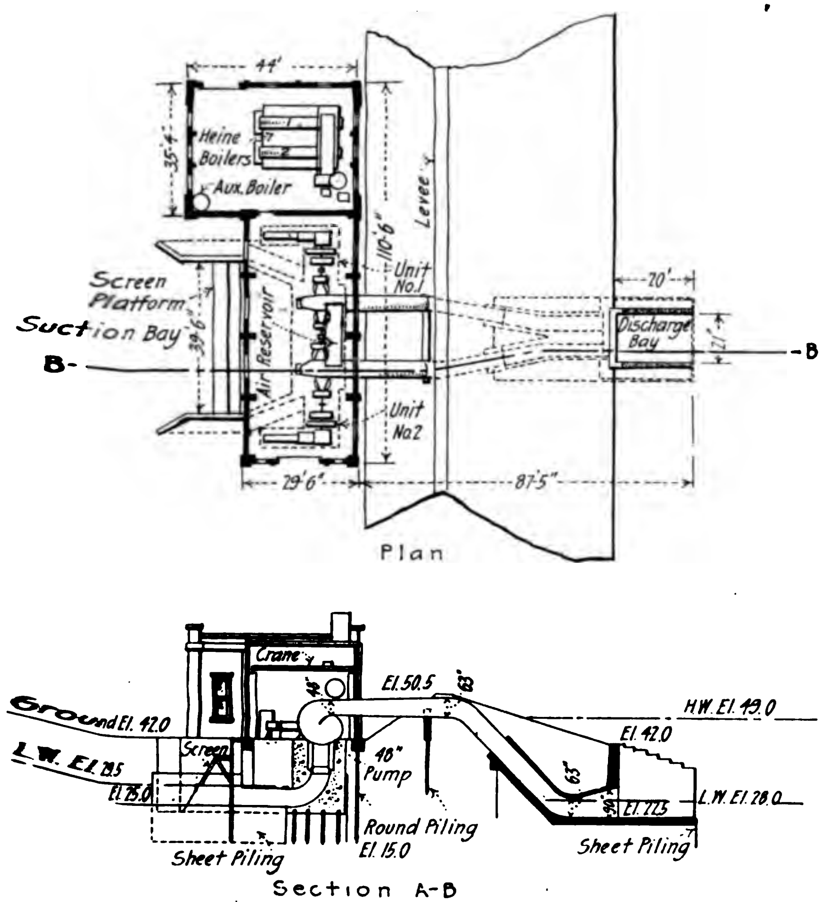


FIG. 43 PUMPING PLANT OF THE ELSBERRY DRAINAGE DISTRICT

Elsberry Drainage District. The Elsberry Drainage District is located in the State of Missouri on the west bank of the Mississippi River, about 70 miles above St. Louis, Missouri. It contains 25,000 acres of reclaimed land. About 113,000 acres of hill drainage is diverted to the Mississippi River above and below the district by means of diversion channels.

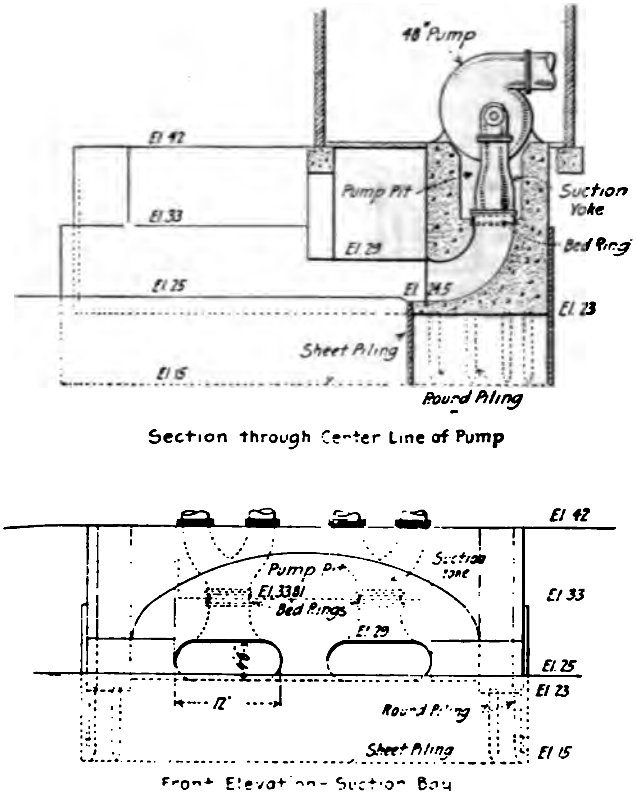


FIG. 44 FOUNDATION DETAILS OF THE PUMPING PLANT IN FIG. 44

The pumping plant contains the following principal equipment:

- Two 48-in. Alberger low-head centrifugal pumps
- Two 21-in. × 26-in. Fleming-Harrisburg heavy-duty uniflow engines
- Two American jet condensers driven by silent chain drives from the main engine shafts
- Two 168-h.p. Heine water-tube boilers, equipped with shaking grates, soot blowers and induced-draft apparatus
- One Webster open feedwater heater, together with miscellaneous auxiliaries and appurtenances.

Fig. 43 shows the general layout of this plant, while Fig. 44 shows on a larger scale the suction-intake arrangement. Particular attention is directed to the permanent character of this design, the intakes being built in solid concrete and the discharge pipes being encased in concrete below normal water level. The friction loss in this intake is so small as to be almost unmeasurable, and the total friction and velocity-head loss in the intake and discharge pipes, with the pumps operating at normal capacity, is less than one foot.

Another feature of this design is the cross-connection of the two units, so that one or both pumps may be operated from either engine. For ordinary pumping heads up to about 7 ft. both pumps are operated by one engine, thus securing much better engine economy at the low heads.

The acceptance test of this plant was made in 1915. The pumping head during the test averaged 7.52 ft., and by converting the data to guarantee conditions the following results were secured:

a	Pounds of 10,800-B.t.u. coal per acre-foot of water pumped against a 6-ft. static head	36.6
b	Pounds of 10,800-B.t.u. coal burned, per useful water horsepower	4.43
c	Contractor's guarantee: Pounds of 10,800-B.t.u. coal per acre-foot of water pumped against a 6-ft. static head	38

Muscatine-Louisa Drainage District No. 13. The Muscatine-Louisa Drainage District No. 13 is located in the State of Iowa on the west bank of the Mississippi River, just below Muscatine, Iowa. It contains 21,400 acres of reclaimed land, and 30,000 acres of additional land drains into the district and has to be pumped. Much of the land is of a sandy nature, so that the run-off is delivered slowly to the ditches. The plant contains the following principal equipment:

Two 54-in. Worthington low-head centrifugal pumps
One 36-in. Worthington low-head centrifugal pump
Two 20 × 30-in. Filer & Stowell uniflow engines, direct-connected to the 54-in. pumps
One 14 × 30-in. Filer & Stowell uniflow engine, direct-connected to the 36-in. pump
Two 7-in. Schutte-Koerting eductor condensers
One 5-in. Schutte-Koerting eductor condenser

Two 200-h.p. and one 100-h.p. Heine water-tube boilers, equipped with Heine superheaters, shaking grates, soot blowers and induced-draft apparatus

One Cochrane feedwater heater and purifier, together with miscellaneous auxiliaries and appurtenances.

The steam is generated at 175 lb. pressure and carries 100 deg. superheat. The condensers produce a vacuum of 26 in., the circulating pumps being driven by silent chain drives from the main engines.

The general layout of this plant is shown in Fig. 3, p. 47, of the U. S. Department of Agriculture Bulletin 304.

The contractor's guarantees on the economy of this plant read as follows: The coal per acre-foot when pumping against a 6-ft. static head is estimated from manufacturer's guarantees not to exceed 46 lb., and will not in any event exceed 56½ lb. when tested as required by the specifications.

The acceptance test of this plant was conducted in June, 1915. The pumping head at the time of making the test was only about 4 ft., but by converting the data to the guarantee conditions by means of the contractor's efficiency and capacity curves, the final results secured were as follows:

a Pounds of 10,500-B.t.u. coal per acre-foot of water pumped against a 6-ft. static head	35.4
b Pounds of 10,500-B.t.u. coal burned per useful water horsepower	4.29
c Contractor's guarantee: Pounds of 10,500-B.t.u. coal per acre-foot of water pumped against a 6-ft. static head	56.5

Fabius River Drainage District. The Fabius River Drainage District is located in the State of Missouri on the west bank of the Mississippi River, opposite Quincy, Illinois. The district contains 14,250 acres of reclaimed land. The North and South Fabius Rivers, having a drainage area of about 1,000,000 acres, are diverted around the lower end of the district, and about 19,000 acres of hill-land drainage is diverted to the river at the upper end of the district.

The pumping plant is now under process of construction, and when completed will contain the following principal equipment:

Two 42-in. Southwark double-suction low-head drainage pumps

Two 250-h.p. Southwark-Harris valveless oil engines (Diesel principle)

One 60-h.p. Winslow high-pressure oil-burning steam boiler for auxiliaries. The layout of this plant is very similar in general character to the layout of the Elsberry Drainage District plant.

While this plant has not been tested, it is expected to meet the contractor's guarantees without difficulty. The guarantees are as follows:

a	Pounds of 18,500-B.t.u. 25-deg. Baumé fuel oil per acre-foot of water pumped against a 6-ft. static head	7.76
b	Pounds of 18,500-B.t.u. 25-deg. Baumé fuel oil burned per useful water horsepower	0.94

Another phase of the problem of drainage pumping, fully as important as design and construction, is the proper maintenance and economical operation of the plant from day to day and from year to year. Reclaimed lands of this type are encumbered perpetually to the extent of this maintenance and operation cost. In some districts, through uneconomical operation, mismanagement and other causes, this has become a very serious burden, amounting in one instance to considerably over $3.00 per acre per year, as compared with 30 to 60 cents per acre in fair-sized districts having economical and well-managed plants. There are a great many factors affecting the cost of operation which cause the operating cost to vary quite materially, even when the plant is well operated and maintained, such as size of district, kind of plant, exceptionally long periods of high water in river, exceptional heavy rainfall, rainfall not well distributed, seepage through leaky levees, and last but by no means least, spring water.

Experience along the Illinois and Upper Mississippi Rivers has demonstrated that spring water may easily add 100 per cent to the normal yearly pumping requirements. The unavoidable variations in operating cost from year to year make it too hazardous for the operation to be contracted on the lump-sum basis, and the only plan which has been used to any extent in this region to increase the operating efficiency of the plants is to employ a supervising engineer, who makes tests at the plant under various operating conditions and determines the most economical pump speeds throughout the range of pumping head. He also keeps in touch with the plant by means of daily reports from the station engineer, containing data which, combined with the pump ratings, will give a complete plant test for each operating day. If anything goes wrong with the machinery in any way, it is thus discovered at once and corrected before any material amount of money is wasted in inefficient operation.

Another common cause for inefficiency is the partial obstruction of the waterways in the pumps or pipes by some foreign substance. The pump capacity is also oftentimes seriously reduced without the knowledge of the attendant by air pockets in the summit of the discharge pipes. The data contained in the daily operating reports would immediately disclose troubles of this kind to the supervising engineer.

Another valuable service rendered by the supervising engineer is in seeing to the proper maintenance of the plants and making the necessary repairs.

These plants always have a season during the summer and fall when they are completely shut down for long periods, and this affords an excellent opportunity for overhauling the plant, making any necessary repairs, and putting all the equipment into first-class operating condition for the next pumping season.

Engineering supervision insures reliable pumping service, economical operation, and proper plant maintenance, results which are of far more value to the drainage district than the cost of the service.

As previously stated, the experience in the region under discussion seems to indicate that engineering supervision of operation is fully as important to the continued success of these pumping-plant projects as engineering design and supervision of construction, but it would appear that the general run of landowners and district officials must be educated considerably before engineering service will be employed in this connection to any great extent. The writer feels that the Drainage Investigation Division of the U. S. Department of Agriculture might very properly make this matter of pumping-plant operation and maintenance the subject of a thorough investigation and bulletin.

S. L. Menge[1] (written). The function of a drainage-pumping station is to maintain the water level on its suction side lower than

[1] Mgr. Constr. Dept., Woodward, Wight & Co., Ltd., New Orleans, La.

the water on its discharge side. When it has just ceased to do this it becomes an obstruction instead of an asset, and its crucial test is its capacity to discharge more and more water with less and less power per unit discharged as the lift decreases and the necessity to pump therefore is most urgent. At the most crucial point, if the power and the pumping capacity are properly adjusted, one will be able to pump with the same power expenditure from three to five times as much water per applied brake horsepower with pumps having the characteristics of the new Wood screw pumps as with pumps having characteristics of the earlier type of pumps installed in New Orleans — or with the best design of centrifugal units.

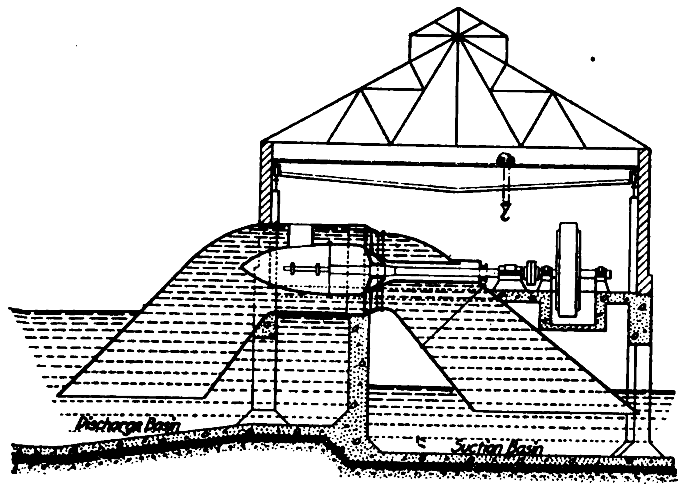


Fig. 45 Section through 12-ft. Wood Screw Pump

The experience heretofore of the Sewerage and Water Board has been that during great storms, when lifts were much below the rated lifts, the older forms of pumps overloaded their motors and also overloaded the power stations, so that all pumping-station units could not be maintained in service. With the eleven additional screw pumps which the Sewerage and Water Board proposes to install, and less than enough additional power to operate them at their rated lift, because the flow at rated lift can never supply all pumps, this condition will be reversed. The decreasing amount of power required to operate the new pumps as their lift decreases will more than compensate for the excess amount required by the old ones when their lift decreases and the necessity to pump is at its greatest.

According to a test conducted by Prof. W. H. P. Creighton, Dean of the Department of Technology, Tulane University, New Orleans, the Wood screw pumps of the New Orleans Sewerage and Water Board give their best efficiency at 7- to 8-ft. lift, and show a better figure for 7½ ft. than the best of the Board's old centrifugal pumps at its best point, 11 ft. The advantages of accessibility, operation, first cost, etc., are, however, more than sufficient to dictate the use of the screw pump, even if the power-consumption characteristics were the same as those of the centrifugal.

The best of the centrifugal pumps adds only 50 per cent to its capacity per applied brake horsepower as its lift drops from its point of maximum efficiency, which is 11 ft., down to zero, while the new screw pump increases its capacity per applied brake horsepower 300 per cent from its lift for maximum efficiency at 7½ ft. down to zero.

In a system like that in New Orleans, operated electrically with power and pump capacity designed to give the required output under those low-lift conditions which require the greatest quantity of discharge, the power requirement to operate all units will be much less with screw than with centrifugal pumps. Since as the lift increases the available flow approaching the stations decreases and fewer pumping units have to be operated, the power required to operate the pumps at a low lift will still suffice to operate enough units as the lift increases to take care of the decreasing flow. With pumps driven by power units capable of some speed variations, the characteristic of the screw pump is such that available power can be utilized at an increased speed at lifts far below the rated lift without the serious sacrifice of efficiency which an increase of speed entails with centrifugal pumps.

Professor Creighton's test was made with a view towards determining the amount of water that one of the new 12-ft. Wood screw pumps could lift, and the efficiency of performance. The result shows that the pump raised 559 cu. ft. per sec. through a height of 5.6 ft. and 519.8 cu. ft. per sec. to 7.6 ft., the pump making between 75 and 76 r.p.m. and the efficiency varying between 76 and 80 per cent.

It should be noted that these pumps are not designed to fit a certain condition of lift, which may be more or less arbitrary and theoretical, but are designed to work with the maximum economy on widely varying lifts, such as actually obtain in service on practically every drainage problem.

The New Orleans units, with capacity of 500 to 600 cu. ft. per sec., can be advantageously installed in a space only slightly wider than the suction bells themselves (22 ft.) and depth of building of 50 ft. inside. They are particularly free from any vibration and, therefore, require little foundation mass. They operate at relatively high speeds, being particularly suited for direct connection to electric motors at constant speed. They are entirely self-oiling, no bearing coming in contact with the water or being subject to grit and wear.

B. Stanley Nelson (written). The author shows clearly the importance of enlarging the ends of the suction and discharge pipes to recover velocity head, because this head, unless converted to pressure head, and so recovered, represents a considerable portion of the total lift.

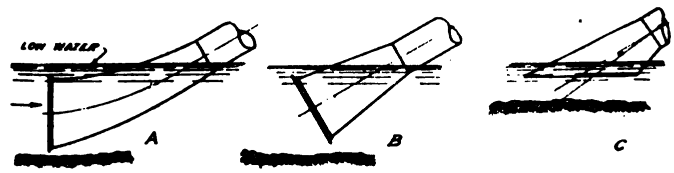


Fig. 46 Typical Suction-Pipe Ends

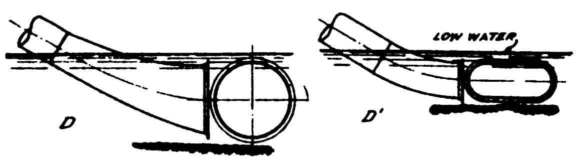


Fig. 47 Typical Discharge-Pipe Ends

An important feature in the design of these expanding ends on the suction pipes is the angle of the plane of entrance with relation to the water line.

The writer has had occasion to design and observe the behavior of a number of installations where one or other of the three forms of ends or modifications of them was used, and the conclusion is that type C (Fig. 46) is the best. Both *A* and *B* (particularly *A*) cause suckholes to be formed — sometimes with a considerable depth of water over them, while with *C* it is possible to pump down to within a few inches of the end of the pipe. Suckholes are objectionable, as they admit air to the pump, and air in a centrifugal reduces both the capacity and efficiency.

The design of the discharge end of the pipes is not so importar other than the ends should expand sufficiently to recover velocit head. In the typical drainage plant of this region, where the pum discharge into an outfall canal close to the plant foundation, ca must be taken to direct the flow of water so as not to cause scourir of the canal bottom, and the design shown at D, Fig. 47, works ve well. The flattened end, D', is used on large pipes to reduce tl depth at which the pipes must be set to maintain the siphon sea

In his description of the test of the Louisiana Meadows oil-e gine-driven plant, the author has touched on a feature of the typic drainage plant of this region, where pumps of large size with ve short suction pipes are set in a dam across a canal, and that is tl relation of the suction basin to the efficiency of the pumps.

At first thought, it would seem that if the suction basin we deep enough at the end of the pipe, it would need no further co sideration, and that if there were some obstruction, the only effe would be to slightly increase the friction head due to restriction area at entrance. It has been found, however, that where tl suction basin is too shallow, or, if deep enough, some irregularit of contour causes greater velocity of approach at one side than tl other, or if there is a mud lump in front of the suction, the effect not only to reduce the capacity, but also the efficiency of the pum This is probably due to the fact that any disturbance at entrance go on up through the suction into the pump impeller because the pi is too short to allow the velocities across a section of the pipe to smoot out before reaching the pump. In the case of the Citrus plan tests showed an increase in efficiency of about 8.5 per cent aft dredging out the suction basin. The remedy is to give more thoug to the design of the suction basin.

A very interesting and important feature of two of the plan mentioned, namely, the motor-driven Citrus plant, and the oil-e gine-driven Raceland plant, is the power characteristics of tl pumps. The usual centrifugal-pump power curve called for a gre increase in power at heads much lower than normal working head, the speed were kept constant. This presented no great difficul in the case of the steam-engine-driven unit, for the steam engine flexible, and the speed and cut-off could be varied to suit the hea But in the case of the motor-driven plant it was, for obvious reasor preferable to use a constant-speed induction motor; it was also d sirable to have the motors fully loaded at normal head, to get be motor efficiency. The problem was to design a pump which, wh run at constant speed at all heads by a motor which would be fully loaded at normal head, would not overload that motor at zero or intermediate heads. The pump builders met these conditions by using an impeller which is a combination of the centrifugal types — the pumps have a characteristic such that the ratios of the power input at 2, 4, 6, 8 and 10 ft. to the power at normal head of 6 ft. are as follows: 0.93, 1.00, 1.00, 0.99 and 0.88.

In the case of the Raceland plant, the pump designer had additional problems to contend with in the characteristics of the oil engine. An oil engine should not be overloaded beyond about 10 per cent of its full-load rating; its speed, for best results, is fixed or nearly so; and in this size unit this speed is considerably higher than the usual speed for a pump with 4- or 5-ft. lift. Direct connection was necessary because the initial cost of the oil engine precluded the use of a gear reduction such as was justifiable in the case of the motor-driven plant. By the use of the combination impeller mentioned, the designer has built these pumps with a practically straight-line power curve, the variation in horsepower at all working heads between zero and 7 ft. lift being about 5 per cent.

A. M. Shaw (written).[1] The author's careful exposition of the various types of pumps which have been used and the description of the conditions under which they have operated will do much toward reducing the labor and annoyances of engineers engaged in preparing plans for future installations for this kind of service. Reference to this paper should suffice as an answer to many of the questions over which we are now wasting time and patience.

Charles C. Trump (written). There are new types of pumping engines already entering this field. The four-stroke-cycle type of Humphrey pump, which uses internal combustion in direct contact with the liquid to be pumped, is hardly suitable for lifts under 15 ft., on which it depends for the compression which makes for efficiency of combustion. A two-stroke type now in process of development gives promise of working well on heads of 8 to 10 ft., but is somewhat high in first cost.

An entirely new type of pumping engine, known as the Tube pump, based in part on the principle of the Humphrey pump, has

[1] Hibernia Bldg., New Orleans, La.

just been developed, and gives promise of working out, not only for large volumes with low lifts, but especially for small plants taking water from wells at lifts from 50 to 100 ft. or more for irrigation.

Designs of pumping engines on the uniflow principle, which uses a large ratio of expansion in one cylinder without condensation losses, and with few valves, have already been made by the Stumpf Una-Flow Engine Co. for many classes of service. A flywheel-and-crank engine with double-acting plunger pump and steam cylinder with but two steam valves of the poppet type; a screw pump with vertical shaft and engine crank rotating in a horizontal plane; a marine type engine for high-pressure steam (500 to 600 lb.) for driving a propeller in a tube for lifting water only 3 to 5 ft., and power cylinders of the uniflow type for Tube pumps have all been worked out and some of them are now in the shop. It should not be long, therefore, before some of these modern pumping engines are in the service this paper describes.

For generating high-pressure steam economically in large units the Winslow safety high-pressure boiler is now being built, the first unit of 500-h.p. capacity now being made for the City of Chicago. A small unit is being worked out with automatic control of water-feed pump and fuel-oil burner for use with a Tube pump of 800 gal. per min, capacity at 40- to 60-ft. lift, suitable for irrigation from artesian wells on individual farms without the need of skilled mechanical attendance.

When some of the above punping engines have proved their advance in economy, measured by the actual total cost of pumping under conditions of which I hope to have intimate knowledge, it would be a great pleasure to me to present before the Society for discussion one or more papers on this subject, with the hope that they may prove as interesting as this one by Professor Gregory.

S. T. Wellman also thought that the conditions described in the paper were ideal for the working of the Humphrey gas pump which has been developed in England and in which a mixture of gas and air is exploded in direct contact with the liquid to be pumped. In this type of pump heavy moving pistons are avoided and the only moving parts are the valves. The gas is made in a common gas producer and does not have to be purified. The economy of the pump is very high, while the cost is very low, as all the weighty parts are not finished at all.

The Author said that an installation of the Humphrey pump for irrigation work existed at Del Rio, Texas, but it did not yet appear that this type of pump is yet suited to the low lifts in drainage work. Most of the pumping in the Gulf Coast country for drainage and agriculture is done with a difference of level of from 3 to 6 ft., often of the former. There are great differences to be overcome in applying the Humphrey pump to these low lifts. For high lifts he thought this pump had a future, particularly in parts of Texas and other parts of the country where lignite and cheap fuels from which producer gas may be made are available.

At present some form of internal-combustion engine seems to offer the best solution of the power problem for pumping plants in this section. Steam plants use much more fuel for an equal output of work and the boiler problems are often perplexing, especially in drainage plants where the feedwater available is liable to cause trouble. Compound condensing Corliss engines have been used quite generally in irrigation and have also been used in some drainage plants. Superheated steam has been employed only in a few cases.

Designers of pumping plants must keep in mind the service for which the plant is intended. If the operation is to be continuous and the lift is through a considerable height it will pay to embody devices to save fuel, and superheaters, economizers, expensive engines, and all other refinements may be justified.

Irrigation plants are operated but 75 to 80 days per year. Drainage plants are often operated but 20 days per year. Due to the nature of the work the operation of both types of plant in a humid country, with occasional heavy rainfall, is intermittent, and this is especially true of drainage plants. The fuel economies for these short runs must be balanced against the interest and depreciation of the more expensive plant. Furthermore, reliability of operation is of greater importance than economy, and complications decrease reliability.

The rapid development of internal-combustion engines, the varying price of fuel oil, and the troubles incident to boilers and to steam plants generally, will all have a bearing on the future pumping plants for irrigation and for drainage.

At present pumps of the centrifugal and the screw types appear to hold undisputed sway in the field of low-lift pumping.

No. 1535

MECHANICAL EQUIPMENT USED IN THE PORT OF NEW ORLEANS

By William von Phul, New Orleans, La.
Member of the Society

The Port of New Orleans, about 100 miles from the Gulf of Mexico, extends for nearly fifteen miles along the Mississippi River, from Westwego and Southport above the city on the northwest, to Chalmette five miles below Canal Street. The principal commercial development is on the east bank of the river, but on the west bank, opposite the city, there are wharves and landings at Westwego, Harvey's Canal, Gretna, Algiers and McLellanville.

2 In volume of business, combining imports and exports, New Orleans is the second port of the United States, being exceeded only by New York. New Orleans is described as a two-way port, the import and export tonnage being practically equal, a condition obviously very favorable to the transportation interests. The value of the combined imports and exports averages about $1,000,000 per day.

3 By the peculiar laws in force, there can be no private ownership of river-front property to the exclusion of the public. The State of Louisiana owns outright the larger part of the commercial front and has the right to use all of it, appropriating any land needed and expropriating any improvements on such property if placed under grants of the proper authorities.

4 Until 1901 the facilities of the Port of New Orleans were provided by private interests through lease of the river frontage. The wharves were temporary structures; there were no sheds; there was no machinery, and charges assessed against vessels were considered excessive.

5 The Board of Commissioners of the Port of New Orleans was created by the Legislature in 1896 and in 1901 secured control

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of the river front. The first Board which took charge of the facilities in 1901 was somewhat handicapped through having no other available funds than revenues produced by charges upon shipping. After a few years, however, the Legislature authorized the Board to issue bonds, and, with the proceeds of these bonds, wharves and sheds were constructed. In 1914 other bonds were issued for the construction of a Public Cotton Warehouse, and in 1915 for a Public Grain Elevator; and financial arrangements were made which will permit the prompt execution of further improvements as adopted.

6 On account of this lack of funds in the early years, the facilities first constructed consisted solely of wharves for landings. The construction a few years later included closed sheds, and the more recent projects include, in addition, switch tracks generally on the wharf front, paved roadways along the rear of the wharves connecting with the city streets, and mechanical devices suited to the purpose of the wharf. It is planned to rebuild promptly a large part of the existing wharf system, including in the rebuilding the additional facilities mentioned.

7 The river front of New Orleans is now administered by two State Boards: (1) The Board of Commissioners of the Orleans Levee District, which is empowered to acquire land and to build protective levees; (2) The Board of Commissioners of the Port of New Orleans, which has jurisdiction over the port and prescribes rules for operating vessels and controls the commercial use of the port facilities.

8 The wharves are served by the Public Belt Railroad, owned by the City of New Orleans and operated through the Public Belt Railroad Commission. The Belt Railroad connects with all railroads reaching the city and switches cars to and from the wharves and industries at a flat rate of two dollars per car. The Belt Railroad Commission was created in 1904 and began work in 1905, since which time construction has been carried on continuously.

9 The river is from one-half to three-quarters of a mile in width, and the depth within ten feet of the wharves ranges from 30 to 70 ft. Unloading can be done in midstream, but most of the vessels land broadside along the wharves, which extend out from 50 to 100 ft. from the bank. Easy access to this whole wharfage front is afforded by city streets and by railroad tracks in the rear of and upon the wharves. The levees for flood protection are very wide in the commercial district, and slope back so gradually as to be hardly noticeable. They serve the purpose of marginal streets

along the water front, the wharves being built at the elevation of the levees and out beyond them.

10 While the developed river front is about 15 miles long, there are 41.4 miles of river frontage under the control of the Port Commission. The developed harbor has a water area in excess of 11 square miles. A plan of the harbor, showing the coördination of rail and water transportation, is given in Fig. 1.

11 The satisfactory design and construction of wharves and buildings on the banks of the river are made more difficult by the wide range of the water level, amounting to about 21 ft., and by the current and large amount of silt borne by the water.

12 The wide range of level, in particular, complicates the design of machines for transferring cargo between wharves and vessels, involving special arrangements for adjusting the ends of conveyors and for adjusting overhead booms without fouling the rigging and wireless equipment of ships.

MECHANICAL EQUIPMENT

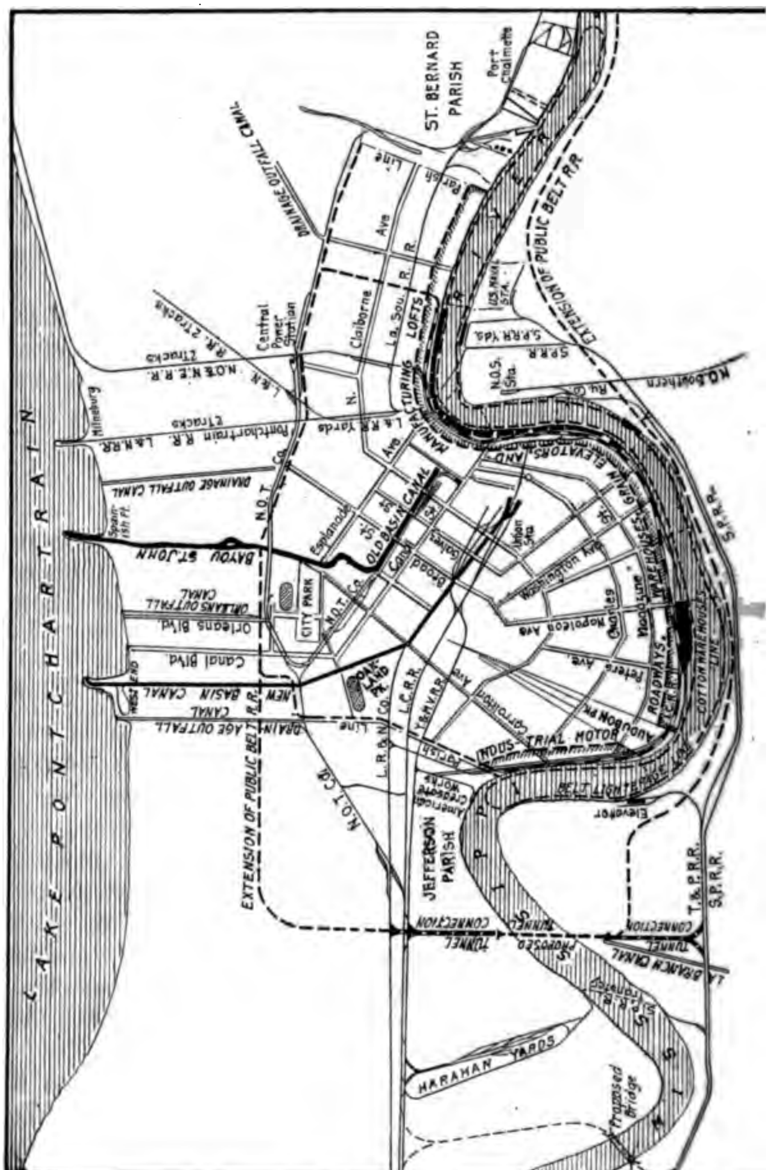
13 The mechanical equipment which has been developed in New Orleans is widely scattered through the harbor and includes several installations of special interest relating to the important industries of the port.

BANANA UNLOADERS

14 Banana importations at New Orleans exceed those at any other port, amounting for the year ended June 30, 1914, to 16,583,000 bunches, or equivalent to an average of 320,000 bunches per week.

15 The handling of this business by mechanical devices has been developed until practically all of the bananas are now unloaded by machines. The design of this machine is complicated by the necessity for providing for a wide range of water level and for ships of various dimensions. The machine consists of a steel A-frame structure traveling lengthwise of the wharf on two rails about 21 ft. centers and carrying a structural-steel main boom about 50 ft. in length, stepped about 20 ft. above the dock level.

16 On this boom, at a distance of about 30 ft. from the heel, is stepped an auxiliary boom twenty or more feet in length, from the end of which is swung the vertical leg of the conveyor. These two booms afford almost any desired combination of vertical and horizontal adjustment, while at the same time the main boom is sufficiently short to clear the stays and wireless rigging of the ships.


LAKE PONTCHARTRAIN
ST. BERNARD PARISH
JEFFERSON PARISH
BAYOU ST. JOHN
CITY PARK
OLD BASIN CANAL
NEW BASIN CANAL
EXTENSION OF PUBLIC BELT R.R.
EXTENSION OF PUBLIC BELT R.R.
DRAINAGE OUTFALL CANAL
ORLEANS OUTFALL CANAL
Canal Blvd.
Orleans Blvd.
Esplanade
Claiborne
Washington Ave.
Napoleon Ave.
Peters Ave.
MANUFACTURING LOFTS
ELEVATORS AND GRAIN WAREHOUSES
COTTON WAREHOUSES
INDUSTRIAL MOTOR
BELT LIGHTERAGE
TUNNEL CONNECTION
HARAHAN YARDS
Proposed Bridge
Union Sta.
Central Power Station
Milneburg
Port Chalmette
S.P.R.R.
T.&P.R.R.
N.O. Southern
L.&N.R.R.
N.O.T. Co.
L.C.R.R.
Elevator

These booms and the framework carry the conveyor proper, consisting of two endless chains spaced about 5 ft. apart and running over pairs of sprockets at the foot and head of the conveyor frame, at the pivot points of auxiliary and main booms, and at additional points at the discharge end of the unloader.

17 The distance between chains is maintained by pipe spreaders and through bolts spaced at about 3-ft. centers, attached to which is a 4-ft. canvas belt, with sufficient slack between each pair of spreaders to hold a bunch of bananas when the chain is moving vertically.

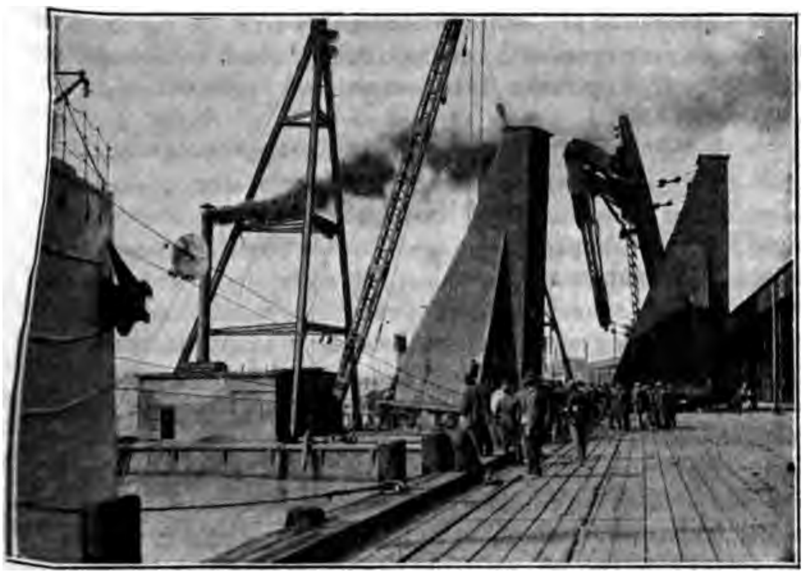


Fig. 2 Banana Unloaders, Pauline Street Wharf

Upon reaching the point of discharge, the bunch is forced out of the pocket by a roller behind the belt and drops upon a short horizontal conveyor, from which the bunches are picked up by hand, inspected, sorted, and loaded in cars, or disposed of locally. The entire machine is driven through gearing and clutches by one 15-h.p. motor. The capacity of each machine is upward of 2600 bunches per hour.

18 Three groups of these unloaders, consisting of 10 units, are owned by the Dock Board, the total investment being about \$90,000. The most recent installation is at the Pauline Street wharf, and consists of four machines costing approximately \$37,500. This installation is shown in Fig. 2. The section of the wharf on

which the machines operate is of pile-and-timber construction, with steel shed, about 360 ft. by 260 ft. As the area is covered, rain does not interrupt the discharging of cargo. The wharf is served by eight transverse spur tracks accommodating five cars each; and as one ship may carry 40,000 to 45,000 bunches or about 120 carloads, the entire track capacity may be filled and moved three times in less than a day's time. The cost to the Dock Board of this wharf with its shed and tracks was about $1.40 per sq. ft. In the rear of the wharf the Public Belt Railroad has yard capacity for 200 additional cars.

19 The Board of Commissioners makes a charge for the use of the banana conveyors of ¼ cent per bunch, which considerably exceeds the cost of operation, maintenance, and interest charges.

TELPHER SYSTEM, CHALMETTE SUGAR REFINERY

20 New Orleans receives large quantities of raw sugar from Cuba, Porto Rico and local production districts. The American Sugar Refining Company alone receives over 300,000 tons per annum at its large Chalmette refinery, just below the city. This plant has a well-equipped telpher system for receiving, storing and delivering raw sugar. This system operates between a wharf about 80 ft. by 800 ft. paralleling the river, and a storeroom about 250 ft. by 260 ft. located inside of the levee at a distance of about 200 ft. in the clear from the wharf.

21 The wharf is a pile-and-timber structure below the floor level and of heavy steel construction above, the shed covering the inshore 60 ft., while the outer 20 ft. is open. The storeroom is of heavy steel construction, with columns set on piles and double hardwood floor laid on packed sand bed. The column spacing on the wharf is approximately 20 ft. both ways, while in the storeroom the spacing is about 20 ft. by 25 ft. Both structures are enclosed with galvanized iron, and the two are connected by four elevated runways, for the telpher system, which are also entirely enclosed.

22 The track system for the telphers is divided into two sections, each consisting of a loop on the wharf and four loops in the storeroom. The telphers can be operated selectively over any of the loops of a section by means of a transfer or switch which connects the tracks as desired. The two track sections can be interconnected by a special transfer or switch on the wharf. The long wharf loop is equipped with a transfer or switch cutting off a large part of the run when ships are unloading at suitable points.

23 The telpher equipment consists of 25 machines; each has a capacity of seven bags of sugar weighing about 330 lb. each, or about one long ton, and is equipped with two travel motors of 6 h.p. each, on a swivel truck, and one hoist motor of 5 h.p. The telphers travel 700 ft. per min., which enables the telpher to pick up its load, make a round trip to the storeroom, discharge therein, and be on the wharf ready for another trip within five minutes.

24 The storeroom has a maximum capacity for storing sugar of about 370,000 bags or 52,500 tons, each panel between four columns containing approximately 2500 bags when tiered 28 bags high. Sugar arriving by boat is practically all in bags. The bags are unloaded from barges or boats by three small electric derricks on the outer face of the wharf, and from ships by the ships' tackle, in both cases being deposited on the uncovered portion of the wharf. The bags are then placed on trucks in lots of two or three and trucked to a portion of the wharf which is in reach of the telphers, in the case of imported sugar, passing over the Government scales on the way. On this floor the bags are sampled and weighed for settlement with the seller, again made up in slings, and transported by the telpher to the storeroom and there tiered as already described.

25 When sugar is to be used, the same telphers remove the bags from the tiers and deposit them at the shore end of the storeroom, where they are sampled and weighed for controlling the manufacturing process. At this point the sugar is removed from the bags and is carried by bucket elevators to the refinery. The estimated capacity of this telpherage system, as outlined, is 3000 tons per working day of ten hours.

26 An auxiliary storage system consists of two smaller sheds located at the lower side of the main storage shed and served by an electric industrial railway which crosses under the telpher runs, where the loads are received by the cars. The tiering in these sheds is done by portable stackers. Eight spur tracks provide facilities for bringing in or reshipping sugar by rail.

CONVEYORS FOR UNLOADING STEAMBOATS

27 The very low freeboard of the river steamboats and the comparatively great height of the wharves have made it difficult to move cargo from the boats to the wharves by hand. Machines have been developed for this work and a number of such machines have been used. At present machines are in use on the steamboat wharf located just below Canal Street, the commercial center of the city.

This wharf is about 1400 ft. long and has the usual shed. The machines are placed at intervals of about 200 ft. and consist of escalators about 50 ft. long by 3½ ft. wide, made up of a trussed timber frame fitted with a slat conveyor on which dogs for engaging the truck axles are spaced at about 5-ft. centers. The outer end of the conveyor is suspended by slings and counterbalance from a pile-and-timber headframe. The conveyors were originally operated by small steam engines bolted to the framing, with boilers alongside, but these engines have now been replaced by 10-h.p. electric motors.

CONVEYORS FOR UNLOADING COASTWISE STEAMSHIPS

28 For loading and unloading cargo from the side ports of their steamers, the Southern Pacific Company, or Morgan Line, uses hinged conveyor gangways. The most recent installation is at the St. Louis Street wharf and consists of three sets of gangways, each set consisting of one single and two double units about 15 ft. in width by 75 ft. in length. The double units permit of simultaneous handling from side ports and deck.

29 Each unit consists of a decked truss span supported by three steel trusses about 6 ft. deep, properly cross-braced. A width of about 5 ft. at one side of each unit is occupied by a slat conveyor set in flush with the decking and operated through gearing by a 15-h.p. motor located under the gangway at the inner end. At the outer end is a substantial structural-steel frame resting on the edge of the wharf and supporting the sheaves for counterbalance weights and hoisting gear. The gangway is raised and lowered by wire rope operating through double blocks and winding on a drum which is driven through spur and worm gearing by a motor. In one set of these gangways 3-h.p. motors are used for hoisting, but the motion was found to be too slow and in the second and third sets installed later, and of heavier construction, 10-h.p. motors are used.

30 The wharf area served by these machines is about 204,000 sq. ft., with steel sheds over the full length of 1887 ft. assigned to this company. In the year ended June 30, 1914, about 625,000 tons of miscellaneous freight crossed this wharf, or the equivalent of an average of 330 short tons per lineal foot per year.

PORTABLE UNLOADERS AND CONVEYORS

31 As a substitute for the conveyors just described for use on wharves not equipped with machines, there are several installations of portable sectional conveyors. These conveyors are also used to

move cargo from the edge of the wharf to storage in the sheds and for stacking. One such unloader and conveyor, which has a maximum capacity of 1 ton per minute, is shown in Fig. 3. The conveyor is made up of a number of sections supported and connected by platforms, some of which carry the power units. The power is distributed over two, and in the case of the heavier machine, over three electric motors instead of driving the entire equipment by one motor. With the assistance of a battery crane truck, one or two men can connect or disconnect all the sections and place them on the

FIG. 3 PORTABLE UNLOADER AND CONVEYOR

ground ready for moving in a few minutes, and when disconnected the whole equipment can then be drawn from place to place by motor trucks.

32 These conveyors have been in successful use, handling a variety of materials, particularly coffee, rice, sugar, cement, etc., in bags, coils of wire, boxes, and miscellaneous packages; one of the larger machines has handled cotton bales.

COAL-HANDLING EQUIPMENT

33 The coal companies, the transportation interests and some large consumers have established coal-handling plants of some interest.

34 The Alabama and New Orleans Transportation Company has a coal-storage plant at Violet, just below the city, on the Lake Borgne Canal near its junction with the river. The plant consists of three towers movable along tracks parallel to the canal. These towers span railroad tracks and with an extension boom reach barges in the canal and storage piles on shore. In connection with this storage plant, the company operates barges for delivering coal in the harbor and a self-propelled steel bunkering barge of the hopper type, driven by a kerosene-oil engine. The handling mechanism consists of a conveyor in the hull of the barge taking coal from the main hoppers and discharging it into the boot of a bucket elevator which in turn delivers the coal to a conveyor boom discharging on the vessel alongside.

35 The New Orleans Coal Company, which carries on a general bunkering business, has a shore plant for loading its barges which consists of tracks which cross a platform scale and a hopper. Below the mouth of the hopper is an electrically driven belt conveyor discharging into a barge; this barge, together with the bunkering elevator, is taken by a tug to the ship which is to be coaled. The elevator is mounted on a barge and consists of a skip raised in guides by a drum hoist, the skip receiving coal from a hopper fed by a revolving derrick equipped with a clamshell bucket. The skip hoists the coal and discharges into hinged spouts leading to the coaling hatches of the ships.

36 The Monongahela River Coal and Coke Company operates floating coaling plants similar in general design to the elevator described above.

37 The Illinois Central Railroad has installed at Harahan, immediately above the city, a plant of 300 tons hourly capacity for transferring coal from cars to barges. Cars loaded with coal are placed in a double-track gravity yard of about twenty cars capacity. The cars pass over and discharge the coal into a concrete hopper under the tracks. The hopper and tracks are covered by a frame shed. From the hopper the coal is carried by a 36-in. belt conveyor through a gallery for about 150 ft. to a 50-ft. adjustable boom which is suspended from a gallows frame and extends over the barges. The plant is driven by electricity, separate motors operating the belt, the boom hoist, and a winch for shifting the barges. The feed to the belt is regulated by an oscillating-plate feeding device. The coal is distributed across the barge by an adjustable discharge chute on the end of the boom. Two men operate the plant, one handling the cars and one the conveyor.

38 The New Orleans Railway and Light Company has installed on the wharf at the foot of Race Street, opposite their central station, a plant of 200 tons hourly capacity, for unloading coal from barges and delivering it across the shed and roadway to the storage pile. The wharf end consists of a heavy steel tower resting on four spread columns and supporting a mast and boom with bucket and a hopper. The boom swings in a vertical plane only and slopes down and toward the river. Traveling on the boom is a two-yard clamshell bucket delivering coal into the hopper. This hopper discharges into small cars which are then weighed and hauled by cable over an elevated runway and dumped on the storage pile. The inshore end has not been completed as yet.

39 The U. S. Naval Station, on the west bank of the river, includes a coaling plant which is designed to take coal from barges or colliers and store it, delivering it again as desired into ships' bunkers. This plant consists of a steel-and-concrete storage house with elevated bins and steel wharf structure which is also provided with bins and surmounted by a moving tower carrying a clamshell bucket. The two structures are connected across the levee by a covered conveyor gallery.

40 In storing, coal is transferred from the vessel by a 2-ton clamshell bucket to the center hopper on wharf. The coal is then spouted to the main conveyor, which carries it along the conveyor-monitor of the storage house and, by means of adjustable trips, distributes it in the bins.

41 For bunkering a ship, the coal is either spouted from these bins direct into a filler at the rear of the storage house or into small cars running under the bins and delivering to the filler. The filler discharges on the main conveyor, which carries the coal to the wharf and dumps it on a smaller conveyor running lengthwise of the wharf and discharging into the wharf bins or hoppers, from which the ship is bunkered direct by means of chutes. All conveyors are of the suspended-bucket type and are provided with automatic weighing devices. The wharf hoppers are fitted with small auxiliary hoppers near the tops, to which the chutes may be attached and used for bunkering in high-water periods.

42 The storage capacity of this plant is 3000 tons. The capacity of the wharf bins is 750 tons. The hoist motor is 125 h.p.; the conveyor monitor motor 75 h.p.; the main conveyor 40 h.p.; the breaker motor 30 h.p., and the trolley motor 15 h.p.

43 The American Sugar Refining Company is now installing at

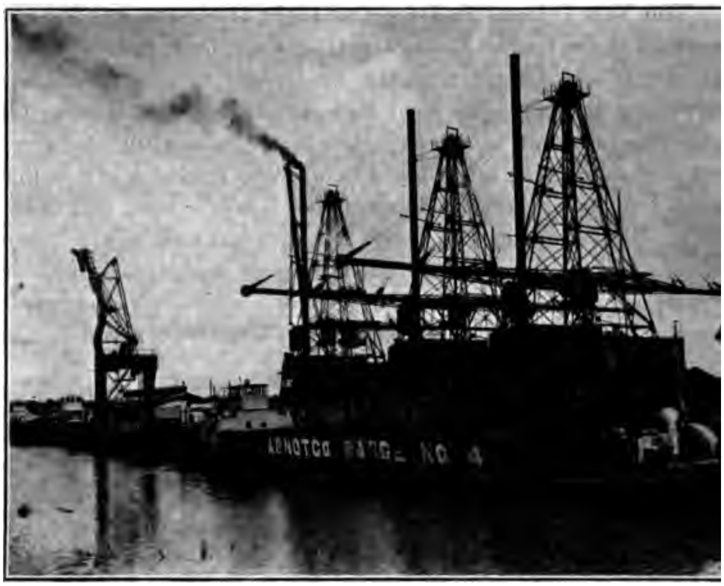


Fig. 4 Typical Coal-storage Plant

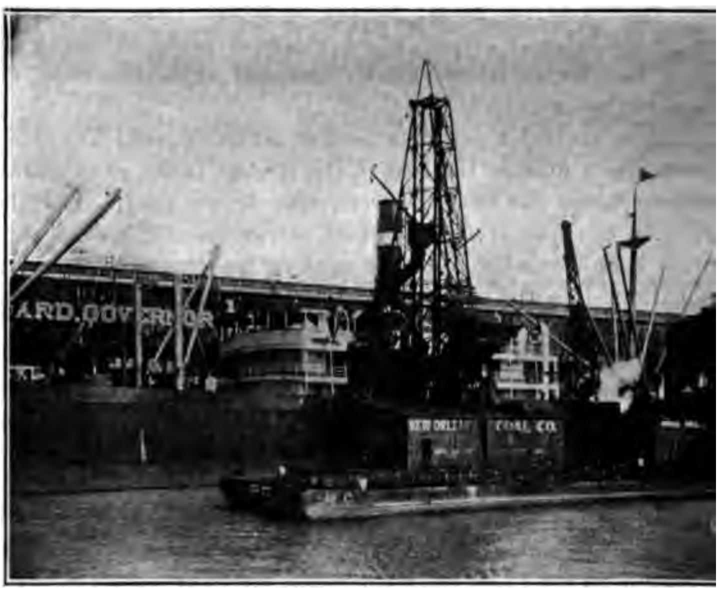


Fig. 5 Typical Floating Coal Elevator

the Chalmette refinery a small coal plant of about 70 tons hourly capacity for receiving coal from barges and delivering it into the track hoppers of the present boiler house.

44 The plant consists of a wharf 36 ft. by 204 ft. paralleling the river, and on the wharf runs an electric locomotive crane fitted with a 2-yd. clamshell bucket and operated by an alternating-current motor of about 100 h.p. This crane lifts the coal from the open barge and deposits it in an elevated dock-storage hopper of 170 tons capacity. This hopper automatically delivers the coal to a conveyor belt by means of beaded flight feeders actuated by a 3-h.p. motor. This conveyor, operated by a 10-h.p. motor, leads over the levee and across the road to a transfer bin of 130 tons capacity. From the transfer bin the coal is dumped into special 50-ton steel cars with hopper bottoms standing on a platform scale. Two cars at opposite points on an endless cable are hauled on a track with automatic turnout between the scale and the boiler-house track hoppers. Fig. 4 shows a typical coal-storage plant, and Fig. 5 a floating coal elevator. The total capacity of the storage plant illustrated is 750 tons per hour; the hoisting engine is 35 h.p. and the trolley engine 20 h.p. The clamshell is 2-ton size.

CAR-FERRY INCLINES

45 On account of the variation in the river level, car inclines with adjustable tracks are required for delivering cars to the lighters or ferries. The West Side roads have four sets of inclines, the most recent installations being for the Texas and Pacific Railroad, near the trans-Mississippi Terminal, and for the Southern Pacific Company at Harahan.

GRAIN ELEVATORS

46 There are in service seven grain elevators of standard frame construction, all owned by railroad companies, except one small elevator privately owned. The large elevators include:

		Bu. capacity storage.
Texas and Pacific:	Elevator A...........	350,000
	Elevator B...........	1,000,000
Illinois Central:	Elevator C.............	350,000
	Elevator D.............	1,000,000
	Elevator E.............	1,500,000
N. O. Terminal Co.:	Chalmette Elevator..	500,000

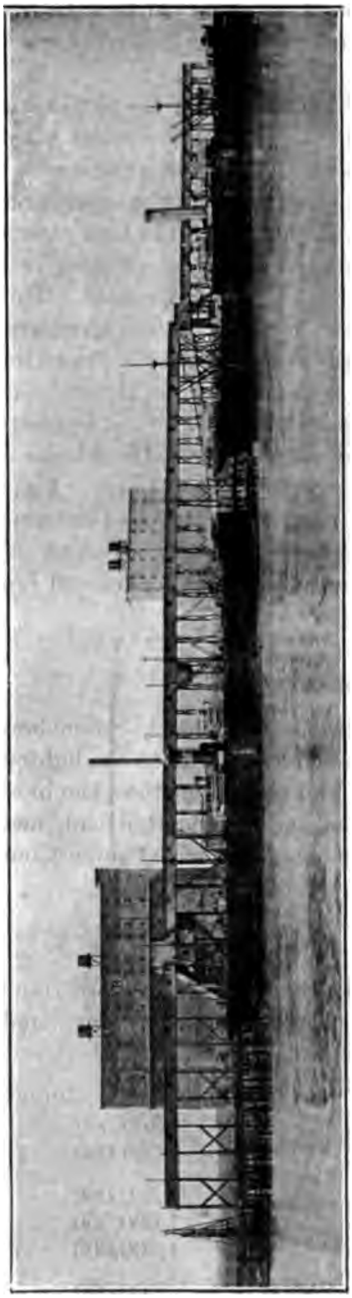


FIG. 6 TEXAS AND PACIFIC RAILROAD CO.'S ELEVATORS A AND B

These elevators, except the Illinois Central Railroad local elevator C, which is assigned to local use, are all of the usual frame type, receiving grain from cars, storing it and delivering it through dock galleries on board vessels at the wharves.

47 The Texas and Pacific Railroad Company's elevators A and B are shown in Fig. 6, and the Illinois Central R. R. elevators D and E in Fig. 7. In addition to these elevators there are at present under construction a reinforced-concrete elevator of 200,000 bu. storage capacity, privately owned, and the Public Grain Elevator.

48 The Public Grain Elevator is being constructed by the Board of Commissioners of the Port of New Orleans, and is of modern type of reinforced concrete. This elevator will have a storage capacity of 1,000,000 bushels and will include some unusual features, being specially designed for rapid handling, conditioning, and blending of grain. The design of this elevator is shown in Fig. 8. It will be noted that the design provides unusual facility in the receiving of grain, secured by the three distributing belts and their equipment of Mayo spouts over tank storage. The additional height required for this special equipment has been taken advantage of in providing also flexibility in distribution spouting in the workhouse, which materially increases the speed of operation. The height of the basement permits an unusual amount of the tank storage reaching each of the four shipping belts, greatly facilitating the blending or mixing of grain. In addition, the workhouse includes the necessary equipment of cross-conveyor belts, etc.

49 The drying plant is a distinct feature: Besides having 2000 bu. drying capacity per hour, the building provides deposit space above the drier for a full day's run, which can be filled from three different parts of the storage in less than half an hour. The building also provides deposit for a half-day's run below the drier. Special attention has been given to the rapid loading of vessels, the plant as designed having three 40-in. belt shipping conveyors, with a total capacity of 100,000 bu. per hour, all of which can be used to load one to four vessels at the same time. On account of the large amount of grain accessible to the Mississippi River, the plant is being equipped with a marine tower and machinery to unload ships or barges. To more readily accommodate the great variation in water level and the varying dimensions of the vessels, a pneumatic unloader has been adopted.

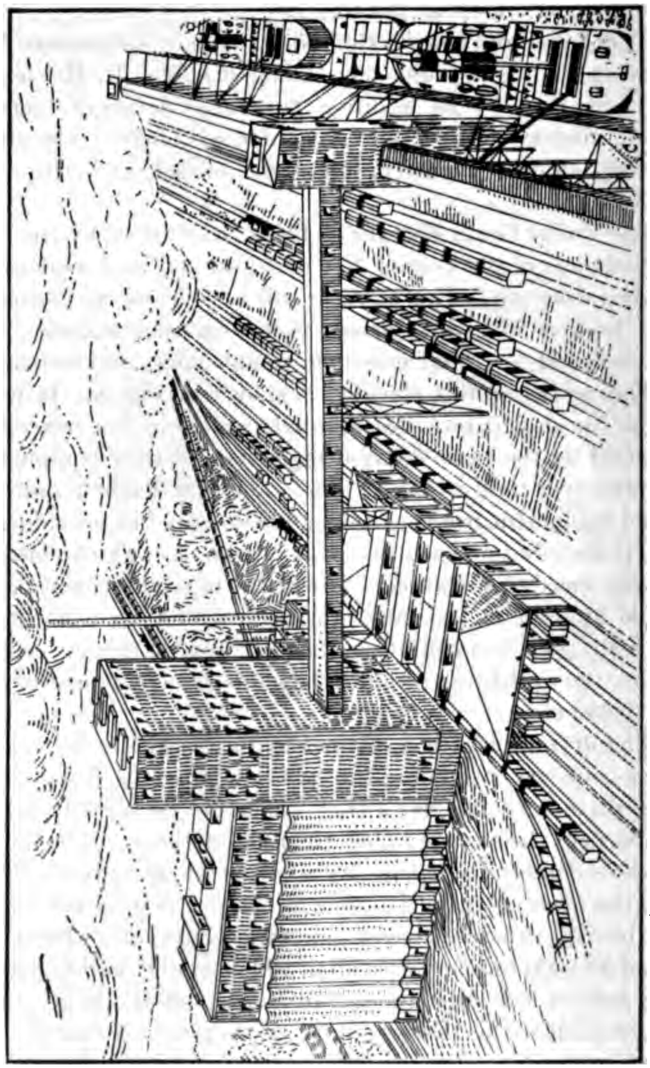


FIG. 8 DESIGN OF THE PUBLIC GRAIN ELEVATOR

50 In addition to these special features, the plant is provided with safety devices of latest type, including a complete dust-collection system, signal systems, strand indicators, journal alarms, etc. On account of the fireproof construction and protection apparatus, it is expected that the minimum insurance rate will apply.

PUBLIC COTTON WAREHOUSES

51 The Cotton Warehouses and Terminal plant of the Board of Port Commissioners is of particular interest because of the cotton-handling machinery with which it is equipped, and because of the consequent unique design of the buildings. The plant has an annual warehousing capacity of 2,000,000 bales and is composed of four large warehouses, a compress building and a two-story wharf house, all of the latest type of reinforced-concrete construction. All of the buildings are connected by two-story runways or bridges upon which cotton is transported from one part of the plant to another.

52 Cotton is handled in the warehouses by high-speed electric traveling cranes which are operated by one man who rides in a cab attached to the trolley. One of these cranes is shown in Fig. 9. The span is 28 ft. 6 in. The hoisting motor is 20 h.p. and speed of hoisting 150 ft. per min. The trolley speed is 150 ft. per min. The travel motor is 10 h.p. and travel speed 500 ft. per min. The capacity is 2 tons. By the use of an automatic grapple this same operator can pick up cotton from the platforms or from trucks and place it at any desired point in the warehouse, or can load it on a truck or trailer for transportation to the wharf. Fig. 10 shows one of these grapples above the bale, and Fig. 11 the grapple holding the bale. Ordinary hooks are also used to some extent instead of the grapples, but require an additional man to place them on the bales. Where it is desired to remove a bale of cotton from the pile, a specially designed bale puller, Figs. 12 and 13, is used in connection with the crane. This requires the help of only one man in addition to the crane operator, but will remove any desired bale from a tier any number of bales high in less than three minutes. The cost is 25 cents for each bale actually pulled.

53 The cranes can be transferred from one warehouse compartment to another on transfer tables, Fig. 14, thus avoiding the necessity of supplying a separate crane for each. The traverse motor is 6 h.p., and the traverse speed 100 ft. per min. The cranes are of the three-motor type and are rated at 2 tons capacity, operating on 220-volt direct current. The hoisting motor is rated at 20 h.p. and

Fig. 9 Traveling Electric Crane in Public Cotton Warehouses

FIG. 10 AUTOMATIC GRAPPLE ABOVE COTTON BALE

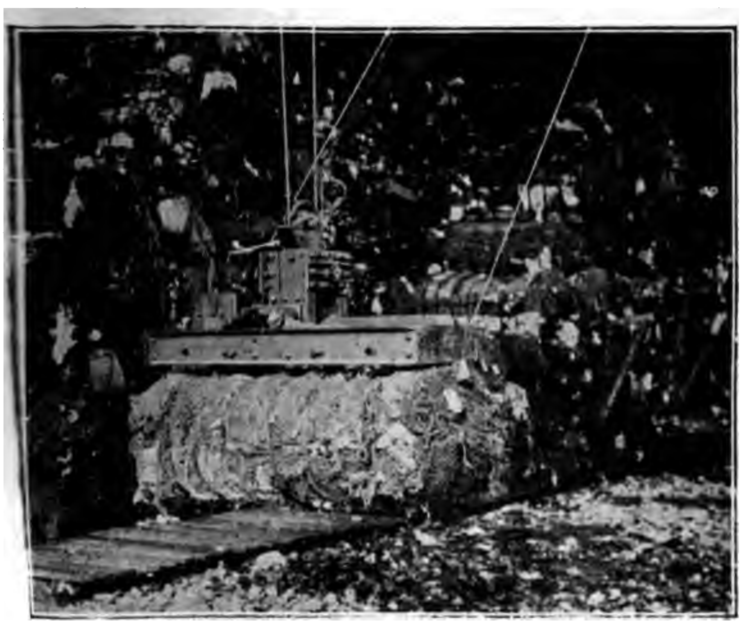


FIG. 11 AUTOMATIC GRAPPLE HOLDING BALE

FIG. 12 COTTON-BALE PULLER IN POSITION

FIG. 13 COTTON BALE REMOVED BY PULLER

Fig. 14 Transfer for Cotton-Warehouse Cranes

raises the load at the rate of 150 ft. per min. The bridge is moved at a speed of 500 ft. per min. by a 10-h.p. motor and the trolley 150 ft. per min. by a 2-h.p. motor.

54 A compress building contains three steam presses of the usual Morse type, one having a 90-in. cylinder, one an 86-in., and one an 84-in., and a hydraulic press of 450 tons capacity has been contracted for. Hand trucks are used to some extent in the immediate vicinity of the compresses, but, for the most part, the cotton is handled from place to place in the plant on small self-propelled storage-battery trucks, Fig. 15. The speed of these trucks is 6 to 8 miles per hour, and the capacity 2 tons. The batteries have 24 cells, U. S. Lighting, 9 plates per cell. The motors are 2 to 3 h.p.

55 It is also proposed to employ electric tractors with trailer trains for the transportation of cotton from one warehouse to another and from the warehouses to the wharf. Each tractor will draw three trailers holding ten bales of cotton each. These trains are to be operated on the overhead runways and will travel at a speed of 6 to 8 miles per hour. A number of gasoline motor trucks with special bodies to carry 17 bales of cotton have been provided for the transportation of cotton about the city.

56 The wharf is to be provided with single-leg gantry cranes for loading and unloading the ships. These are to be 2-ton cranes of the three-motor type with a luffing or derricking boom giving a maximum radius of 50 ft. The cranes will serve the apron wharf with its two railroad tracks, as well as a second-story platform of the wharf house and will take cotton from any of these places and lower it directly into the holds of the ships. They can, of course, also be used for unloading cotton from vessels and placing it on the wharf or platform, or in cars. It is possible that a continuous belt conveyor similar to the banana conveyors will be developed for the purpose of loading the vessels.

57 Table 1 shows the labor costs at the Public Cotton Warehouses.

LOCOMOTIVE CRANES

58 Several locomotive-crane installations are of interest. In one plant bricks are placed in crates at the kilns, and the crates loaded on barges. On arrival the crates are removed from the barge by crane and loaded on teams or first placed in storage; the bricks remaining in the original crates from the kiln to the consumer. Other

locomotive cranes, both steam- and electric-driven, are used for handling sand, gravel, and shells from barges to and from storage and to teams or cars.

TABLE 1 LABOR COSTS AT COTTON WAREHOUSES

Operation		Cost in Cents Per Bale	
		Minimum	Average
Unloading		1.3	2
Inspecting and Sampling		2.3	3
Weighing		2.0	4
Compressing		6.0	8
Band Making		1.0	1
Trucking	Hand Trucks	....	2
	Electric Trucks	....	1
Storing	By Hand	2.0	6
	With Crane	1.5	2.25
Repiling	By Hand	....	4.00
	With Crane	....	1.75
Turning Out	By Hand	2.0	7
	With Crane	2.0	7
Pulling Machines	Hand	25.0	50
	Crane	10.0	25
Ranging		....	1.5
Marking		....	0.5
Spotting (Part Output)		....	1.5
Shipping	By Trucks	....	2.5
	On Cars	....	3.0

Average Total Labor in Plant	26 cents	(Laborers paid from 20 cents to 25 cents per hour)
Compressing	8 "	
Total	34 "	

FLOATING DERRICKS

59 Privately owned by individuals, the steamship companies and contractors, are many floating steam derricks (Fig. 16) of various capacities, there being one each of 100, 50, 40 and 35 tons, and a considerable number of smaller machines.

SAND AND GRAVEL HANDLING

60 Special elevators are used for unloading sand, gravel, and shells from barges and delivering to storage piles. The unloader shown in Fig. 20 consists of a barge carrying a revolving crane with clamshell bucket which delivers the material to a hopper on the barge. From the hopper the material is spouted to a belt conveyor on an adjustable boom which delivers it to stock piles on shore.

Fig. 15 Storage-Battery Cotton Trucks and Trailers

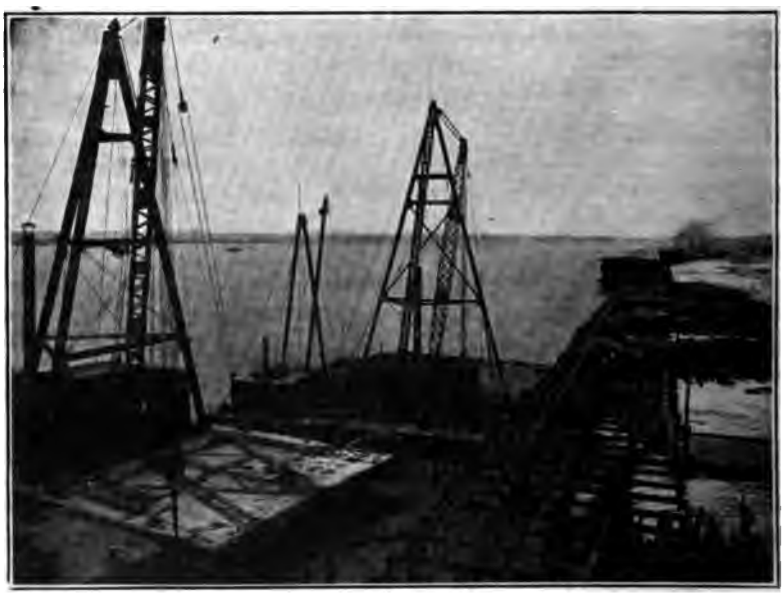


Fig. 16 Floating Derricks

FIG. 17 STORAGE-BATTERY CRANE TRUCK

FIG. 18 PORTABLE CRANE HANDLING MAHOGANY LOGS

FIG. 19 TIMBER JACK

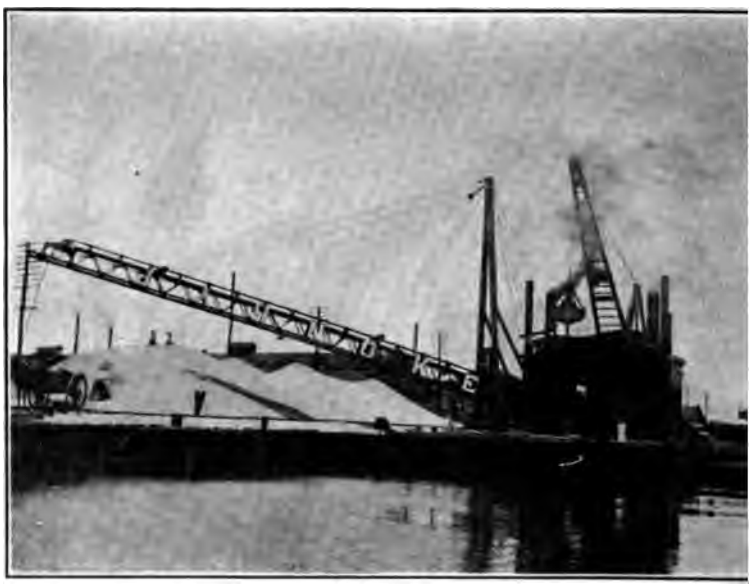


FIG. 20 FLOATING ELEVATOR UNLOADING AND PILING SHELLS

The clamshell is of 1½ yd. capacity and the unloader handles 150 yd. per hour.

MISCELLANEOUS EQUIPMENT

61 In addition to the larger equipment already mentioned, many smaller devices are in general use. Included in this class are the electric and hand-operated portable cranes, electric trucks frequently with trailers, gasoline trucks, timber jacks, tiering machines, etc. Fig. 17 shows a small storage-battery crane truck. Fig. 18 illustrates a portable crane handling mahogany logs. Fig. 19 shows a timber jack.

DISCUSSION

Willard C. Brinton. Water-front terminals up to this time have been designed chiefly by civil engineers who give first consideration to the pier construction as related to docking ships. Comparatively little thought has been given to the needs of the terminal in so far as the actual movement of freight is concerned. The civil engineers have apparently not realized that immense differences in cost of operation can result from care given in the preliminary planning and construction of a steamship terminal.

Even when careful study is made of the operating needs of any steamship terminal, it frequently occurs that the problems met are such as to render the engineer almost powerless. In most cities certain business habits have gradually grown up which result in one organization doing only a small portion of the work required in handling a given lot of steamship freight. This specialization under separate organizations is so well established that it is difficult to change. Thus, though the engineer is able to plan and furnish facilities to permit the rapid and economical handling of freight, unless the business habits and customs of the people in the district are considered, there is danger that the facilities may not be used.

It is not unusual to find four or five distinct organizations successively handling a cargo at a steamship terminal, so that after one organization lays down the freight another organization picks it up again. If all the work were done under one management with proper supervision to secure good coördination, there would be not only a vast saving in the direct cost but an even greater saving in the indirect gains due to the increased tonnages which it would be possible to handle with a given investment in terminal facilities.

Overhead telpher systems are often advantageous for sugar refineries or fertilizer works, where whole cargoes are handled in one large lot or consist of one uniform type of package. Telphers cannot, however, be readily applied to existing pier sheds, and even where a completely new water-front terminal plant is built, it is not usually wise to install telphers for handling miscellaneous cargoes at their present stage of development.

For general service at steamship terminals, handling miscellaneous freight, the best arrangement at the present time seems to be a combination of the ships' hoisting machinery with either tractors and trailers or industrial trucks carrying loads on the back of the truck. Where there is a fairly long haul with a large turning radius possible at either end, tractors and trailers are frequently desirable. For crowded pier conditions there is, however, a disadvantage with tractors and trailers due to the fact that the trailers are difficult to maneuver properly at the side of the ship without hand labor. Then, too, tractors and trailers are not feasible in crowded warehouse aisles where single industrial trucks can be used economically. Also, it does not seem feasible to weigh individual trailer loads over a platform scale without disconnecting the couplings between separate trailers to avoid all chance of error in the weights. Industrial trucks can be run directly over platform scales and the weight obtained without any extra lifting of the cargo. The difficulty of weighing freight thus assumes such importance as to justify very careful consideration of industrial trucks as compared with tractors and trailers.

Another gain with industrial trucks is possible because they can be used on fairly steep grades to go from the second story of a pier to the third or fourth story of a warehouse building over an inclined bridge across a marginal street. With electric trucks of sufficient battery and motor capacity, it seems feasible to construct storage warehouses so that material may be taken directly from the side of the ship over platform scales and thence on ramps to the proper floor and to the exact position inside the warehouse without rehandling between the side of the ship and piling in the warehouse. Sorting at the side of the ship into lots of different marks is feasible with industrial trucks especially designed for the work and using separate plank platforms.

One of the chief weaknesses of the ordinary industrial truck as compared with tractors and trailers is due to its small platform space, resulting from the truck designer's wish to have a small turning radius. It is, however, possible to obtain a small turning radius combined with a sufficiently large loading space to carry all the tonnage desired. Ordinary steamship freight is so bulky that the question in truck designing is not one of handling tonnage, but one of carrying sufficient bulk without having the turning radius so large as to make operation difficult in crowded places.

Another objection to present-day industrial trucks compared with tractors and trailers for steamship-terminal work is the necessity for delaying the industrial truck while it is being loaded and unloaded. With tractors and trailers, however, the trailers can be disconnected and the tractors sent on to other work. But this advantage is more apparent than real, for the reason that when large quantities of freight are transferred between two definite points it is generally necessary to use the tractors to move the preceding trailers before the next set of trailers can be placed in that same position for loading or unloading.

What is needed at steamship terminals is a truck capable of carrying a load from any floor of any building to any floor of any other building over the whole area of the terminal, and yet capable of being driven through narrow aisles, turned in a small space and backed up out of blind alleys. This truck should have large-size detachable platforms which can be raised and lowered by power in a very small amount of time to save delay in loading and unloading and to save the unnecessary handling of freight; and it must have sufficient power to be a good tractor for use with trailers whenever trailers can be used economically. With this combination, the advantages of both tractor and trailers and industrial trucks can be obtained without the commonly found disadvantages of either.

If a steamship terminal is designed to utilize the full possibilities of the type of truck suggested, the savings due to reductions in direct labor costs and the increase in tonnage which the terminal can handle are great enough to be most attractive from a financial standpoint.

FRANK B. GILBRETH. In this matter of devices for handling materials at docks, I want to call attention to the use of gravitation for this purpose.

The efficiency of gravity as a transporter and for handling materials in different parts of a building by simply hoisting them and having them descend by gravitation on roller conveyors, has been demonstrated time and again. The cost of this method is low.

In the public cotton warehouse described horizontal transportatic is being done by trucks. This is inefficient and is wasteful both c time and money. The vertical transportation of material at dock costs very little comparatively, for the reason that it can easily b systematized, standardized, and controlled. Handling materials a this public cotton warehouse by hoisting vertically by power an distributing horizontally by gravitation would reduce the cost t a point that would be surprising.

A further reduction of costs could be secured by standardisin the handling devices, but the first step is an appreciation of tl unusual force, of the availability, and of the efficiency of grav tation as a transporting force, and of the advantageous by-product as well as products, of its utilization.

JOHN R. FORDYCE discussed certain features of the cottoı handling warehouse, as follows:

Cotton Storage and Piling. The compartments are designed s that four rows of bales are arranged with two open aisles, to enabl access to the ends of each bale, so that any bale can be pulled out c the piles, and also that water from the sprinkler above can reacl the origin of any fire. The compartments are, I believe, approxi mately 35 ft. in height from the floor, and the compartments near th wharf are to be 40 ft. The two piles of bales that lean against th wall can doubtless be carried up to twelve bales high, or about 2 ft., without fear of toppling over, but the two piles in the cente which lean on each other cannot safely be carried up more than eigh bales high, or about 14 ft. (I assume the thickness of a compresse bale as 20 in.) Even at this height it will be hard to get a bale ou of the lower tiers of the side or wall piles because of the great weigh above and the rough and uneven character of the bales. The pul of the ropes of the bale puller will throw the pile over against th wall and the pile will of course not fall down, but if this same oper ation were tried on the middle piles there would be danger of th whole pile being thrown over. In case many bales had to be take out, the piles would no doubt have to be repiled, as the holes, unles filled, would cause the collapse of the pile. A system of jackscrew working from the floor could raise the weight of the bales above an enable the lower bales to be pulled out just as easily and cheapl as the plan described.

The overhead electric crane, while ideal for machine shops an other places where the packages have to be carefully handled an placed, does not seem to me to be the best thing for cotton handling. However, I realize that if this warehouse system is likely to be used for other commodities, then the cranes and the high warehouses will probably be ideal. For cotton, however, you have wasted at least one-third of the height of your buildings, expense of preparing your craneways, power plant, and crane-transfer structure, and maintenance of the cranes. The cost of these is out of proportion to the benefits gained, in handling a slow moving, non-damageable package such as a bale of cotton.

Assuming that 20 ft. is the practical height of a pile of cotton. A building 25 ft. high would give ample room and clearance for the automatic sprinkler, and an elevator working from the floor of the aisle would hoist bales equally as well as overhead cranes, and at the same time could be arranged to lift off the weight of the bales above the one which was wanted. Thus the additional cost of building the runways and all the attending apparatus would be saved and a cheap elevator used which could be dragged in and out of a compartment much quicker than a crane could be. Another and far more serious objection to the overhead electric crane is the danger from fire. I tried an overhead monorail hoist over cotton in 1901 in Little Rock, Ark., and had to abandon its use because it set fire to the bales below. I understand that several fires have already occurred in these New Orleans warehouses, and have before me a pamphlet issued by the Louisiana Fire Preventive Bureau which attributes the cause to defective mechanism in the cranes, which no one of course foresaw. Even if these are corrected others may develop, and I believe the only safe way to do it is not to try to use electricity in overhead apparatus. Overhead apparatus operated by cables similar to aerial trams can be applied and the winding engines kept on the outside of the compartments. I have devised one for the warehouse in Galveston, where the sheave runs on a rope so that the entire strain is taken off the roof, which is too light for any increased weight. This rope can be shifted from side to side in the bays and thus come over any pile of cotton. The bale of cotton can be dragged back and up over a pile of bales when no aisles are left, which sometimes occurs in congested times. This whole apparatus is cheap and can be carried out of one compartment and erected in another in a short time and at little cost. The necessary rings and hooks which have to remain in a compartment holding one thousand bales cost less than $50, so that the amount of money locked up when the compartment is full or idle and not being worked is a minimum.

Transportation of Bales. Mr. Brinton stated that the carrying of bales by trailer pulled by electric-storage-battery tractors is not as practical or economical as if the load was carried by the truck itself. It has been said that a slight grade should be given the roadways from the warehouse to the wharfs, as the loads are always to be in the same direction. Here again the designing engineers no doubt had to provide for goods from ships moving in as well as cotton from warehouses moving out. I have had quite a lot of experience in handling cotton by trolley systems, and have found that a slight fall of 6 in. per 100 ft. in the direction of the movement of the cotton is a great help. In Memphis, Tenn., in the plant of the Memphis Terminal Corporation, there is a trolley track with that fall on which one mule can push one hundred bales of cotton in one string of trolleys. His work in one day at that rate would be the delivering of 4000 bales in ten hours over an average distance of half a mile. A system similar to that could be applied very easily in the New Orleans plant, for you already have all the headroom needed. The bales hoisted out of the compartments on to the upper runways by the overhead cranes could be dropped on a roller platform which could be easily moved along opposite the opening of each compartment. The trolley track running along the outside of the posts could have the required grade and the bales be hung on to the trolleys as they were rolled past this roller platform. They could then be gathered into strings and pushed over to the wharves by tractors. The same tractor which could pull four trailers loaded with six bales each or twenty-four bales in all, could push one hundred bales if they were hung from trolleys. The empty trolleys could be returned to the warehouses along the outer runways and hoisted to the upper levels by inclined moving chain elevators such as are used in packing-house practice. This system could be easily designed to carry other commodities than cotton and to work both in and out with loads.

Gardner T. Voorhees. One of the biggest things that I have seen at this meeting was the warehouse receipt of the public cotton warehouse, and I think this has an immense value as a negotiable matter. I am confident that a similar receipt on the very valuable goods that have to go into cold storage would have an equally great value, although, of course, the goods that go into cold storage, more particularly in the North, the butter and eggs, form the big things on which money is loaned by the banks. But in the future a warehouse receipt having the city and the state both back of it, the same as the cotton warehouse receipt inaugurated in this port, would, I believe, be of great value to the users and producers of any material passing through cold storage.

George H. Davis emphasized the point that the handling machinery of a port is not an engineering problem entirely. It is possibly 25 per cent engineering, possibly 25 per cent business, 25 per cent policy in running — as to whether this will be acceptable or that will be acceptable — and 25 per cent miscellaneous. While the machines themselves are simple, and such that a casual observer would think ought to be used everywhere, as a matter of fact, it is rather difficult to establish mechanical handling through a port of this kind. The matter is one of slow development, and the slowness does not come from the engineering side, but from other sources.

No. 1536

ESTABLISHING A STANDARD OF MEASUREMENT FOR NATURAL GAS IN LARGE QUANTITIES

By Francis P. Fisher, Bartlesville, Okla.
Member of the Society

NEED FOR A STANDARD

The original business of natural gas handling and selling was confined to the discoverer, or owner, taking the gas from his own wells through his own pipes to various purchasers, and supplying them, in consideration of certain fixed monthly payments, with all the gas they could use. This eliminated the whole problem of gas measurement.

2 Some twenty years ago this simple and effective solution became impossible because of its extravagance, and the familiar domestic meter of the artificial gas business was slightly modified to take care of the domestic and small industrial consumer of natural gas. The operation of these meters is based on actual displacement from alternate filling and emptying chambers, and is comparatively simple. Where properly cared for and installed, these meters have given continuous measurements well within the commercial limits of required accuracy over long periods of time.

3 The discovery of new gas fields has been practically a continuous process, and many fields were opened up in the vicinity of towns previously piped for artificial gas. In time, selling contracts were made between the owners of gas wells and the artificial gas companies already equipped to serve domestic consumers, the sale of gas being effected at the town border. Also, large industrial consumers were connected up to gas lines and took gas in quantities too large to measure by the standard type of meter, leading to suggestions of many devices for taking care of the measurement of these larger quantities.

Presented at the Spring Meeting, New Orleans, La., April, 1916, of The American Society of Mechanical Engineers.

4 Today, approximately two and a half million people in this country make or receive a monthly settlement based on the measurement of natural gas. The gas must be handled from its original location in the gas-bearing rock strata to its point of consumption without being seen or weighed. The basis on which the gas changes ownership at various periods during its handling consists of indications on dials and charts of various apparatus; and in numerous cases where the faith of one or both parties in these indications is insufficient, the quantity of gas which has changed owners must be settled by arbitration and compromise.

AVAILABLE MEANS OF MEASUREMENT

5 While very accurate, the large wet meters used for station meters at the producing plants of artificial gas companies, by reason of size, cost and care required for operation, were not adapted to the much larger quantities of lower priced gas handled in the new industry. The measuring devices suggested to meet this problem, up to within the last three or four years, were of three general classes: *a* proportional meters; *b* pitot tips in ordinary pipe lines; and *c* calibrated tubes containing pitot tips.

6 The proportional meter is simple and ingenious in principle. The gas passing through this meter is divided automatically into a large and a small channel, each having a definite ratio of resistance to the gas flow, so that the quantities of gas passing through the two channels have a fixed ratio, varying from $\frac{1}{50}$ to $\frac{1}{200}$ according to the size of the meter. The smaller quantity is measured by an ordinary domestic meter with a multiplying dial register.

7 The proportional meter has been developed with great ingenuity by different manufacturers, considering the difficulties of the mechanical problem of accurately dividing the gas with such a very large proportion of unmeasured gas, so that any inaccuracy, either in the division of the gas, or recording of it, would be multiplied from 50 to 200 times in the resultant indication. It is remarkable that such good results were obtained with it.

8 If the gas to be measured had been a clean, pure, perfect gas, it is probable that proportional meter records would have been satisfactory. However, to the initial mechanical difficulties of devising such a meter were added difficulties caused by dust and sand from the rock formation, water, salt, oil and gasoline from the wells, dirt left in the pipes when the line was laid and small bits of debris from the same source, including occasionally portions of carcasses of

animals that ventured into the lines during construction, and in winter time, quantities of hoar frost forming on the inside of the pipe line, and moving along with the gas whenever the temperature rose sufficiently to loosen the frost from the pipe. Such actual conditions call for very frequent expert attention to keep these meters in a theoretical condition of correct registration. The number of competent men in this country today, trained to adjust and calibrate these meters, is wholly inadequate to secure a continuous and satisfactory record of the gas to be measured. The best condition for proportional meter operation is in the distribution plant, a long distance from the fields, where the gas has passed many drips and cleaning devices and is comparatively clean. The most unpropitious location is in the fields close to the wells.

9 The pitot tube furnishes a more certain and dependable method of measurement of very large quantities, and has been widely used in various forms. There are two general types of installation: The first consists of inserting pressure and velocity tips in the ordinary pipe line through a tap made for the purpose, assuming an average velocity at a given distance from the circumference (taken at various values by different writers). The quantity of gas passing is measured by observing the difference of pressure between static and velocity tips, and calculating the quantity passed by the fundamental pitot tube formula $V = \sqrt{2\,gh}$. The fundamental weakness of the formula is that a different and unknown coefficient should be applied to each individual installation to take care of the slight variation from nominal diameter and the effect of roughness of the pipe. The coefficient is also modified by irregularity of velocity in the same zone, shown by the erratic results obtained in different pipe traverses where the velocity tip (or both tips) is stationed progressively at regular intervals across the diameter of the pipe, and observations plotted to show the variations in velocity at different distances from the center. Fig. 1 shows several such curves for an 8-in. pipe. The results of many such traverses point to the inevitable conclusion that the conditions of flow in the line are sensitive to the slightest disturbing influences, such as the insertion of even a very small tip in the line. The position of average velocity cannot absolutely be predicted.

10 The second and more refined form of installation of the same instrument, and one which for many years was considered the standard for large quantity measurements, consists of tubes of polished brass carefully compared with similar standardized tubes

and the coefficients determined by experiment. This instrume been fully described by Thos. R. Weymouth[1] and B. C. Oliph

11 These tubes at their best are capable of giving ver results. They are calibrated by experiment individually, an one, therefore, is liable to have a coefficient varying slightl the personal equation of the man making the test. In actual p the tubes are occasionally found not to run harmoniously in t on the same quantity of gas. Only recently have satisfacto ferential gages been devised for recording automatically the ob differential pressures used in the formula for computing the tities. It was, therefore, found necessary to have these

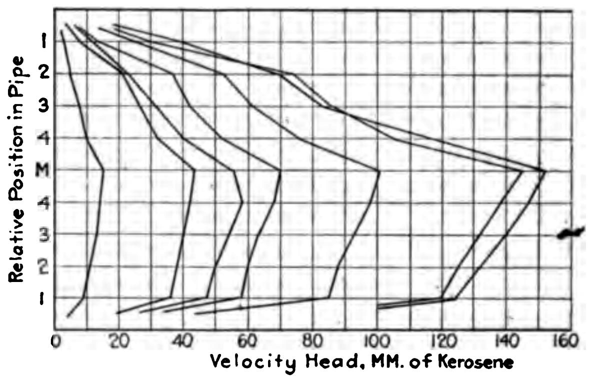


FIG. 1 PITOT TUBE TRAVERSE, 8-IN. PIPE

installed at regular 24-hr. operating points and the gage re recorded generally at 15-min. intervals by observers, thus br another personal equation into the result. The long piping u installed from these tubes to the observing station would a material effect on the indication from small leakages. In cases also it was found that sand or salt lodged against the ve opening, partially or wholly closing the exposed tip, again af materially the indication without the knowledge of the ob In one case a small tin can hung itself neatly over the ve tip and put an important measuring station temporarily commission.

[1] Measurement of Natural Gas, Trans. Am.Soc.M.E., vol. 34, p. 109

[2] Proc. Nat.Gas.Assn. of America, vol. 4, p. 212.

MORE RECENTLY SUGGESTED MEANS OF MEASUREMENT

12 In the last few years, additional standards of measurement which promise a distinct advance have been proposed. The Thomas electric meter, fully described by J. C. Wilson,[1] operates on the most ingenious principle of imparting a small quantity of heat to the gas electrically, and maintaining a constant temperature increase between the incoming and outgoing gases. The quantity of gas is directly calculated from the quantity of heat supplied, as measured by the electrical input necessary to furnish the heat. This meter has the great advantage of being theoretically accurate without a specially calibrated coefficient, and also of working independent of the pressure at which the gas passes, giving in effect a measurement of mass. The question arises, however, as to the adequacy of available data on the specific heat of gas of widely varying composition and with daily changes in specific gravity. The meter would seem to call for further affirmative work on this point. Its installation involves considerable expense per unit as compared with other types.

13 The so-called orifice meter is another comparatively recent suggestion. Orifices calibrated by individual experiment from the standardized pitot tubes previously referred to, and the differential across the orifice recorded by automatic recording gages, have now been on the market several years. This well-known principle of flow of fluid through a thin-edged orifice is very simple of application to pipe lines, and seems to give promise of good results; but there has been great difficulty in securing recording gages delicate enough to retain their accuracy and rugged enough to keep in continuous operation under severe field conditions, — occasionally withstanding excessive pressures, either forward or backward. The three or four types of gages available were subject to sudden derangement and showed friction lag, seeming to require attention of a very similar order to that given proportional meters.

14 The practicable range of capacity for a given orifice, with the differential varying from minimum to maximum limits of the recording gages, varies from a ratio of 3½ to an extreme ratio of 5, and unless the installation is very accurately adapted to the required quantities, the differential reading is liable to be either too low to be significant, or beyond the limits of the gage part of the time, resulting, in spring gages, in the derangement or destruction of the pressure spring.

[1] Proc. Nat.Gas.Assn. of America, vol. 6, p. 219.

15 The venturi meter has also been applied to some large and important measurements. The comments above apply practically to the venturi meter with the exception that the range is somewhat wider, the cost of installation is considerably heavier, and a true formula is more complex.

16 The rotary station meter, quite generally adopted in artificial gas plants, has also been applied recently to a comparatively few installations.

17 Very delicate and complex apparatus that has proved successful for steam meters in expert hands is wholly impractical under the rough conditions of natural gas measurement.

18 With the meter difficulties enumerated, for many years most of the gas companies did not attempt a complete measurement of the gas produced from their own wells until it reached the point of consumption and sale. Even when measurement was attempted in the field, the conditions were so severe that the meters not involving any change of ownership of the gas were apt to be neglected, and attention concentrated on the field meters where gas was bought from separate producers and well owners. Thus, comparatively little has been known of the actual loss of natural gas from field to consumer, and apparent losses have been charged back as part of the unmeasured production, or lack of production, in the field. In other words, while the extravagance of the ultimate consumer has been soundly and properly eliminated by selling him gas in measured quantity, the loss of gas between the producing wells and the consumer remains hidden, owing to difficulties of measurement.

19 Due to various causes, however, in many fields the tendency in recent years has been towards the separation of the ownership of the wells from that of the great pipe line systems connecting the producing fields with ultimate markets. This has raised an imperative call for field measurement, involving financial settlement, for large quantities of gas taken at the wells. Of course, this change has been gradual and many small lines already reached that point several years ago, but in general the great pipe line systems are only beginning to feel the effects of this tendency in the last few years. The result is that the comparison of the amount of gas purchased and sold has been forcibly emphasized by its financial aspect, with very surprising results in many cases, showing enormous unaccounted-for losses of gas.

20 To avoid the difficulty of large quantity measurement, many contracts for sales have been made with separate distribution com-

panies having pipe line systems, where the supplying company takes a percentage of the domestic meter sales. Thus, the losses in towns themselves were not generally realized. Instances have developed where from 35 per cent to an extreme of 80 per cent of all gas entering town distributing systems appears as unaccounted for in meter sales, and some large pipe line systems have discovered overall losses approximating 50 per cent of all gas entering the system. In fact, it was discovered recently that losses in excess of 25 per cent are the general rule rather than the exception.

A SPECIFIC PROBLEM OF LOSS

21 Some three years ago the writer became associated with the reorganized management of a natural-gas piping system in the Mid-Continent field, comprising something over 1200 miles of main lines with corresponding field and distributing plants. In the course of investigations preparatory to outlining a general policy, there was discovered between purchase and sales a loss of gas which seemed wholly unreasonable and the elimination of which would be the shortest and most effective road to early increase of earnings.

22 The total loss could be made up from losses from six possible causes:

1 Leakage of field lines
2 Leakage of main lines
3 Leakage in distribution systems
4 Errors in large capacity meters
5 Errors in domestic meters
6 Errors in estimates of unmeasured gas.

The first step in eliminating this waste in various parts of the system was to provide an accurate gas measurement at strategical points, for segregating the leakage so that the repair work could be made efficient. Manifestly, the first problem was that of large capacity meter installation, and with part of our losses possibly due to meter inaccuracy, the first step was the establishment of a definite standard of accuracy in meters. The work done to establish this standard has been continued by a technical staff for approximately two and one-half years at considerable expense. It has now passed the second phase of installing measuring stations to segregate and locate definitely the remaining leakage losses.

23 Meanwhile, repairing of the most evident weak places and reduction of 95 per cent of our purchase and 75 per cent of our larger

sales to standard bases, have reduced the unaccounted-for losses about one-half, and we are prepared to get after the more difficult remainder with more efficient tools and better promise of quick results. The matter of critical study of domestic meters, believed to be the final step in this campaign, is reserved until other corrections and improvements are initiated and in operation.

POINT OF ATTACK

24 In selecting a point of attack, the large capacity measurements were chosen as promising quicker results for a given amount of work. The small capacity meters as now operated are more reliable, but the number is so great that improvement through checking these meters would necessarily be slow.

25 In connection with large capacity measurement, a catalogue of the weaknesses and objections to all known and available methods has been given earlier in the paper. It was determined that the relative objections be weighed and one of the systems chosen as a standard to be developed to a closer degree of accuracy than heretofore available. The method chosen for development was the orifice meter, in the belief that the objections to the existing types could be overcome at less cost and more simply than in any of the others. Further consideration was the absence of delicate mechanism, and the simplicity of the formula. There is no question but what any one of the other methods of measurement is capable of an equal degree of refinement, but the present paper is chiefly concerned with the work actually done on this one type of meter.

26 In order to give the work a sound foundation, it was considered necessary to establish a standard of reference entirely independent of existing meter indications, and a 250,000-cu. ft. capacity artificial gas holder at Joplin, Mo., was taken as a reference quantity. This part of the work has been described concisely and accurately in a paper presented to the Society in December, 1915, by E. O. Hickstein. Some 500 standardized flows were made, covering a period of nearly a year, for establishing a wholly independent and authoritative coefficient which could be relied on in actual service. This work was confined to various-sized orifices in 8 and 10-in. pipes, inasmuch as the quantities to be measured first were large and would necessitate such installations.

27 In the meantime, while these calibrations were being made, the manufacturers of differential recording gages were urged to perfect and deliver recording differential gages having a range of zero

to 50 in. water pressure, and operated by an iron float in a mercury U-tube. (Fig. 2.) These gages were finally perfected and delivered about the time of the completion of the Joplin tests, and about twenty of the most important installations were made in rapid succession.

ORIGINAL CONSTANTS CALLED IN QUESTION

28 With the results of these installations, after the first two months' operation some serious inconsistencies appeared which were attributed to fundamental error of some kind in the principle being

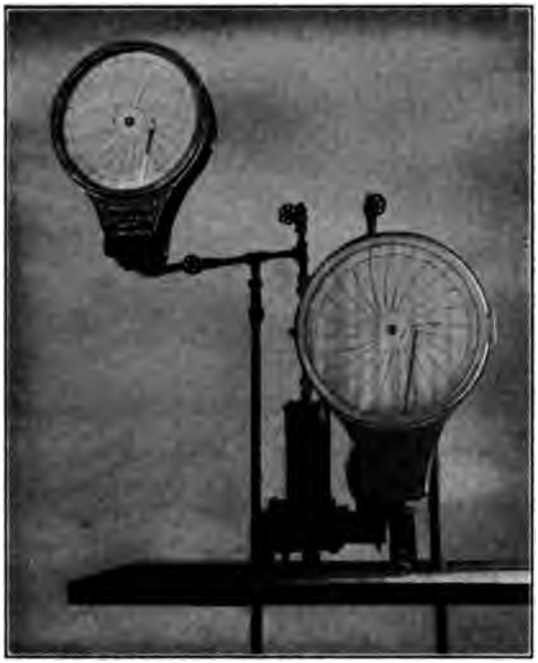


FIG. 2 MERCURY FLOAT DIFFERENTIAL GAGE WITH SEPARATE PRESSURE RECORDER

employed. We considered it necessary at once to verify the following basic assumptions not previously experimented on.

First Assumption. Coefficients determined by experiments with differentials of $4\frac{1}{2}$ in. of water or less, representing velocities of less than 100 ft. per min., are constant with velocities up to 300 ft. per min., represented by differentials up to 50 in. of water, which were utilized in actual service.

Second Assumption. Coefficients determined at atmo
pressure are constant and unchanged by static pı
up to 365 lb. absolute.

Third Assumption. Coefficients determined with air a
of saturation is applicable to natural gas by a
correction for specific gravity.

Fourth Assumption. Natural gas is a perfect gas,
pressure and volume relations according to Boyle'

FIG. 3 ORIGINAL ARRANGEMENT OF ORIFICE FLANGE IN THE L

All the above assumptions had been taken previously
question, but the operating results pointed out that the
be verified or rejected and suitable correction factors determi

WORK TO REËSTABLISH CONSTANTS

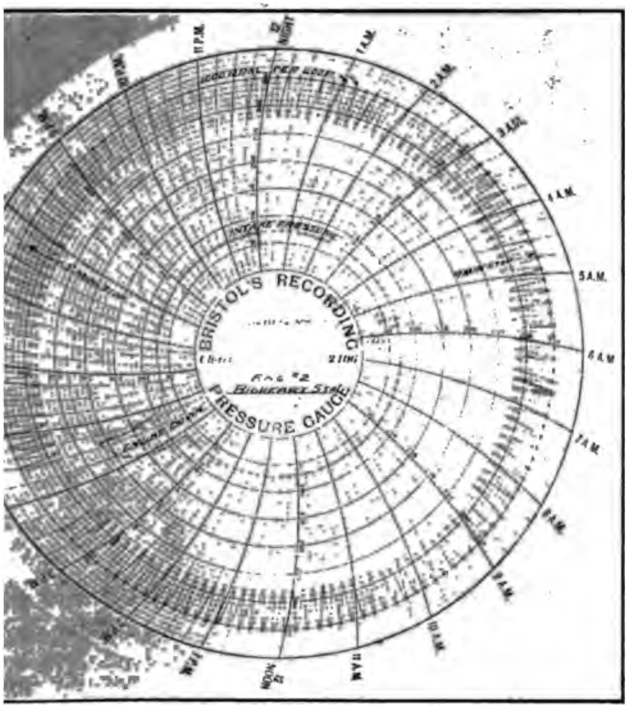
29 The work of verifying these assumptions was undert
three ways. The first was by a careful comparison under o
conditions of orifice meter registration with a calibrated com
displacement.

30 There was available at Bigheart, Okla., a compressor
consisting of three duplex units of about 950 h.p. rating twin-
gas engines, each driving two double-acting compressor c
with poppet valves, only one engine being used in the test.

31 Fifty miles distant, near the north end of the Cushin
was located a measuring station with a 10-in. orifice mete

disk with 6½-in. opening. The dimensions of the inter-
: are:

Actual I.D.	Area, Sq. In.	Area, Sq. Ft.	Length	Capacity, Cu. Ft.
17.375	237.10	1.646	42,091	69,300
15.375	185.66	1.289	79,823	102,900
10.192	81.585	0.5661	6,747	3,822
12.250	117.860	0.8185	84,549	69,090
11.4316	102.636	0.7125	28,947	20,630
		Total Cu. Ft.		265,742



HART OF REVOLUTIONS OF COMPRESSOR AND INTAKE PRESSURE

actor to be determined was the leakage. The line was
out 300 lb. and closed in. After the pressure had equalized,
res were carefully taken at half-hour intervals until the
eached 270 lb. Then the compressors were started and

the pressure was reduced to 230 lb. After the pressure was equalized again, the pressures were taken as before. In this way the leakage was determined at a number of points, approximately 280, 225, 165, and 115 lb., giving the same results that would have been obtained by allowing the whole volume of gas to leak out. Fig. 6 gives a graphical representation of the pressure log during the leakage test. Fig. 7 was derived from Fig. 6 by plotting to a large scale and drawing a fair curve through the resulting points to eliminate errors

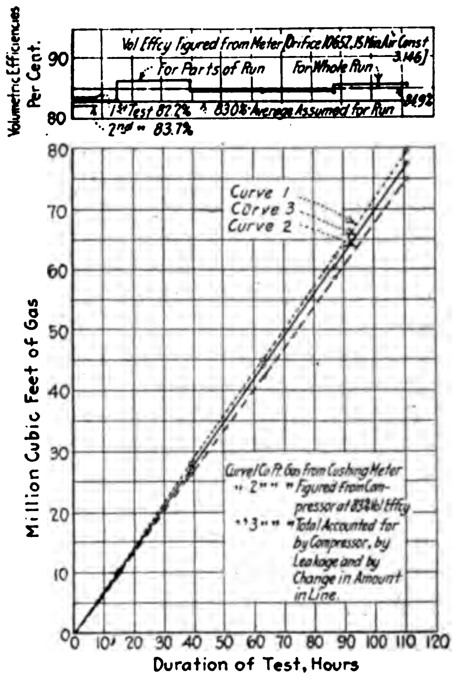


FIG. 5 CURVE SHOWING NET RESULTS OF PUMP STATION DISPLACEMENT TESTS

in gage readings. Then the rates of drop at various pressures were plotted. To determine rate of leakage at any pressure, multiply line volume by drop in pressure per hour and divide the result by 14.41, which will give the cubic feet of gas leakage per hour.

32 After determining the leakage as shown in the curves in Figs. 6 and 7, one of the compressor units was carefully overhauled to secure good valve and packing-ring conditions. An automatic counter equipped with indicating chart was mounted on the second-

ary shaft with a pressure spring automatically recording the intake pressure of the compressor cylinders, tracing a line on the same chart. This gave a chart as shown in Fig. 4, in which the outer loops each represent 1000 revolutions. The line was filled and shut in at both ends and allowed to equalize. The compressor was started and the intake pressure held constant by regulating a gate on the station intake. Fifteen-minute readings were taken on all gages and thermometers. The total and intermediate numbers of revolutions of the compressor were taken from a revolution counter. The line

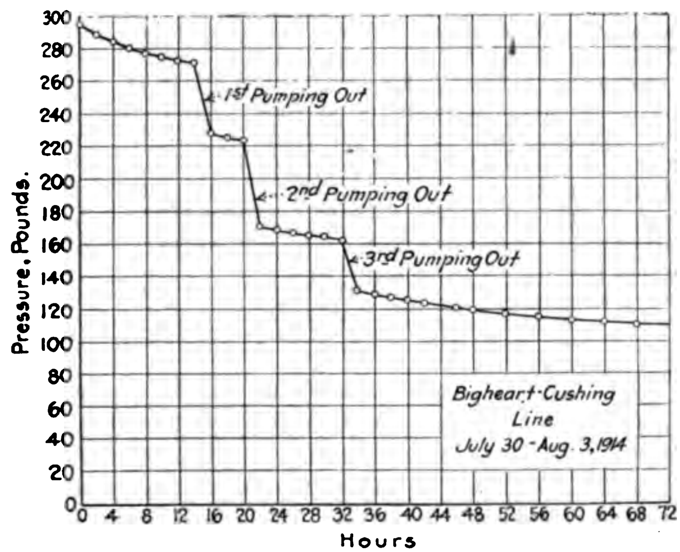


Fig. 6 Pressure Log During Leakage Test

pressure being taken at the start and finish of the test, the volume of gas taken from the line would equal the initial pressure minus the final pressure divided by 14.41, the result multiplied by the line volume. From this the leakage was deducted, leaving the volume of gas taken by the compressor. With the proper temperature corrections, the volume of gas taken by the compressor divided by the cylinder displacement during test, equals volumetric efficiency. Two reliable tests were made, and the average of the two results was taken as the final figure. The results of two such runs were as follows:

	August 4, 1914	August 5, 1914
Time of Start	8 : 35 : 30 a.m.	8 : 52 : 15 a.m.
Time of Stop	12 : 09 p.m.	1 : 10 p.m.
Duration of Test	213½ min.	258 min.
Average Intake Pressure, lb. per sq. in.	121.5	122.3
Average Intake Temperature, deg. fahr.	81	81
Revolutions during Test	22,445	27,085
Average rev. per min.	105	105
Cylinder Displacement per Revolution, cu. ft.	13.496	13.496
Cylinder Displacement for Test, cu. ft.	2,859,543	3,447,034
Average Line Temperature, deg. fahr.	80	79.5
Correct Line Pressure { Start, lb. per sq. in.	258.25	289.75
Correct Line Pressure { Stop, lb. per sq. in.	127.75	129.50
Drop in Line Pressure during Run, lb. per sq. in.	130.50	160.25
Rate of Leakage at average L.P., lb. per hour	0.98	1.01
Drop of Pressure due to Leakage, lb. per sq. in.	3.43	4.34
Drop in Pressure due to Pumping Out, lb. per sq. in.	127.0	155.91
Effective Drop in Atmospheres	8.81	10.82
Volume of Line, cu. ft.	265,742.0	265,742.0
Gas taken from line by Compressors, cu. ft.	2,344,643.0	2,875,328.0
Corrected to Intake Temperature, cu. ft.	2,345,544.0	2,886,829.0
$\frac{\text{Gas from line}}{\text{Compressor Displacement}}$ = Vol. Efficiency	82.2%	83.7%

Average Volumetric Efficiency = 83.0%

33 Following this determination gas was drawn through the measuring station, and under test conditions the compressor and meter were operated in tandem for 111 hr. divided into 5 periods. In this test, all conditions at Bigheart were practically the same as during the volumetric efficiency test. The Cushing end of the line was opened and the line filled until the pressure at Bigheart was above 125 lb. Then the engine was started, and after running conditions were constant the test was started and continued for 5 days. Half-hourly readings were taken at both ends of the line. With the exception of the first period, the results were worked up in 24-hr. periods. Figures for a typical period are as follows:

Third Period of Run

4:00 p.m., August 8, to 4:00 p.m., August 9, 1914.
Leakage and Change of Line Pressure during Test:
Bigheart 167 lb. to 178 lb. = 11 lb. increase
Cushing 210 lb. to 218 lb. = 8 lb. increase
Assumed average increase = 10 lb.
Cushing average Pressure 214 lb.

Bigheart average Pressure 173 lb.

41 lb. $41 \times \frac{2}{3} = 27.4$

$173 + 27 = 200$ lb., assumed average Line Pressure.

Rate of Leakage at 200 lb. from Test Curve = 0.97 lb. per hour or 23.3 lb. per 24 hours.

Volume of Line = 265,742 cu. ft.

$$\frac{10 + 23.3}{14.41} \times 265{,}742 = 613{,}000 \text{ cu. ft. metered but not compressed.}$$

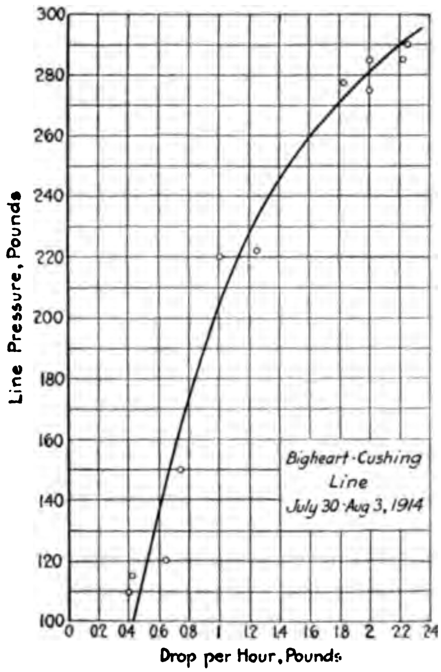


Fig. 7 Pressure Log During Leakage Test

COMPRESSOR CALCULATIONS

Average Intake Pressure, 123 lb. Average Intake Temperature 82° F.

Displacement per Revolution = 13.496 cu. ft.

Revolutions during Test = 151,899. R.P.M. = 105.3

$$151{,}899 \times 13.496 \times \frac{122.5 + 14.41}{14.41} = 19{,}420{,}000 \text{ cu. ft. displaced at } 100\%$$

Volumetric Efficiency.

Average Volumetric Efficiency from Test = 83%.

19,420,000 × 0.83 = 16,120,000 cu. ft. of Gas pumped at 14.41 lb. and 82° F

16,120,000
613,000

16,733,000 cu. ft. = amount of Gas accounted for at 14.41 lb. and 82° F.

METER CALCULATIONS

15-Minute Air Coefficient at 14.41 lb. and 60° = 3.146 per M.

Specific Gravity of Gas during Test = 0.714

$$\frac{3.146}{(0.714)^{\frac{1}{2}}} = 3.725 = \text{15-Min. Coefficient for Gas at 14.41 lb. and 60° F.}$$

Average Meter Temperature = 82° F.

Sum of Extensions for Test = 2240.551

2240.551 × 2 = 4481.102, extensions for 15-Min. Coefficient.

3.725 × 4481 = 16,690,000 cu. ft., Quantity that would pass meter with a meter temperature of 60° F.

$$16{,}690{,}000 \times \left(\frac{520}{542}\right)^{\frac{1}{2}} = 16{,}350{,}000$$ cu. ft. of Gas at 14.41 lb. and 60° F. with Meter Temperature of 82° F.

$$16{,}350{,}000 \times \frac{542}{520} = 17{,}045{,}000 \text{ cu. ft. at 82° F.}$$

COMPARISON

17,045,000
613,000

16,432,000

$$\frac{16{,}432{,}000}{19{,}440{,}000} = 84.6\%,$$ Volumetric Efficiency of Compressor as shown by Meter.

$$\frac{17{,}045{,}000}{16{,}733{,}000} = 101.8\%$$

The object of this test is to account for the gas metered by the Cushing meter through orifice No. 10,652, using the meter coefficient determined by the Joplin experiments and the recording pressure and differential gages that are in use at the Cushing station. The factors entering into the comparison, for a certain interval of time, are: quantity of gas metered; quantity lost by leakage; quantity put into or taken out of line by change in line pressure; and quantity taken from line by compressors.

$$\text{Quantity of gas metered} = \begin{cases} \text{Gas lost by leakage} \\ \text{Gas added to or taken from line by change of pressure} \\ \text{Gas compressed} \end{cases}$$

All gages and thermometers used were calibrated.

34 The final result of this comparison is illustrated graphically by Fig. 5, in which the net result of a comparison of the compressor

station with the orifice meter shows a constant difference of approximately 2 per cent. Following this test a continuous comparison was carried on by checking monthly totals to secure information as to the consistency of this agreement over a long period of time. At other points these monthly checks were carried on by means of the pipe line flow formula proposed by Thos. R. Weymouth in a paper paper read before the Society.[1] For instance, during the month of June, 1915, the amount of gas through the Cushing-Bigheart line compared day by day showed a total compressor station displacement of 377,110,000 cu. ft., using the same volumetric efficiency found by the test. The total orifice meter reading for the same period was 382,480,000, an excess of 1½ per cent. A check of another orifice meter measuring gas through a line consisting of 5.59 miles of 8-in. inside diameter and 9.23 miles of 10-in. inside diameter line gave a flow, by Mr. Weymouth's formula for drop in line pressure, of 333,420,000; and the orifice meter gave totals for the same month of 338,900,000 cu. ft., an excess of 1½ per cent. The pressure in the last two comparisons ranged from 200 to 290 lb. Leakage allowances were made in each case on the same basis as referred to and shown in Fig. 7, for one of these lines.

WANN DISPLACEMENT TESTS

35 The pump station and line flow checks gave approximate corroboration in certain specific instances of the value of the coefficient determined at Joplin, but it was admittedly necessary to get a verification covering as wide a range of pressures and differentials as possible. For this purpose we chose a section of line running from Caney, Kan., to a point near Wann, Okla., which was not in use at the time, and was located so that it could be filled with gas at any time up to a pressure of 250 lb. The dimensions of this line were:

1 From Wann-Portland to 12-in. gate, 1 mile west, Jan. 23, 1914.

5,236 ft. of 12-in. (11½ in. I.D.)	× 0.7213 =	3,776.73 cu. ft.
286 ft. of 10-in. I.D.	× 0.5476 =	156.61 cu. ft.
48 ft. of 8-in.	× 0.3474 =	16.67 cu. ft.
Total volume between 12-in. and 8-in. gates =		3,950.01 cu. ft.

2 From Wann-Portland to 18-in. gate, 3½ miles west.

2,554 ft. of 18-in. (17$\frac{5}{8}$ in. I.D.)	× 1.6347 =	4,175.00 cu. ft.
18,554 ft. of 12-in. (11½ in. I.D.)	× 0.7213 =	13,383.00 cu. ft.

[1] Problems in Natural Gas Engineering, Trans.Am.Soc.M.E., vol. 34, p. 185.

286 ft. of 10-in.	$\times$ 0.5476 =	156.61 cu. ft.
48 ft. of 8-in.	$\times$ 0.3474 =	16.67 cu. ft.
Total volume between 18-in. and 8-in. gates	=	17,731.28 cu. ft.

3 From Wann-Portland to 18-in. gate at Caney, Kan., station.

17,776 ft. of 18-in. (17$\frac{5}{16}$ in. I.D. $\times$ 1.6347	=	29,058.4 cu. ft.
Total of (No. 2)	=	17,731.28 cu. ft.
Total cu. ft. in line	=	46,789.68 cu. ft.

There were three gates on the line, so located that the latter could be used as a reservoir having a computed volume of either 46,730, 17,670, or 3,950 cu. ft., according to the size of the meter to be tested. Careful leakage tests were run on the various sections of

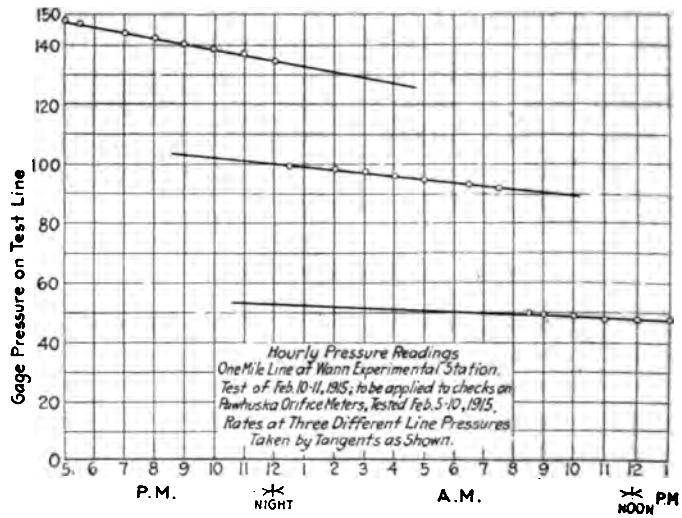


FIG. 8 HOURLY PRESSURE READINGS IN WANN TESTS

this line. Figs. 8 and 9 show the leakage on the smallest division at different line pressures. In making these leakage tests, care was taken to seal the gates at each end of the section in use and drain the line beyond the gate at each end through an observed opening to prevent unknown leakages in, or out, through the gate valves. This station was connected up as shown in the diagram, Fig. 10, so arranged that gas from this reservoir was allowed to escape through three orifice-meter flanges in tandem, one 8-in. and two 10-in. It was then led through a reducing valve manipulated by an operator, and through a section of line about 300 ft. long to a large discharge chamber, Figs. 11 and 12. This chamber consists, as will be seen, of an initial space containing a series of conical baffles to break up eddies in the current, and a second chamber of larger dimensions with an additional baffle having at the outlet end a large flange for conveniently changing the outlet plate. These plates had a central orifice through which gas was discharged to the air by a head in the discharge chamber, maintained as near an absolute constant as possible for giving a test run. This head was observed at one-minute intervals by a sensitive inclined draft gage, and a second similar draft gage attached to the same line was near the man who regulated

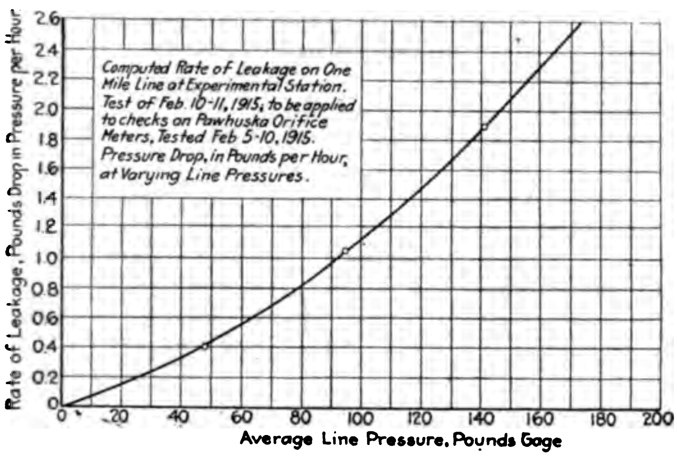


FIG. 9 COMPUTED RATE OF LEAKAGE IN WANN TESTS

the reducing gate. This manipulation was carefully controlled to maintain this head as nearly uniform as possible.

36 It was the aim to maintain a uniform rate of flow from start to finish of each run, although with a wide variation in pressure and differential on the orifice meters in series. Temperatures were taken at each succeeding point on the line and corresponding corrections made. This work was continued from June 20 to the end of September, 1914. Nearly 1200 determinations of coefficients were made. For the purpose of showing the factors taken into consideration, the complete record of a typical day's run on September 27, 1914, is here given. The disks installed in the flanges that day were:

Flange A. Disk 10,503, 5-in. orifice in 10-in. line
Flange B. Disk 10,402, 4-in. orifice in 10-in. line
Flange C. Disk 8,409, 4-in. orifice in 8-in line.

37 Table 1 is the observed data for determining the amount
gas escaping from the reservoir in the six successive runs, and t
specific gravity of the gas in each instance.

38 The following is a key to the symbols used as summaries
readings and calculated values, as tabulated in Table 2, in regard
the Wann displacement tests.

Column number	Symbol	DESCRIPTION
1	h	Observed differential drop in in. of water, across met orifice disk
2	P_c	Gage reading (P), corrected to calibration
3	P_a	Corrected gage reading, lb. absolute

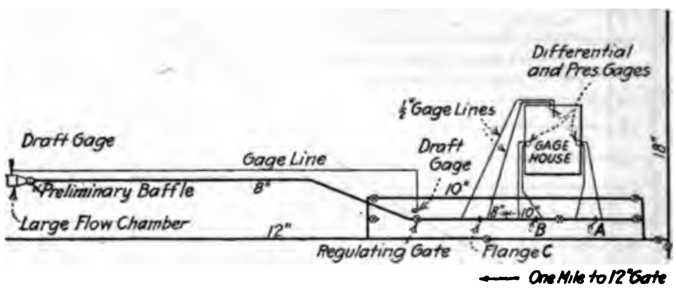


FIG. 10 WANN TESTING STATION CONNECTIONS

4	$(hP)^{\frac{1}{2}}$	Calculated square root of the product of "h" and "P_a"
5	R_1	Calculated rate flow, expressed in cu. ft. per 15 min., average line temperature and standard barometric pr sure of 14.41 lb. per sq. in. This value is based on t average rate of displacement and corrected for variatic in prover temperature
6	T_p	Observed temperature of gas at prover
7	T_m	Observed temperature of gas at orifice meter
8	R_2	Calculated value of R_1 reduced from average line te perature to observed orifice meter temperature
9	C_m	Calculated 15-min. meter constant, for gas at temperat T_m. Found by dividing R_2 (column 9) by $(hP)^{\frac{1}{2}}$ (column
10	C_g	"C_m" corrected to 60 deg. fahr.
11	C_a	Calculated 15-min. meter constant, for air at standard c dition of 14.41 lb. per sq. in. absolute, and 60 deg. fa

Table 2 gives the actual observations for determining the meter constants. The significant results of this series of observations is column 12 giving the air constant for each of the disks tested.

39 A study of the results of this work led to the belief that assumptions 1 and 2, Par. 28, were not perfect, and that it would be necessary to provide means for using a variable factor instead of a constant heretofore applied on orifice meters.

TABLE 1 READINGS AND RESULTS OF DISPLACEMENT CHECKS

Small and Large Provers in Tandem with Three Meter Orifices
(Nos. 10,503, 10,402 and 8,409)

September 27th, 1914.

12-in. Gate, one mile west of station, shut.
Computed Volume, 3,950 cu. ft.

No. or Size of Prover Orifice	Time, Start and Finish	Duration of Test, Min. Sec	Corr. Line Pressures, Start and Finish	Barometer, In. Hg.	Quantity in Cu. Ft. at Line Temperature: Gross for Entire Test	Leakage for Test	Net for Entire Test	Net for Fifteen Min.	Temp. of Line	Avg. U-Tube, In.	Avg. Prover Temp.	Gravity of Gas
3605	1: 22: 00 1: 05: 35	16 25	218¾ 71	29.60	40,490	350	40,140	36,690	70	4.43	72	0.696
3005	2: 06: 00 1: 45: 55	20 05	225¾ 71¾	29.60	42,190	420	41,770	31,200	70	4.17	75	0.696
2705	3: 42: 45 3: 10: 55	31 50	230¼ 3	29.60	62,250	570	61,680	29,060	68	4.19	77	0.696
16 in.	4: 54: 15 4: 46: 20	7 55	220 30½	29.60	51,920	140	51,780	98,170	68	2.74	65	0.696
12½ in.	5: 36: 50 5: 23: 05	13 45	214¼ 13¼	29.60	54,930	230	54,700	59,670	66	2.76	66	0.696
12½ in.	6: 08: 30 5: 53: 10	15 20	207¾ 13¾	29.60	53,160	240	52,920	51,790	65	2.04	63	0.696

40 Inasmuch as the number of records to be computed in ordinary operation is so great, the considerable clerical labor involved could not be put in the hands of technical men and it was deemed advisable to devise a slide rule which could be continually used with a variable factor in accordance with the pressure and differential shown at different points on the chart. A dividing head was constructed and a slide rule as shown in Fig. 13 laid off thereon to be used instead of the familiar book of extensions used ordinarily in pitot and

TABLE 2 WANN DISPLACEMENT TESTS, SEPTEMBER 27, 1914

Large Prover in Tandem with Three Meter Orifices. One Mile of 12" Line Used

Disk No. 8,409[1] Flange C

1 h	2 P_c	3 P_a	4 $(hP)^{\frac{1}{2}}$	5 R_1	6 T_p	7 T_m	8 R_2	9 C_m	10 C_g
16" Orifice: B — 29.60"; Line Temperature 68°; Average Prover Temperature 65°; Aver Flow, 98,170 Cu. Ft.; Gravity of Gas, 0.696									
31.38	190½	205.00	80.15	97,880	68	70	98,270	1.226	1.214
40.35	148½	163.00	81.05	98,070	66	69	98,270	1.212	1.201
46.88	123½	138.00	80.30	98,270	64	68	98,270	1.223	1.213
12" Orifice: B — 29.60"; Line Temperature 66°; Average Prover Temperature 66°; Aver Flow, 59,670 Cu. Ft.; Gravity of Gas, 0.696									
11.20	197	211.50	48.62	59,440	70	69	59,790	1.229	1.219
12.05	185	199.50	49.01	59,560	68	69	59,910	1.222	1.212
12.82	175	189.50	49.25	59,610	67	68	59,850	1.215	1.206
13.97	162½	177.00	49.70	59,670	66	68	59,910	1.204	1.195
15.02	147½	162.00	49.30	59,730	65	68	59,970	1.214	1.205
16.71	132½	147.00	49.53	59,790	64	67	59,910	1.208	1.200
18.40	119¼	133.75	49.53	59,790	64	67	59,910	1.208	1.200
20.14	108	122.50	49.65	59,790	64	66	59,790	1.204	1.197
22.73	96	110.50	50.10	59,790	64	66	59,790	1.193	1.186
27.72	75¾	90.25	49.97	59,790	64	66	59,790	1.196	1.189
38.61	51½	66.00	50.42	59,790	64	66	59,790	1.185	1.178
46.05	41½	56.00	50.73	59.790	64	66	59.790	1.178	1.171
12" Orifice: B — 29.60"; Line Temperature 65°; Average Prover Temperature 63°; Ave Flow, 51,790 Cu. Ft.; Gravity of Gas, 0.696									
8.86	194	208.50	42.98	51,640	65	69	52,040	1.211	1.201
9.18	180¼	194.75	42.29	51,690	64	68	51,990	1.228	1.219
9.97	170	184.50	42.84	51,740	63	68	52,040	1.215	1.206
10.43	158½	173.00	42.46	51,740	63	67	51,940	1.222	1.214
11.43	145½	160.00	42.80	51,740	63	67	51,940	1.213	1.205
12.16	136	150.50	42.77	51,790	62	66	51,890	1.213	1.206
13.14	126¼	140.75	42.97	51,790	62	66	51,890	1.208	1.201
14.48	113½	128.00	43.00	51,790	62	66	51,890	1.206	1.199
15.65	103	117.50	42.85	51,790	62	66	51,890	1.209	1.202
17.68	91½	106.00	43.23	51,790	62	65	51,790	1.197	1.191
20.68	74½	89.00	42.90	51,790	62	65	51,790	1.207	1.201
25.73	58	72.50	43.19	51,790	62	65	51,790	1.197	1.191
31.73	46	60.50	43.80	51,790	62	65	51,790	1.182	1.176
41.94	31	45.50	43.61	51,790	62	65	51,790	1.187	1.181

Small Prover in Tandem with Three Meter Orifices. One Mile of 12" Line Used
Disk No. 8,409; Flange C

1 i	2 P_c	3 P_a	4 $(hP)^{\frac{1}{2}}$	5 R_1	6 T_p	7 T_m	8 R_2	9 C_m	10 C_g	11 C_a
No. 3005 Orifice; B — 29.60"; Line Temperature, 70°; Average Prover Temperature, 72°; Average Flow, 36,690 Cu. Ft.; Gravity of Gas, 0.696										
3.81	207¼	221.75	29.03	36,580	75	74	36,860	1.268	1.252	1.044
4.22	191¼	205.75	29.41	36,620	74	74	36,900	1.252	1.236	1.031
4.47	182¼	196.75	29.61	36,620	74	73	36,830	1.243	1.228	1.024
4.63	176	190.50	29.68	36,650	73	73	36,880	1.242	1.227	1.023
4.95	162	176.50	29.52	36,690	72	72	36,830	1.247	1.233	1.028
5.25	153¼	168.00	29.88	36,690	72	72	36,830	1.241	1.227	1.023
5.74	143¼	158.25	30.11	36,730	71	70	36,730	1.220	1.208	1.008
5.92	136	150.50	29.80	36,760	70	70	36,760	1.232	1.220	1.017
6.34	125¾	140.25	29.80	36,800	69	70	36,800	1.232	1.220	1.017
7.49	108¾	123.25	30.39	36,830	68	70	36,830	1.212	1.200	1.001
8.08	100	114.50	30.39	36,830	68	70	36,830	1.212	1.200	1.001
9.56	82	96.50	30.37	36,870	67	70	36,870	1.214	1.202	1.003
No. 3005 Orifice; B — 29.60"; Line Temperature 70°; Average Prover Temperature, 75°; Average Flow, 31,200 Cu. Ft.; Gravity of Gas, 0.696										
2.65	215½	230.00	24.69	31,050	80	74	31,290	1.268	1.251	1.044
2.94	202¼	216.75	25.22	31,110	78	74	31,290	1.242	1.225	1.022
3.07	190	204.50	25.08	31,140	77	73	31,320	1.249	1.233	1.028
3.25	179¾	194.25	25.12	31,170	76	72	31,290	1.245	1.230	1.025
3.43	169½	184.00	25.12	31,170	76	72	31,290	1.245	1.230	1.025
3.65	155½	170.00	24.90	31,200	75	72	31,320	1.257	1.242	1.036
4.37	144	158.50	26.30	31,230	75	70	31,230	1.187	1.175	0.980
4.54	137	151.50	26.20	31,230	74	70	31,230	1.192	1.180	0.984
4.80	129½	144.00	26.29	31,230	74	70	31,230	1.188	1.176	0.981
5.20	117½	132.50	26.19	31,230	74	70	31,230	1.198	1.181	0.985
5.46	109½	124.00	26.00	31,230	74	70	31,230	1.201	1.189	0.993
6.07	99½	114.00	26.29	31,260	73	70	31,200	1.189	1.177	0.982
6.53	91	105.50	26.24	31,260	73	70	31,260	1.191	1.179	0.984
7.33	83	97.50	26.71	31,260	73	70	31,260	1.170	1.158	0.966
No. 2705 Orifice; B — 29.60"; Line Temperature, 68°; Average Prover Temperature, 77°; Average Flow, 29,060 Cu. Ft.; Gravity of Gas, 0.696										
2.30	214¾	229.25	22.94	28,850	84	73	29,000	1.263	1.248	1.041
2.55	195	209.50	23.10	28,940	81	71	29,120	1.261	1.249	1.042
2.70	184	198.50	23.13	28,970	80	71	29,150	1.259	1.247	1.040
3.10	161½	176.00	23.35	29,150	74	70	29,270	1.253	1.242	1.036
3.79	133½	148.00	23.68	29,150	74	70	29,270	1.236	1.225	1.022
4.53	113½	128.00	24.07	29,150	74	70	29,270	1.216	1.205	1.006
6.07	81½	96.00	24.13	29,150	74	70	29,270	1.212	1.201	1.002
6.85	70	84.50	24.06	29,150	74	68	29,150	1.212	1.201	1.002
7.[illegible]	63	77.50	24.39	29,150	74	68	29,150	1.195	1.186	0.990
8.94	52½	67.00	24.46	29,150	74	68	29,150	1.192	1.183	0.987
10.05	43½	58.00	24.13	29,150	74	68	29,150	1.207	1.198	1.000
11.28	37	51.50	24.10	29,180	73	68	29,180	1.211	1.202	1.003
13.40	29¼	43.75	24.19	29,180	73	68	29,180	1.206	1.197	0.999
16.23	23	37.50	24.62	29,180	73	68	29,180	1.184	1.175	0.980
19.96	16	30.50	24.65	29,210	72	66	29,090	1.179	1.172	0.977
26.70	8½	23.00	23.00	29,210	72	66	29,090	1.174	1.167	0.973

orifice meter computations. This extension book is merely a tabu lation of the value of the $\sqrt{hP}$, to which the meter constant ma be applied as a direct multiplier, to get the 15 min. or hourly flow a desired.

41 Fig. 14, developed in collaboration by Prof. P. F. Walke is a series of curves showing results of the entire tests segregated a to differential and pressure, with a view to the determination of th available factor to be used if a variable coefficient should becon necessary.

QUESTION OF AN IMPERFECT GAS

42 Assumption 3, Par. 28, that coefficient determined for a would apply for gas, and assumption 2, in part, that coefficie

Fig. 11 Outflow Chamber with Separate Orifice Plates

determined at low pressure would apply for high static pressur were called in question in a paper read by Prof. P. F. Walker befo the Society.[1] It was therefore deemed necessary to pursue the wo suggested by him at that time to a conclusion, which could be use as a basis of calculation. In consultation with Professor Walker was decided to inaugurate simultaneously analogous experiments i the University of Kansas laboratory and at the Wichita Natural Ga Company's laboratory in Bartlesville. Independently and simu taneously, S. S. Wyer, of Columbus, and Robert F. Earhart, workin in the laboratories of the Ohio State University, were conductin

[1] Physical Laws of Methane Gas, Trans.Am.Soc.M.E., vol. 36, p. 781.

similar experiments. Up to the present time the findings in these three series of tests are somewhat diverse and not yet ready to submit. The final conclusion of this research will be of great importance to the natural gas industry, and will determine finally the necessity for a variable coefficient in orifice and pitot meter determinations.

PROBLEM OF SPECIFIC GRAVITY

43 A very important factor hitherto unmentioned is that of specific gravity. Below are given gravity determinations in three

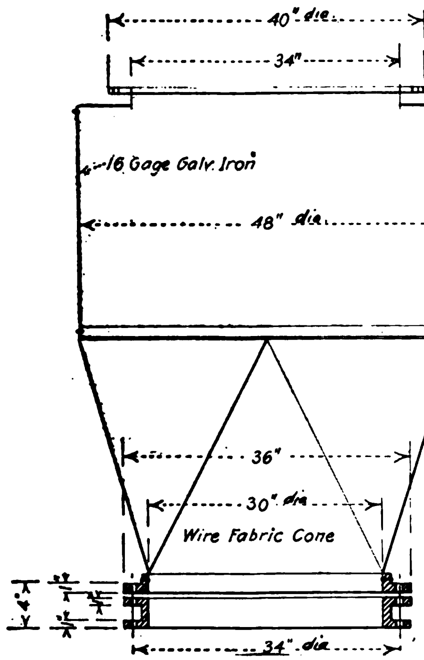


Fig. 12 Outflow Chamber with Separate Orifice Plates

different fields taken at several months' intervals, and a series of simultaneous gravities in seven different fields:

North Osage Gravities		Cushing Gravities	
January, 1914	0.585	May, 1914	0.640
September, 1914	0.635	November, 1914	0.6995
November, 1914	0.653	March, 1915	0.7505
April, 1915	0.647	October, 1915	0.695

Lot 51

March, 1915	0.597
May, 1915	0.603
October, 1915	0.627

Simultaneous Gravities]

Sycamore Field	0.5765	Augusta Field	0.666
Local Iola Field	0.590	Chilocco Field	0.6925
Humboldt Field	0.601	Sedan Field	0.765
Pawhuska Field	0.630		

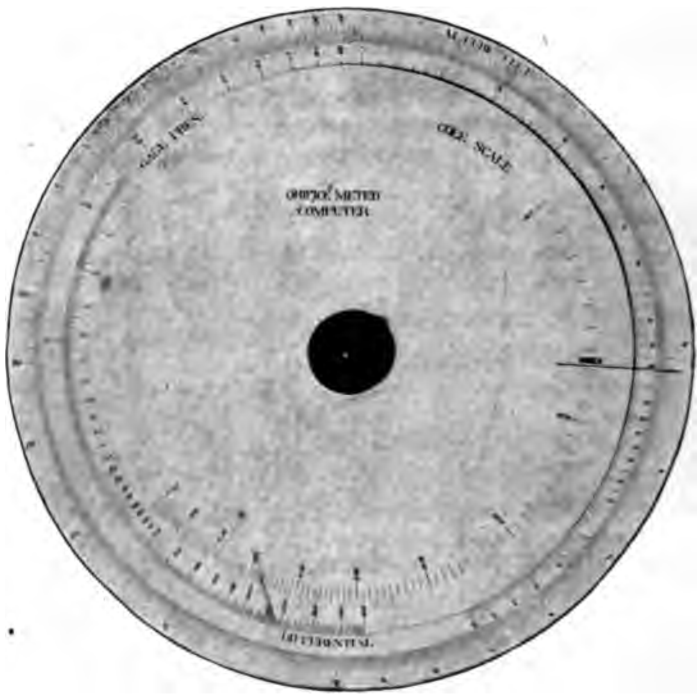


FIG. 13 SLIDE RULE FOR CALCULATING ORIFICE METER RESULTS

In actual practice, gravity determination of gas at a given purchase meter frequently affects a correction in the monthly settlement of from $3000 to $8000 as compared with the gravities taken a month or two previous, or an estimated gravity such as is frequently used. The method of determining this specific gravity with diffusion instruments is rather a delicate manipulation. Even with experienced operators results are liable to vary with the same sample, and it is of great commercial importance to eliminate this uncertainty. It was, therefore, considered necessary to develop a type of gravity determining apparatus which would give approximately accurate and consistent results in the hands of inexperienced and unskilled manipulators, so that instruments could be installed at significant points on the piping system and gravities reported at frequent intervals as a matter of routine record; and thus proper correction factors could be applied to the meters in the districts affected. Several types of draft tube and gravity balance were built experimentally to solve this problem, but were rejected. Finally at the suggestion of W. C. Baxter a method was proposed using apparatus shown in Fig. 15. The principle involved is simple, consisting of an indicating balance

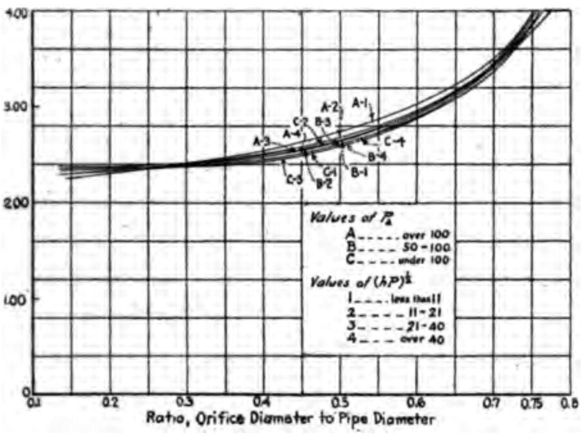


FIG. 14 CURVES ANALYZING WANN COEFFICIENTS

with a beam carrying opposed weights of widely different volume. This balance is enclosed in a chamber with observation glasses at the ends. The balance is adjusted to equilibrium with the chamber filled with air. The air is then completely expelled by a flow of gas and the gas pressure increased in the chamber until the balance is again at equilibrium. The specific gravity is determined by the relative absolute pressure of air and gas necessary to secure the same buoyancy, as shown by indicating balance. The apparatus, even in the crude form shown, seems to be quite sensitive enough to improve materially our gravity records and to involve the personal equation to but a slight degree.

NEW TESTS WITH AIR AT ERIE

44 A subsequent series of calibrations was run in 1915 to supplement the large capacity meters previously developed by the addition of a series of relatively small capacity meters in 6-in. and 4-in. pipe. The reference quantity chosen for this work was a small

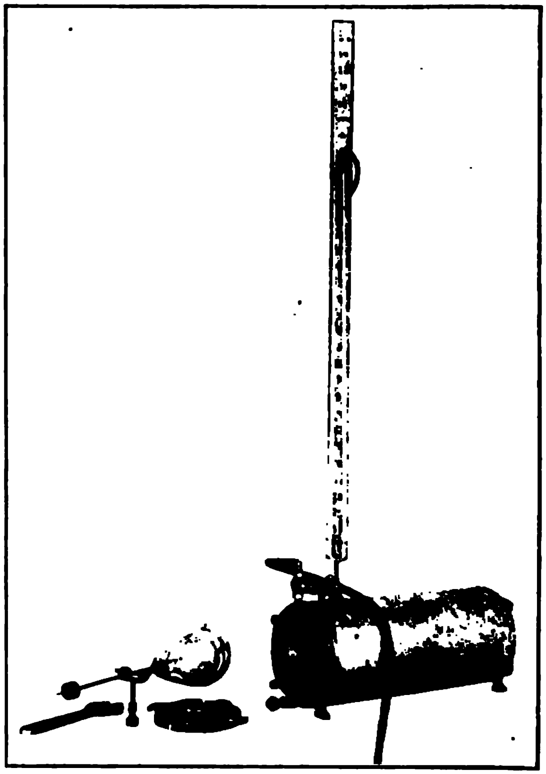


FIG. 15 BAXTER GRAVITY APPARATUS

holder located at the testing plant of the Metric Metal Works, Erie, Pa., which was courteously loaned to the company's technical staff for the purpose. Check tests were also taken with 8-in. and 10-in. pipe line orifices; in all about 130 determinations were made covering the following orifices:

No.	Size of Pipe, In.	Size of Orifices, In.	No.	Size of Pipe, In.	Size of Orific In.
4051	4	0.506	6204	6	2.002
4071	4	0.755	6302	6	3.003
4103	4	0.996	6401	6	4.000
4123	4	1.250	8101	8	1.010
4154	4	1.500	8205	8	2.006
4174	4	1.754	8304	8	3.006
4205	4	1.997	8451	8	4.500
4223	4	2.251	8506	8	5.005
4251	4	2.504	10151	10	1.500
4301	4	3.002	10302	10	3.007
6101	6	1.002	10451	10	4.502
6151	6	1.502	10601	10	5.999

45 Table 3 gives data for determination of leakage in the holder and lines. This gives a leakage correction factor for these tests. Table 4 gives the dimensions of the holder. Table 5 is a recapitulation of the derivation of the formula used in determining the air constant in this series of tests. The following is a key to the tabulation employed in Table 6:

1st Column.	Heading:	Date of Test	
2nd	"	Size of Disk	
3rd	"	Barometer	In inches of mercury.
4th	"	Feet Drop of Holder	Each foot drop is equivalent to displacement of 200 cu. ft.
5th	"	Time in Sec.	Taken by stop watch.
6th	"	P_h	Holder pressure (lb. per sq. in., absolute).
7th	"	T_h	Temperature of holder air, deg. fahr.
8th	"	T_f	Temperature of flowing air, deg. fahr.
th	"	H	Differential across disk, inches of water.
th	"	P	Absolute pressure, in lb. per sq. in. on inlet side of disk.
h	"	C_A	Fifteen-minute air constant for disk.
h	"	C_V,	Velocity coefficient, per cent.

le 6 is a specimen sheet showing a summary of determinations le on various orifices in 6-in. pipe.

PRESENT STATUS OF COEFFICIENT

46 All the orifice calibration work, up to and including the Erie tests, has been compiled and reduced to a basis shown in Figs. 16 and 17, showing the velocity coefficient for all sizes of pipe plotted on a basis of the ratio of diameters of orifice to diameters of pipe in Fig. 16, and on a basis of the ratio area of orifice to internal area

TABLE 3 LEAKAGE TEST, AUGUST 7, 1915

The readings and calculated results given here show Leakage Test made to ascertain rate of leakage from holder, so proper allowance could be made. This test was started about noon on a Saturday and ran until early Monday morning.

Time

Start	11:30 a.m.	Aug. 7, 1915.
Finish	7:00 a.m.	Aug. 9, 1915.

Readings	Temperature of Holder, Deg. Fahr.	Reading of Tape
Start	70	26.0
Finish	71	25.85

Leakage (without any allowance for change of temperature during test): 0–15 ft. drop in 43½ hr. This is equivalent to 0.0115 cu. ft. per min. Leakage (with temperature correction made): 0.0130 cu ft. per min.

TABLE 4 CALCULATED VOLUME OF HOLDER, AUGUST 10, 1915

Measured Diameter of Holder Top (Outside).
16′ 0.5″; 16′ 0.1″; 15′ 11.8″; 15′ 11.9″; 15′ 11.7″;
16′ 0.1″; 15′ 11.7″ and 16′ 0.0″

Average	15′ 11.975″
Allowance for thickness of metal, 2 thicknesses No. 16 gage iron,	0.125″
Inside Diameter,	15′ 11.850″
or	15.988 ft.

Calculated area = 200.76 sq. ft., which means that one foot drop of holder displaces 200.76 cu. ft. The nominal capacity is 200 cu. ft. per ft. drop; the actual capacity is therefore 3/8 of 1 per cent above the theoretical.[1]

Measured Circumference of Holder

Near Top	50′ 3¾″
Near Middle	50′ 4″
Near Bottom	50′ 4½″

[1] This error of 3/8 of 1 per cent in holder capacity is ignored in the calculation of all tests given in this report. It exactly counterbalances an error of 3/8 of 1 per cent in stop watch.

of pipe opening in Fig. 17. This shows the degree of variation under the conditions of the different tests, and the solid black line is an averaging line on which coefficients for actual use at the present time are based.

47 It is the belief of the author that this value may be safely applied within the limits of accuracy shown by the curve to any practicable size of pipe line without further question. The number of experiments which are incorporated in it, and the great variety of conditions under which it has been developed, have practically eliminated any personal equation of observational error of any individual test or series of tests.

48 When the exact data are available as to the factor which must be applied to Boyle's Law for gases of differing composition, it is now apparent that the observational points slightly off the curve in

TABLE 5 DERIVATION OF GENERAL FORMULA FOR CALCULATING HOLDER TESTS ON ORIFICE METER DISKS

Subscript " o " means standard conditions of pressure and temperature, i.e., 60° and 14.41 lb. per sq. in.

Subscript " h " means actual conditions of air or gas in holder. This will vary from day to day, even for the same holder.

Assuming Flow Temp. of 60°.

$$Q_o = C_a \sqrt{hP} \text{ for air,}$$

or

$$= \frac{C_a}{\sqrt{G}} \sqrt{hP} \text{ for gas.}$$

At any Other Flow Temp. T_f.

$$Q_o = \frac{C_a}{\sqrt{G}} \sqrt{hP} \sqrt{\frac{520}{T_f}}.$$

$$Q_h \times \frac{520}{T_f} \times \frac{P_h}{14.41} = Q_o.$$

$$Q_h \times \frac{520}{T_h} \frac{P_h}{14.41} = \frac{C_a}{\sqrt{G}} \sqrt{hP} \sqrt{\frac{520}{T_f}}.$$

$$C_a = Q_h \sqrt{\frac{T_f G}{hP}} \frac{P_h}{T_h} \frac{\sqrt{520}}{14.41}.$$

In this general formula derived above, there are substituted special values for reducing quantity and time of test giving

$$C_a = 284.6 \frac{\text{Tape Difference (in ft.)}}{\text{Number of Sec.}} \frac{P_h}{T_h} \sqrt{\frac{T_f}{hP}}.$$

Figs. 16 and 17 will be brought still closer to agreement, and we shall have then the necessary data to apply intelligently a variable coefficient if it is found desirable. With the slide rule shown in Fig. 13, it would be no more difficult to make routine meter calculations with a variable than with a constant coefficient.

RECORDING APPARATUS

49 It was found necessary to develop a differential recording gage to eliminate certain operating difficulties which resulted in loss

TABLE 6 SUMMARY OF ERIE TESTS ON 6-IN. ORIFICE METER DISKS

August, 1915

1	2	3	4	5	6	7	8	9	10	11	12	
8/17/15												
1	6302	29.4	6	231.0	14.63	531	529	4.645	14.57	0.5677	72.8	
2	6302	29.4	6	232.7	14.63	531	529	4.56	14.57	0.5692	73.0	73.
3	6302	29.4	6	253.5	14.63	531	529	3.86	14.54	0.5676	72.8	
4	6204	29.4	4	367.9	14.63	531	529	5.17	14.59	0.2253	65.0	
5	6204	29.4	3	292.2	14.63	531	529	4.37	14.57	0.2313	66.6	
6	6204	29.4	3	317.5	14.63	531	529	3.685	14.55	0.2315	66.9	
7	6204	29.4	4	365.5	14.63	531	529	5.17	14.59	0.2252	65.0	
8	6204	29.4	3	276.2	14.63	531	529	5.115	14.59	0.2255	65.0	
9	6204	29.4	3	290.5	14.63	532	531	4.43	14.57	0.2312	66.9	
10	6204	29.4	1	94.7	14.63	532	531	4.855	14.58	0.2256	65.3	
11	6204	29.4	1	95.6	14.63	532	531	4.61	14.58	0.2294	66.4	
12	6204	29.4	1	94.6	14.63	532	531	4.74	14.58	0.2293	66.4	
13	6401	29.4	8	166.1	14.63	532	530	3.12	14.52	1.286	92.7	
14	6401	29.4	8	169.7	14.63	532	530	2.98	14.51	1.291	93.0	93.
15	6401	29.4	8	194.3	14.63	532	530	2.26	14.49	1.296	93.3	
16	6401	29.4	8	165.6	14.63	532	530	3.13	14.52	1.289	92.9	
17	6151	29.4	1.450	239.8	14.63	532	531	5.295	14.59	0.1239	63.6	
18	6151	29.4	1.343	240.6	14.63	532	531	4.465	14.58	0.1243	63.8	64.
19	6151	29.4	1.135	240.4	14.63	532	531	3.19	14.52	0.1245	63.9	
20	6151	29.4	1.070	240.5	14.63	532	531	2.84	14.51	0.1245	63.9	
8/18/15												
1	6101	29.4	1	384.2	14.63	528	527	5.29	14.60	0.05345	61.6	
2	6101	29.4	1	425.0	14.63	528	527	4.35	14.56	0.05331	61.6	
3	6101	29.4	1	470.4	14.63	529	528	3.55	14.53	0.05326	61.6	61.
4	6101	29.4	1	556.9	14.63	529	528	2.54	14.48	0.05332	61.6	
5	6101	29.4	1	652.0	14.63	529	528	1.815	14.45	0.05397	62.3	
6	6204	29.4	1.7	154.6	14.63	530	529	5.19	14.60	0.2278	65.7	
7	6204	29.4	1	94.6	14.63	530	529	4.69	14.58	0.2306	66.7	
8	6204	29.4	1	98.6	14.63	531	529	4.30	14.57	0.2306	66.7	
9	6204	29.4	1	105.1	14.63	531	529	3.715	14.55	0.2325	67.4	
10	6204	29.4	1	111.7	14.63	531	530	3.28	14.54	0,2330	67.5	
11	6204	29.4	1	121.6	14.63	531	530	2.765	14.52	0.2334	67.6	
12	6204	29.4	1.2	165.7	14.63	531	530	2.165	14.51	0.2326	67.4	
9/2/15												
1	6204	29.6	6.3	573.0	14.71	532	530	5.16	14.71	0.2278	65.7	
2	6204	29.6	3.5	356.0	14.71	532	530	4.05	14.67	0.2306	66.7	
3	6204	29.6	2.0	235.0	14.71	532	530	2.99	14.60	0.2325	67.4	
4	6204	29.6	1.4	202.4	14.71	532	530	1.99	14.58	0.2323	67.3	
5	6204	29.6	1.0	195.8	14.71	532	530	1.09	14.55	0.2313	67.0	
6	6204	29.6	1.2	199.7	14.71	532	530	1.49	14.56	0.2333	67.7	
7	6204	29.6	1.2	245.5	14.71	532	530	1.00	14.55	0.2315	67.1	
8	6204	29.6	1.2	219.6	14.71	532	530	1.235	14.56	0.2331	67.6	
9	6204	29.6	1.4	267.4	14.71	532	530	1.12	14.55	0.2345	68 0	
10	6401	29.6	7.9	161.4	14.71	535	533	3.14	14.63	1.304	93.9	
11	6401	29.6	7.7	158.0	14.71	535	533	3.12	14.63	1.301	93.7	
12	6401	29.6	6.8	156.8	14.71	535	533	2.505	14.61	1.296	93.3	
13	6401	29 6	6.3	172.0	14.71	535	533	1.82	14 59	1 285	92.6	

of record over considerable proportion of the time for most of the meters in service. The difficulties to be eliminated were:

a rupture of spring from overload in spring type gages

b erratic calibration in spring type gages

c retarding of indicating needle by stuffing box friction in all types

d loss of mercury through leakage in mercury float type recording gages, resulting in derangement of zero level of mercury and false reading

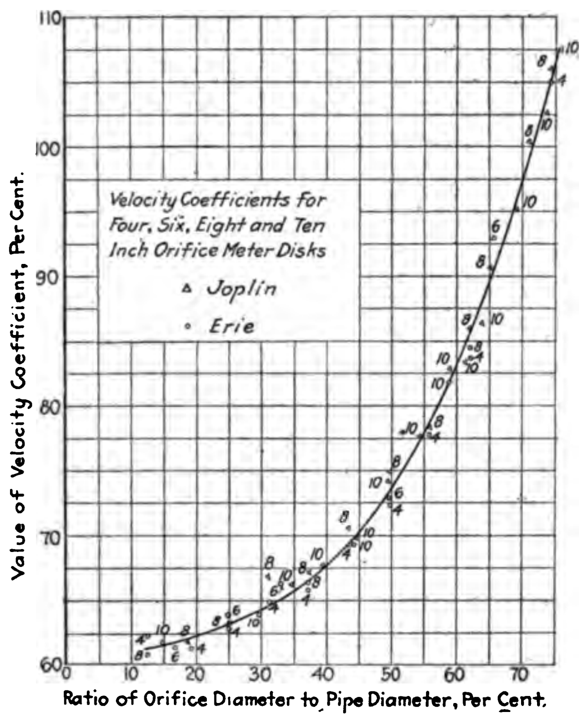


FIG. 16 VELOCITY COEFFICIENTS FOR ORIFICE METER DISKS BASED ON RATIO OF DIAMETER

e loss of mercury through stuffing box, same type gages and same difficulties

f blowing over of mercury into pipe line with strong flow of gas in same type of gages.

All these difficulties resulted in loss of record from the time any of the enumerated troubles occurred until a trained attendant could again visit the station and restore the gage to normal condition.

50 A type of mercury float gage was suggested by H. O. Ballard, superintendent of lines for this company, and has been gradually developed step by step to overcome the various difficulties enumerated. The design of the gage is shown in Fig. 18. This gage is so built that when the gas pressure exceeds the range of the gage the mercury level in the concentric manometer leg drops below the opening at *a*, Fig. 18, and a free flow of gas is permitted without violently disturbing the body of mercury, so that when the pressure again drops within the limits of the gage the fluid returns to normal registering position and the operation of the gage goes on correctly.

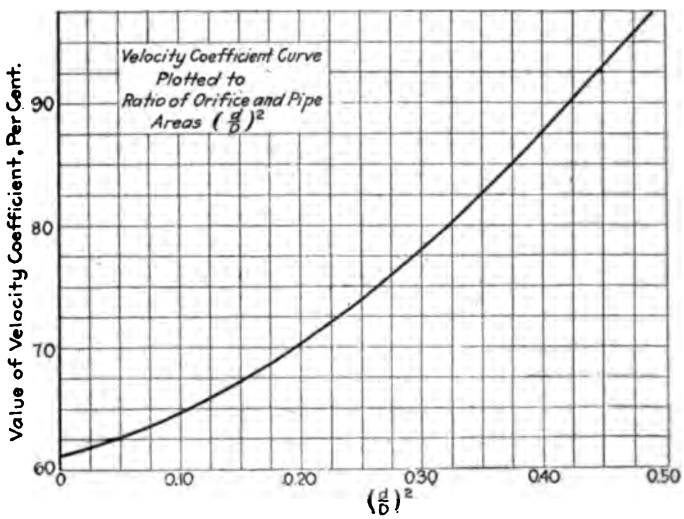


FIG. 17 VELOCITY COEFFICIENTS FOR ORIFICE METER DISKS BASED ON RATIO OF AREA

The stuffing box is far from the mercury body, and a large chamber above the mercury with a baffle at the top prevents the splash of the mercury through to the outlet pipe. A small automatic check is inserted in the line, close to the gage, which will allow a gentle flow of gas but shut off with any violent current in either direction by the raising of a light ball to the "choke" position. There is also installed a small mercury separating chamber on the high-pressure line to prevent loss of mercury by a back flow.

51 A convenience for eliminating occasional error caused by using the wrong pressure chart with a given differential record consisted of installing the pressure spring in the same apparatus and

concentric with the stuffing box of the differential gage, so arranged that differential and pressure records appear on the same chart, using one color of ink for the pressure and another for the differential. This reduces the number of clocks required and allows the installation to be made with simpler piping.

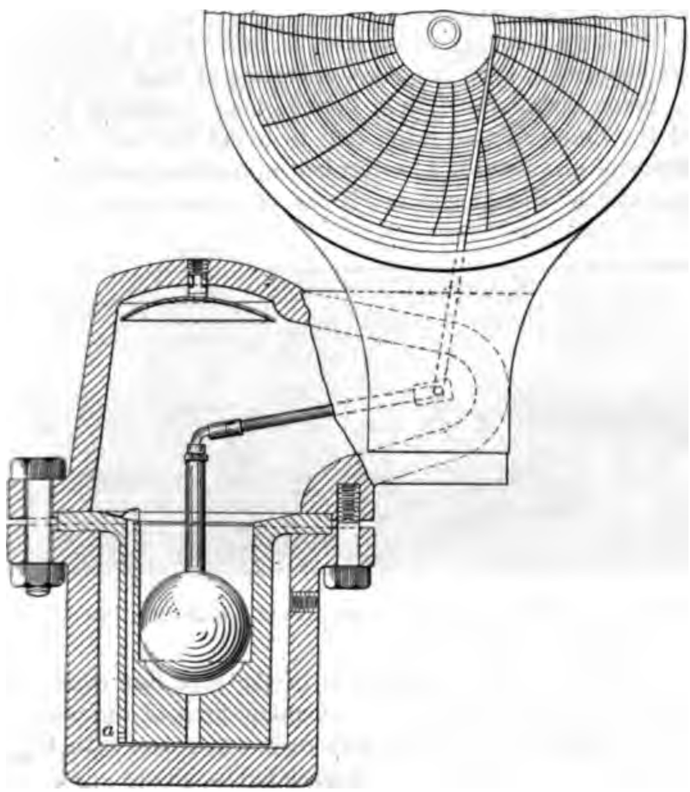


FIG. 18 BALLARD DIFFERENTIAL GAGE

LIMITATION AS TO RANGE

52 All large capacity gas meters have maximum capacity limits which must be made sufficient to take the greatest flow to be measured. The minimum limits are not so definite.

53 In all the types enumerated the accuracy deteriorates at the minimum limits. In the orifice meter, the degree of accuracy is apparent on inspection of the chart, for the reason that the resulting

percentage of error occasioned by falsely estimating the position the differential line becomes a much greater factor in the result the total differential pressure diminishes. In other words, an err which in a 10 or 12-in. differential might be negligible, would be important factor of a 1-in. differential. As a result, with our 50-i range recorders, we consider the practical limits of capacity to ha a ratio of about one to four.

54 We keep a supply of all sizes of orifices on hand and ma frequent changes to keep the gages operating at their most accura range. Some installations have hour-to-hour variations which r beyond the limits of this ratio, and to avoid the cost of consta attendance of an operator a convenient apparatus has been devis for automatically changing from one size of orifice to another witho

FIG. 19 APPARATUS FOR AUTOMATICALLY CHANGING ORIFICES

personal attention. The installation in the pipe line consists o large revolving plate, pierced by several orifices of different si which may be shifted to come in succession concentric with the pi This apparatus, shown in Fig. 19, is actuated by an automatic shif deriving its power from the pressure of gas in the line and co trolled by a diaphragm operated by trip valves so that the disk v be shifted to the next larger orifice when the maximum different reaches a predetermined value, say 40 in., and will be shifted to t next smaller orifice when the differential reaches a minimum p determined value, say 4 or 5 in. This shift makes a sudden and u mistakable jump in the chart, which is readily interpreted. Wheth or not it will be necessary to use a third pen on the chart which w trace a line indicating which orifice is in service will be determin by actual operation.

DIFFICULTIES OF ADMINISTRATION

55 The more recent phase of the measurement problem shows us that the administration of meters should be treated in the same spirit as the technical problem of developing the apparatus. The meter records are sensitive to the slightest disturbances of operating conditions, and indicate troubles other than measurement and difficulties which would otherwise go unsuspected for some time.

56 We have made it the duty of trained technical men to inspect constantly all arriving charts to get early indication of wells suddenly producing unusual quantities of fluid, sand or salt, for

FIG. 20 PULSATION EQUALIZER

freezing up branch or main lines; for the accumulation of water or fluid at low places in the line; for the development of unsuspected leaks or breaks, and even for irregularities in compressor station operations, which would otherwise not be made evident to the management. All these conditions, in addition to their other bad effects, disturb the accuracy of measurement whether by orifice or any other form of meter, but they are especially evident when irregularities of measurements are shown on differential charts sensitive to slight changes in velocity.

PULSATION OF FLOW

57 One problem at present unsolved is obtaining correct measurement with a pulsating flow. This is particularly true where the pulsations are rhythmic, as in the vicinity of compressor stations

with reciprocating compressor pistons. An orifice meter early in stalled at such location failed to check with the station or with meter some 17 miles away on the same line. In endeavoring to locate th difficulty, a series of recording gages was installed, both with an without devices for damping out the pulsations in the lines leadin to the gages. There were finally connected in parallel a spring re cording gage, a mercury float gage of the type originally installe mentioned in Par. 27, a Ballard recording gage as described in Par. 5 and a water U-tube. These gages were all calibrated in unison, an agreed very well under condition of steady flow. When the com pressor station was started, the gages took widely varying position some dropped down to half their former reading despite the increase flow, one took a negative reading as though the flow were reverse and the water column took a wholly indeterminate condition o churned foam; some of the gages moved about in an erratic way an others gave steady indications, but wholly unrelated to the quantit of gas. A proportional meter installed in tandem at this point gave over a period of months, a record erratic and irreconcilable as com pared with pump station displacement, line flow formula or meter 17 miles away operating on the same gas with steady flow.

58 Similar disturbances in the accuracy of the record are oc casioned by irregular pulsations resulting from the action of fluid in the line. Disturbances are particularly serious when occasioned by irregular or imperfect action of automatic pressure regulators in th vicinity of the meter. After many unsuccessful attempts to solv the problem of measuring under these conditions, our efforts are now bent on eliminating pulsation. One attempt is shown in Fig. 20 where a device was installed ahead of a compressor station with th idea of dividing the gas into about 20 different streams and mak each stream traverse a path of different length so that the wav motion from different parts of the cycle would be made to interfer at the point where the gas was again brought to a common line This was almost successful, and it is believed that by a little furthe calculation and change of arrangement to secure more perfect in terference a measurement at this point may be secured.

59 Pulsations due to fluid and imperfect regulators are obvi ously questions of simple correction, by separators, drips and me chanical repairs.

OMISSIONS

60 There are many phases of the problem of measurement which would be interesting to discuss if space permitted. Such are: *a* s

new type of flow prover for proportional meters, using a single hole instead of the multihole flow meter commonly used; *b* organizing and administering a chart calculating department, proof against personal errors in calculations; *c* relative accuracy of chart integrating apparatus and visual inspection; *d* proper time involved to use in computing charts; *e* integrating dial orifice meter suggested in isolated sales where daily inspection is impracticable.

PUBLIC CONFIDENCE

61 The most vital factor, next to technical accuracy, is securing and maintaining the confidence of the "other party to the transaction," in the measurement, especially when obtained by apparatus unfamiliar to him. In the first place, confidence in the accuracy of the gas company's meter measurements in general has not been entirely popular, customary or contagious. Added to that initial handicap is the lack of confidence of the company itself displayed by abandoning a standard method of measurement and adopting a new one. It requires great tact, candor and liberality to develop new confidence from a non-technical public. Our strongest ally has been the fact that it has become generally known that we were working on this problem two years before we actually put it into commercial effect, and many of those with whom we do business realize that the work being done was gone into very carefully. Our next strongest factor is that we have put the same process of measurement simultaneously in use in our purchase and sales meters, and the exact agreement between the constants and apparatus used in purchase and sales meters are open to the inspection and verification of any interested party.

CONCLUSION

62 In conclusion the writer wishes to acknowledge the loyal, able and enthusiastic service of the technical staff entrusted with carrying out these experiments, particularly Mr. E. O. Hickstein, Jun. Am.Soc.M.E.

63 Much of the apparatus referred to has been suggested by members of this staff, as follows: *a* Fundamental concentric U-tube recording gage, Mr. H. O. Ballard; *b* Gravity apparatus and safety device for recording differential gage, Mr. W. C. Baxter; *c* Automatic orifice shift, Mr. J. P. Fisher, Mem. Am.Soc.M.E.

64 Many incidental developments and improvements should be attributed to others who took important part in the work. Acknowl-

edgment is especially due to Prof. P. F. Walker, Mem. Am.Soc.M. for very hearty and enthusiastic coöperation in reviewing data a visiting and inspecting experimental conditions. Further acknow edgment is also due to the many excellent papers that have a peared on measurement of natural gas referred to at various poin through the paper.

65 Particular acknowledgment is due to Mr. Alfred J. Diesch who has supported the work of arriving at a more sound basis measurement. This support has not only consisted in generous approving the considerable cost of the work but also in taking enlightened and detailed interest in the progress throughout, a adopting a broad policy that the truth in regard to measureme whether favorable or unfavorable from a financial standpoint, shou be adopted without coercion or pressure.

DISCUSSION

W. M. Welch[1] (written). In the natural gas fields there is wide divergence in the constants or coefficients used in measuri gas. At the present time, engineers throughout the country working independently on these problems, such as Weymouth a Cooper in Pennsylvania and West Virginia, Wyer and Biddison Ohio, Fisher in Oklahoma, Paine and Moeller in California, a numerous others. In addition to these, the Bureau of Standards Washington is conducting experiments, and the Bureau of Mir under Burrell, in Pittsburgh is also conducting experiments on Boy Law and general work in connection with the measurement gravity of gases. Also, the state universities of Ohio, Kansas, W consin and elsewhere are working in connection with these subje The work being done by these various investigators leads to the us different coefficients, and where a money consideration result from the purchase or sale of gas is dependent upon the coefficie the matter becomes very important from a commercial standpoi and much confusion and possible litigation may result from the settled condition and uncertainty. It would seem that some eff should be made by some body such as this Society, working conjunction with the Bureau of Mines, the Bureau of Standards, state universities, and others, to correlate the work and establ uniformity in the constants or coefficients, in the types of appa

[1] Mgr., Tidal Gasoline Co.. Tulsa, Okla.

tus to be used in the determination of specific gravity and measurement of gases, and in differential pressure-recording instruments.

D. M. Hill (written). I am more familiar with the type of orifice meter consisting of a plate ⅛ in. thick with flat edges, and with pressure connections made at the flange 1 in. either side of the plate. The static-pressure connection is made from the downstream side of the orifice.

From very careful tests of this type of meter with standardized pitot tubes, it has been established that for differentials up to 100 in. of water the orifice coefficient is constant. I believe, therefore, that the variation of the coefficient with increasing differential, of which Mr. Fisher speaks, is due to the location of the pressure connections at 2½ times the pipe diameter on the inlet and 8 times the pipe diameter on the outlet side of the orifice. By using flange connections a larger differential for a given flow is obtained. This helps somewhat in low flows, where the chances for error are great, due to gage inaccuracies.

I fail to see how the correction to Boyle's Law will bring the observational points closer to the curve in Fig. 16, inasmuch as both the Joplin and Erie tests were performed under the same pressure conditions, and the points are about evenly distributed above and below the curve.

The device which Mr. Fisher has described for automatically changing from one size of orifice to another without personal attention is most ingenious. I would like a little more information as to how the sticking of the plate against the gasket and the friction against the flange were overcome without permitting serious leakage of gas past the plate. In practice, it has been found that after an orifice plate has been in service for some time, it frequently sticks to the gasket, even with the flange spread apart with jackscrews.

W. F. M. Goss. It is evident that our practice in dealing with natural gas is rapidly improving. I recall a pipe line in Indiana which, in the declining days of the gas field in that state, was found to be delivering a million feet of gas per day as shown by consumers' meters, while the compressor displacement in the field, 45 miles away, showed that six million feet of gas were being delivered into the line. The line was reported by the field inspectors to be in good condition.

Measurements taken along the line revealed a steady decline in the quantity of gas passing different sections of the pipe. Differences were accounted for only on the assumption of leakage. The condition of this pipe line was not exceptional. When the supply of gas was abundant and the amount delivered sufficient for consumption, there was little concern for what leaked out on the way.

All this, it will be noted, is in striking contrast with the attitude displayed by Mr. Fisher's excellent paper.

P. F. WALKER. The conditions in the mid-continental field are not so dissimilar from those which Dean Goss has just cited, except for certain differences in the character of the gas, or, rather, in the character of the substances, mainly vapors and sometimes actual liquids, carried with the gas, and the advancement of the art since the Indiana days. About five years ago I was making investigations of the pipe-line conditions for the Kansas Natural Gas Company, the company supplying Kansas City and other towns in Eastern Kansas and Western Missouri. There were three compressor stations on the line, one receiving the gas from the fields, the second 40 miles away, a step-up station, and the third, 40 miles further, a second step-up station. The volume of gas flowing in the line measured by the volume of piston displacement, fell off as the gas passed successive stations, but not, perhaps, more than 8 or 10 per cent each time. The loss, of course, in the main, was due to leak or to uncharted small users along the lines.

The mid-continental gas carries a large amount of matter foreign to gas and gas measurement. At places it is absolutely necessary to take out some of the lighter oils and gasolines in order to avoid danger. A very good grade of gasoline is being manufactured at points along the company's lines.

The trouble with the Thomas meter is that its action is based upon the amount of heat required to heat a given amount of the substance during a given rise in temperature. With the impure gas the specific heat is an extremely variable quantity, and even though the specific heat were known, many of the heavier condensable vapors would probably have been removed before the gas reached its destination, so that the measurement would not be the one required. Moreover, the rock and sand in the gas would destroy so delicate an instrument as the Thomas meter.

THE AUTHOR. The instance of which Professor Goss speaks where six million feet of gas was put into the line to maintain one million feet 45 miles away, was, of course, a very serious case, but by no means unique. Similar conditions have gone on unsuspected in many fields during the early history of the business, for lack of any reliable means of checking gas at the input end.

Mr. Hill refers to a statement (and to Fig. 14) in the original paper as to the variable coefficients at different values of the differential. This statement should perhaps have been modified by the statement that the total variation from a constant of these differentials at the various values was relatively small and has never been

Fig. 21 Present Standard Setting for Orifice Meter

taken into consideration as a commercial factor, but merely in some of the theoretical work it was considered that a law might be found governing these slight variations.

In plotting distances of the outlet pressure tap from he orifice flange as abscissæ against the differentials as ordinates for various constant rates of flow, the writer has found that the recovery of static pressure beyond the orifice with the decrease of velocity, while practically negligible at 1 in. differential, becomes noticeable at higher differentials, and that with the point of observation spaced at 8 diameters there is full recovery and practically a horizontal line on differentials up to and exceeding 50 in., which is the highest used in our work. The point of observation on the inlet side was 2½ diameters distant from the orifice flange.

In answer to Mr. Hill's further query, no gaskets are used in the orifice-shifting device, but leakage of gas is prevented to an absolutely satisfactory degree of "tightness" by the pressure of accurately ground brass surfaces against accurately ground steel surfaces, the surfaces being kept in contact by springs. The average differential that is used as a maximum, at which the disks are shifted, is about 40 in., at which pressure the leakage through such ground joints is a negligible factor compared with the quantities of gas flowing.

One advantage of using these points of observation (2½ and 8 diameters) of the differential is to give orifices which can be used satisfactorily with a wider range between maximum and minimum flow quantities. It also does away with the possibility of the serious effect of violent eddy currents at the point of expansion beyond the orifice where these changes of energy from dynamic to static pressure are taking place.

I would like to further acknowledge Mr. Hill's criticism of the paragraph dealing with Boyle's Law and Fig. 16, as I find that the points referred to in that paragraph, which were largely observations at Wann at high pressures, have been in some way omitted from the sheet from which the curve was drawn.

Fig. 21 shows what we now consider our standard setting, in which fittings are largely eliminated by the use of oxy-acetylene welding, and the meter is raised out of the ground for convenience in handling and changing disks and eliminating water or fluid from the line adjacent to the meter disks.

No. 1537

DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

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and

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The law that "the volume of a gas at constant temperature varies inversely as its absolute pressure, or that the product of the volume and absolute pressure of a gas is constant," discovered by Boyle in 1660, is the starting point of natural-gas compressor and high-pressure natural-gas measuring calculations. The extent of the commercial application of the law becomes evident when we consider that in the United States over 600,000,000,000 cu. ft. of gas per annum are now measured under high-pressure conditions, and that there are in this country over 200 natural-gas-compressing stations, aggregating more than 315,000 h.p. of compressor capacity and compressing more than 85 per cent of all the gas used.

2 The universal custom in the natural-gas business has been to assume that the gas obeys Boyle's law. That the law is not followed rigorously by natural gas, or by any other commercial gas, is evident, however, from the following: Boyle, in his original memoirs in 1661, called attention to the fact that undoubtedly there would be deviations from the law. In the nineteenth century, experiments by Regnault, Amagat and Cailletet showed marked deviations.[2] Professor Walker, in 1912, was probably the first to call

[1] Prof. of Physics, Ohio State Univ.

[2] Meyer's Kinetic Theory of Gases, 1877; Preston's Theory of Heat, 1894; Vol. 1, Wüllner's Lehrbuch der Experimental Physik, 1895; Harper's Scientific Memoirs, Laws of Gases, Edited by Carl Barus, 1899.

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attention to the deviation of natural gas from Boyle's law on account of the then known deviations of methane gas, the principal constituent of natural gas,[1] and again in 1914, in his paper on the Physical Laws of Methane Gas.[2]

3 All the data published heretofore referred to the so-called permanent gases. The data given in the present paper were obtained by making actual volume measurements of representative natural gases, from atmospheric pressure up to 48 atmos. The tests were made in the Physics Laboratory of the Ohio State University with the apparatus shown in Figs. 1 and 3. *A, B, C, E* and *I* were made by the Geneva Society.

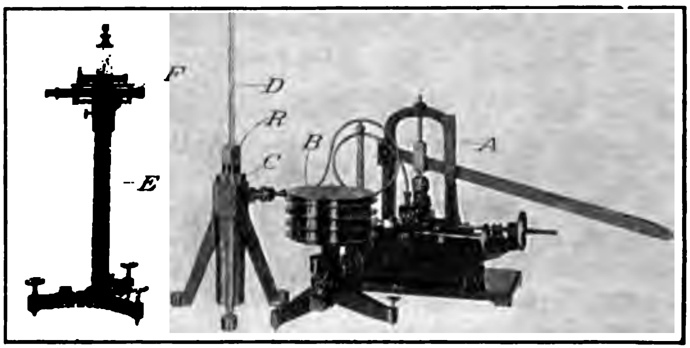


FIG. 1 ASSEMBLY OF APPARATUS

4 Referring to Figs. 1 and 2, *A* is an oil pump, *B* a gage tester connected to the oil pump so as to indicate the pressure developed by *A*. *C* is a Cailletet piece (named for the French scientist who designed it) with calibrated capillary glass tube *D*. The inside construction of *C* is shown in Fig. 2. *E* is a cathetometer with moving telescope *F* used for reading the height of the mercury column in *D*.

5 In Fig. 3, *G* is a Gaede box vacuum pump, driven by the electric motor *H*. *I* is a manometer with cocks *J*, *K* and *L*, and a small mercury gage underneath the glass, which indicates the degree of vacuum obtained. *M* is a bent glass tube with glass cock *N* and short rubber hose *O*. The hose *O* is tightly wired to *M* to prevent

[1] Proc. Natural Gas Assoc. of America, vol. 4, p. 172.

[2] Trans. Am. Soc. M.E., vol. 36, p. 781.

its slipping off. *P* is a standard gas tank used for shipping the gas to the laboratory.

6 The capillary glass tube *D* had an inside diameter of 3 mm. and was sealed off squarely at the top and held in a vertical position in collar *S* by means of a stuffing box and gland on the inner side of the collar, as shown in Fig. 2. The tube was selected from a large number, and was calibrated in a vertical position by forcing the column of mercury up into the tube and then separating a slug of mercury about 1 in. long from the main column by an air cushion, raising this short slug of mercury through the entire working length

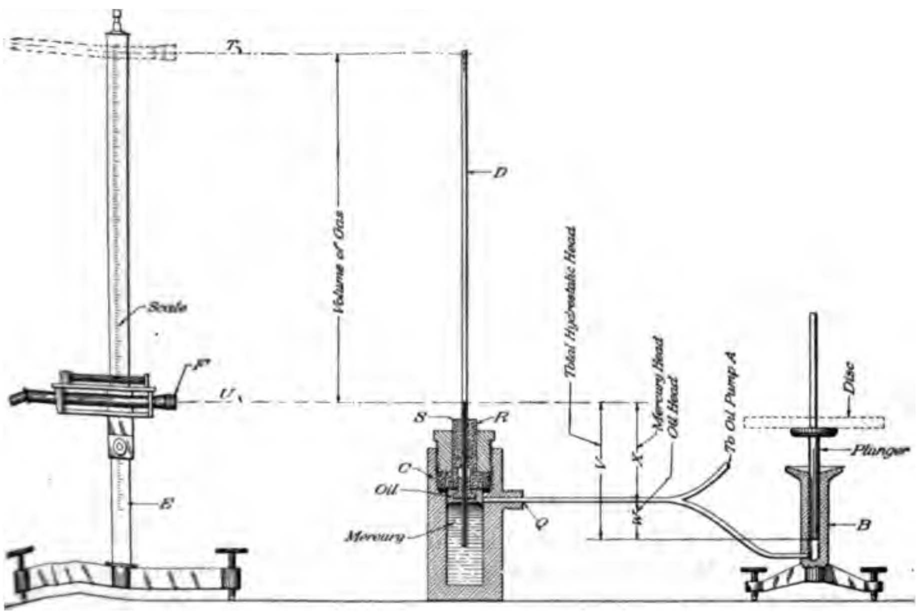


Fig. 2 Diagram of Apparatus

of the tube and noting its variations in length by means of the telescope *F* on the cathetometer *E*. The variation in diameter of the tube was so small that volume could safely be considered to be directly proportional to length. The tube *D* was originally surrounded by a water bath, but on account of the practically constant temperature conditions prevailing in the room this was abandoned. The readings for volume contractions were always made with a rising meniscus in *D*.

7 The order of conducting the tests was as follows: The capillary tube *D* was connected as shown in Fig. 3, with cocks *J*, *L* and *N* open, and *K* closed. By means of *G* the capillary tube *D* was then evacuated and the degree of vacuum noted by the small mercury column in *I*. In operating, the vacuum was pulled down to 1 mm. of mercury; cock *J* was then closed and *K* opened, allowing *D* to be filled from the natural-gas tank *P*. This operation was repeated at least four times, to wash out the inside of *D* thoroughly and leave it free from every trace of air. In the last operation, cock *N* was closed and *M* disconnected from the rubber hose leading to *L*. *D* and *M* were then held in a vertical position as shown in Fig. 4, and

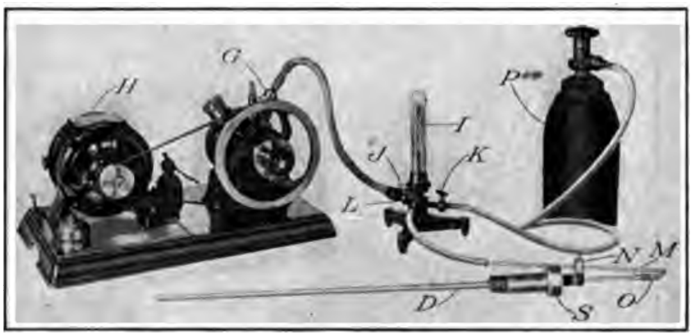


FIG. 3 METHOD OF FILLING CAPILLARY TUBE

inserted in the mercury inside of *C* so that the mercury ran out of the opening *Q*. When *O* and the lower end of *D* were well under the surface of the mercury in *C*, *D* was held stationary by the one hand and *M* pushed down with the other hand, in that way slipping the rubber *O* from off *D* underneath the mercury surface in *C*, thereby entraining the gas in *D*. The nut *R* was then screwed down loosely on *S*, connections made to the oil pump as shown in Fig. 1, and the pump worked slowly so as to force the air out around *R*. After the air was all driven out, *R* was then screwed down tightly onto *S*.

8 The maximum pressure to be used on *B* was now applied so as to produce the maximum strain on *D*, and by means of telescope *F* and cathetometer *E* the top position of the inside of *D* was determined by the line *T*, as shown in Fig. 2. The pressure in *D* was now released and the smallest weight that would make the mercury visible in *D* was applied to *B*, as indicated by *U* in Fig. 2. Successive

increments of weight on *B* now produced corresponding contractions in the gas volume in *D*. The gas in *D* was also allowed to expand, and several readings taken with each gas checked with those taken on compression. The volume changes were accurately observed by means of telescope *F* and the centimeter scale on cathetometer *E*.

9 All readings were made in metric units and pressures reckoned in standard atmospheres of 76 cm. of mercury, or 14.7 lb. per sq. in.

10 The plunger on the gage tester was of such diameter that one kilogram of weight was equal to one standard atmosphere of

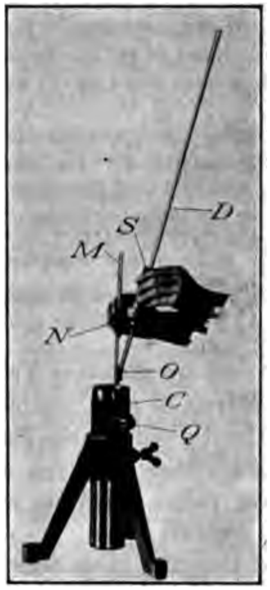


FIG. 4 METHOD OF INSERTING CAPILLARY TUBE IN MERCURY SEAL

pressure. The plunger itself weighed 0.354 kg. and therefore represented 0.354 atmos. In addition to the disks that belonged to the gage tester *B*, standard laboratory weights were used for getting the smaller increments of pressure.

11 The total hydrostatic head indicated by *V* in Fig. 2 is the sum of a variable represented by mercury column *X* and a constant represented by the oil column *W*. The length of this oil column *W* was kept at 20 cm. in all the tests. The specific gravity of the oil, as determined by a Mohr balance, was found to be 0.866 and there-

fore equal to 17.32 cm. of water, which in turn was equal to $17.32 \div 76 \times 13.6 = 0.016$ standard atmos.

12 The readings were classified as follows:

		Typical Values
	Barometric pressure in standard atmospheres	0.981
a =	Weight on plunger $+ 0.354$	1.176
b =	Reading of upper end of gas column in D as T	42.31
c =	Reading of lower end of gas column in D as U	1.00
d =	Length of gas column = relative volume = $b - c$	41.31
e =	Height of mercury in C	-8.00
f =	Hydrostatic head of mercury $= X = c + e$	9.00
g =	f in standard atmospheres $= f \div 76$	0.12
W =	Hydrostatic head of oil in atmospheres	0.016
V =	Total hydrostatic head of oil and mercury $= g + W$	0.136
h =	Total net absolute pressure on gas = barometric pressure $+ a - V$	2.021
PV =	Product of volume and pressure $= d \times h$	83.50
	Deviation in per cent corresponding to various pressures = $100 \times$ "PV" value of lowest pressure divided by "PV" value corresponding to any other pressure, minus 100; $= (100\, P_0V_0/P_xV_x) - 100$.	

13 The results obtained are shown in Tables 1 to 8 and Figs. 5 to 12. The chemical analysis of each sample of gas was made by George A. Burrell, of the United States Bureau of Mines, Pittsburgh, Pa. The specific gravities are given with reference to air as unity. The gross heating values given are computed on a basis of 32 deg. fahr. and 76 cm. pressure.

14 It may be noted that the gases which contain a large percentage of C_2H_6 (wet gases) show larger deviations than the dry gases. The explanation lies in the fact that we are dealing with a physical mixture of gases having two principal constituents, CH_4, methane and C_2H_6, ethane. These have different critical temperatures, one of which lies below, and the other above, the temperature at which our experiments were made.

15 CH_4 has a critical temperature of -140 deg. fahr.; consequently no pressure, however great, could produce liquefaction at the operating temperature, approximately 68 deg. fahr., which prevailed during our experiments. On the other hand, C_2H_6 has a critical temperature of 93 deg. fahr. It follows, therefore, that as the volume of the mixture of gases is diminished, the C_2H_6 con-

stituent approaches the condition of a saturated vapor. When saturation is reached further diminution in volume will produce no change in the pressure exerted by this constituent, but a liquefaction. In case a mixture contains initially a large amount of C_2H_6, this saturated condition will be reached at a lower stage of compression. The fact that further diminution of volume will not thereafter alter the pressure produced by this constituent indicates why a wet gas at high pressures shows such a marked deviation when compared with the pressure-volume product computed for the lower pressures.

16 Our observations tend to confirm this explanation. There was at no time any visible collection of liquid within the tube, nor could this be expected under the conditions of the experiment. The initial volume of the gas was so small and the surface of the capillary so large, relatively, that we could not expect visible droplets to form.

17 We noted in the case of wet gases that when a particular stage in compression was reached (peculiar to the gas operated upon) there was a curious unsteadiness and agitation of the mercury meniscus. We observed this carefully in several cases, and found that the range in pressure and volume through which this occurred was very limited. It was probably due to a deposit of liquid taking place on the mercury meniscus, resulting in a change in surface tension. We could confine this region where agitation occurred to variations of pressure of about one-fifth of an atmosphere. Below and above this pressure the meniscus was steady.

18 We call attention to the bearing of Par. 14, 15 and 16 to gas handling problems. A gas containing a high percentage of C_2H_6, i.e., a wet gas, will show a very different correction curve than the same gas after having been run through two or three compressors and cooling tanks where presumably the condensate would be removed. On the other hand, dry gases would show small differences before and after compression.

19 In conclusion, attention is called to the following facts:

These tests were all made at room temperatures. We believe it would be desirable to have further tests made at higher and lower temperatures.

With the two gases given in Tables 3 and 5, and with another piece of apparatus, we made tests between absolute pressures of 0.978 atmos. and 2.337 atmos. and found no perceptible deviation from the law.

TABLE 1 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

West Virginia natural gas, sampled at Sugar Grove, Ohio, from the Ohio Fuel Supply Company's 18-in. line at Ravenswood, West Virginia. Graphical data for this are shown in Fig. 5.

P Absolute Pressure in Standard Atmospheres	V Relative Volume Observed	Boyle's Law Volume	PV Product of P and V as "Constant"	Per Cent Deviation
2.029	41.31	41.31	83.82	0.00
2.117	39.71	39.59	84.06	0.997[1]
2.187	38.14	38.32	83.41	0.49
2.268	36.67	36.96	83.17	0.78
2.350	35.34	35.67	83.05	0.92
2.434	34.08	34.43	82.95	1.05
2.518	32.90	33.29	82.84	1.18
2.604	31.77	32.19	82.73	1.32
2.690	30.74	31.16	82.69	1.36
2.776	29.71	30.19	82.47	1.64
2.863	28.71	29.27	82.20	1.97
2.951	27.81	28.40	82.07	2.13
3.040	26.93	27.57	81.87	2.38
3.130	26.16	26.78	81.88	2.37
3.219	25.38	26.04	81.70	2.59
3.310	24.60	25.32	81.72	2.57
3.401	24.01	24.64	81.66	2.65
3.493	23.35	23.99	81.56	2.77
3.584	22.71	23.39	81.39	2.99
3.677	22.14	22.79	81.41	2.96
4.137	19.58	20.26	81.00	3.48
4.610	17.54	18.18	80.86	3.66
5.088	15.87	16.47	80.75	3.80
5.570	14.45	15.05	80.49	4.13
5.838	13.72	14.36	80.10	4.64
6.811	11.71	12.31	79.76	5.00
7.792	10.21	10.76	79.56	5.35
8.777	9.06	9.55	79.52	5.407
9.764	8.11	8.58	79.19	5.85
10.754	7.36	7.79	79.15	5.90
11.746	6.70	7.14	78.70	6.51
12.738	6.16	6.58	78.47	6.82
13.732	5.68	6.10	78.00	7.46
14.727	5.29	5.69	77.91	7.58
15.722	4.89	5.33	76.88	9.00
16.718	4.59	5.01	76.74	9.22
17.714	4.31	4.73	76.35	9.78
18.711	4.11	4.48	76.90	8.99
19.709	3.89	4.25	76.67	9.32
20.706	3.69	4.04	76.41	9.69
25.695	2.88	3.26	74.00	13.27
30.689	2.41	2.73	73.96	13.33
35.684	2.04	2.35	72.80	15.14
40.681	1.77	2.06	72.01	16.40
48.677	1.49	1.72	72.53	15.56

August 13, 1915. Barometer 29.4 in. Temperature at start 75.2 deg. fahr., temperature at close 74.4 deg. fahr.

[1] The peculiarities of this value are due either to an error in observation, or an unaccounted for behavior of the gas at that pressure.

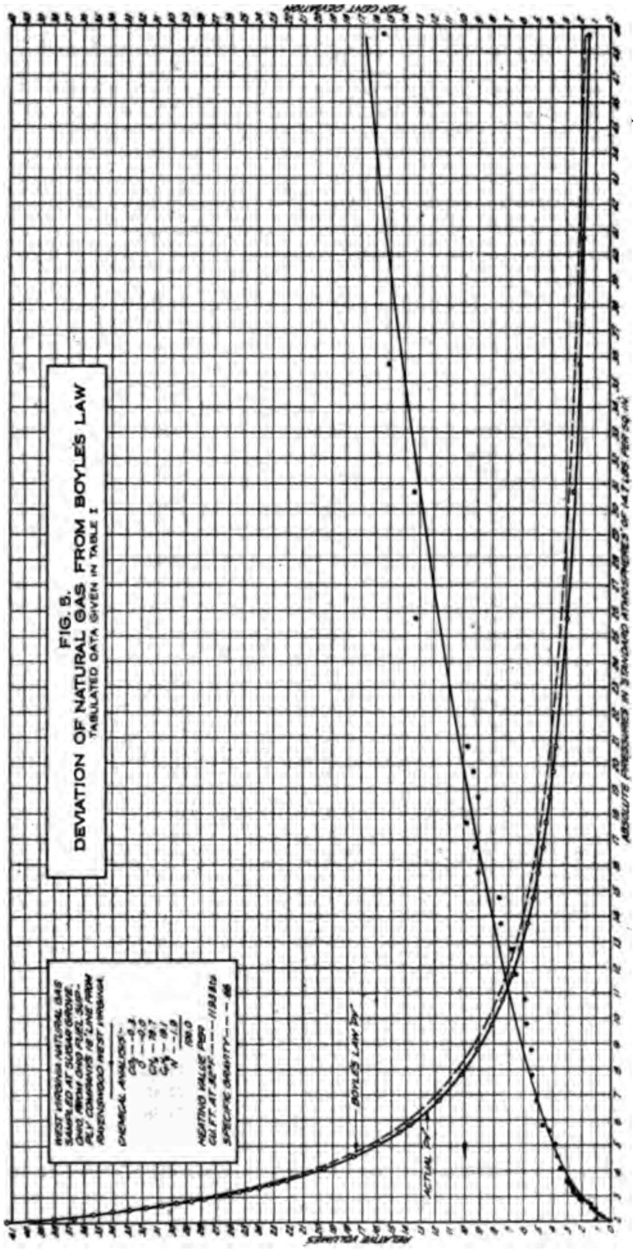


FIG. 5 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 2 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

WEST VIRGINIA NATURAL GAS SAMPLED AT SUGAR GROVE, OHIO, FROM THE CONNECTING GAS COMPANY'S TWO 16-IN. MAINS TO WEST VIRGINIA. GRAPHICAL DATA FOR THIS ARE SHOWN IN FIG. 6.

P Absolute Pressure in Standard Atmospheres	*V* Relative Volume Observed	Boyle's Law Volume	*PV* Product of *P* and *V* as "Constant"	Per Cent Deviation
2.020	41.35	41.35	83.53	0.00
2.098	39.70	39.81	83.29	0.29
2.179	38.23	38.33	83.30	0.28
2.260	36.77	36.96	83.10	0.52
2.342	35.41	35.66	82.93	0.72
2.425	34.15	34.44	82.81	0.87
2.510	32.98	33.28	82.78	0.906
2.595	31.85	32.19	82.65	1.06
2.681	30.78	31.15	82.52	1.22
2.768	29.78	30.17	82.43	1.33
2.943	27.93	28.38	82.20	1.62
3.121	26.28	26.76	82.22	1.59
3.302	24.78	25.29	81.82	2.09
3.485	23.47	23.97	81.79	2.13
3.669	22.27	22.77	81.71	2.23
4.135	19.72	20.20	81.54	2.44
4.608	17.67	18.13	81.42	2.59
5.086	15.99	16.42	81.33	2.705
5.842	13.90	14.83	81.29	2.75
6.816	11.85	12.14	80.77	3.66
7.795	10.30	10.74	80.29	4.03
8.779	9.12	9.54	80.06	4.58
9.767	8.18	8.47	79.89	4.80
10.756	7.36	7.78	79.16	5.52
13.237	5.92	6.25	78.36	6.50
15.725	5.00	5.32	78.62	6.49
18.216	4.29	4.60	78.15	6.88
20.709	3.73	4.03	77.24	8.40
25.707	2.93	3.25	75.32	10.90
30.692	2.45	2.73	75.20	11.08
35.687	2.06	2.35	73.62	13.46
40.683	1.79	2.06	72.82	14.707
48.679	1.47	1.72	71.56	16.73

September 3, 1915. Barometer 29.6 in. Temperature at start 66.2 deg. fahr., temperature at close 66.2 deg. fahr.

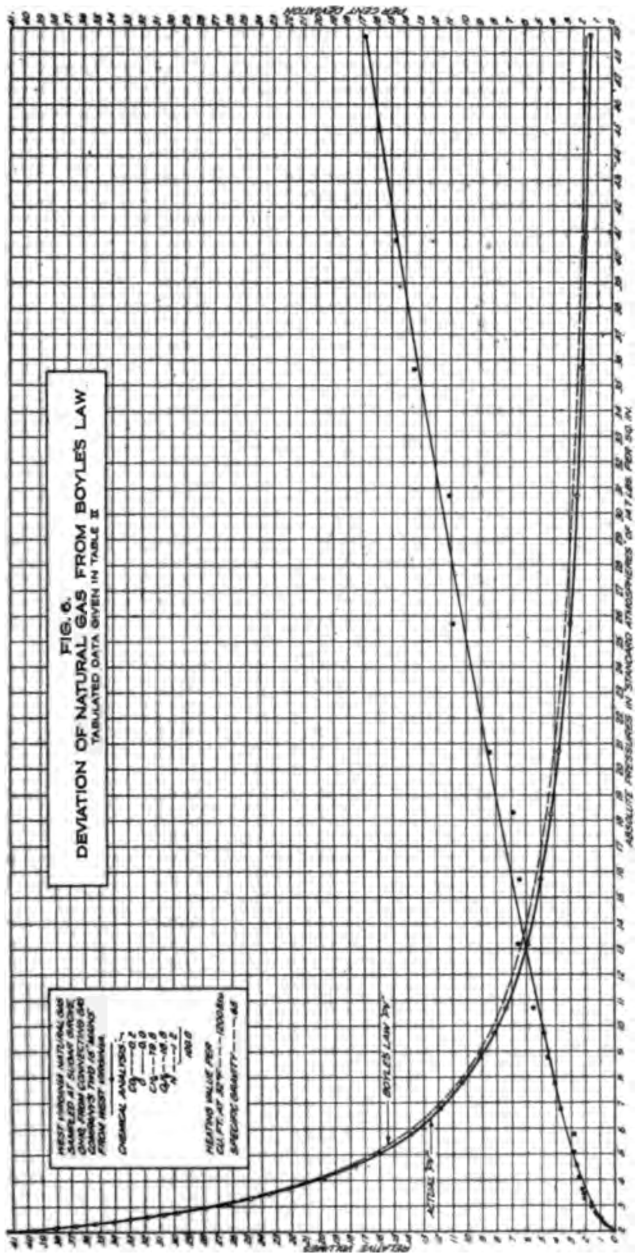


FIG. 6 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 3 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

PENNSYLVANIA "DRY" NATURAL GAS FROM TRI-COUNTY GAS COMPANY, ROYSTONE, PA.
GRAPHICAL DATA FOR THIS ARE SHOWN IN FIG. 7

P Absolute Pressure in Standard Atmospheres	*V* Relative Volume Observed	Boyle's Law Volume	*PV* Product of *P* and *V* as "Constant"	Per Cent Deviation
2.024	41.62	41.62	84.24	0.00
2.103	39.97	40.06	84.06	0.21
2.183	38.49	38.59	84.02	0.26
2.264	37.04	37.21	83.86	0.45
2.345	35.58	35.92	83.44	0.96
2.429	34.37	34.68	83.48	0.91
2.513	33.18	33.52	83.38	1.03
2.598	32.04	32.42	83.24	1.201
2.684	30.98	31.40	83.15	1.31
2.771	29.98	30.40	83.07	1.406
2.947	28.11	28.62	82.84	1.69
3.124	26.39	26.96	82.44	2.18
3.305	24.93	25.50	82.39	2.24
3.487	23.59	24.16	82.26	2.407
3.671	22.38	22.95	82.16	2.53
4.137	19.79	20.36	81.87	2.89
4.611	17.77	18.27	81.94	2.806
5.088	16.06	16.56	81.71	3.09
5.570	14.65	15.12	81.00	4.00
5.838	13.99	14.43	81.67	3.14
6.822	11.97	12.35	81.66	3.15
7.792	10.42	10.81	81.19	3.75
8.776	9.22	9.60	80.91	4.12
9.764	8.29	8.63	80.94	4.08
10.753	7.48	7.83	80.43	4.73
11.744	6.82	7.17	80.09	5.18
12.737	6.27	6.61	79.86	5.48
13.731	5.80	6.13	79.64	5.78
14.726	5.40	5.72	79.52	5.98
15.721	5.05	5.36	79.39	6.109
20.705	3.82	4.07	79.09	6.51
25.695	3.04	3.28	78.11	7.85
30.688	2.52	2.75	77.33	8.94
35.682	2.12	2.36	75.65	11.35
40.679	1.85	2.07	75.26	11.98
48.675	1.56	1.73	75.93	10.94

August 30, 1915. Barometer 29.4 in. Temperature at start 67.2 deg. fahr., temperature at close 65.0 deg. fahr.

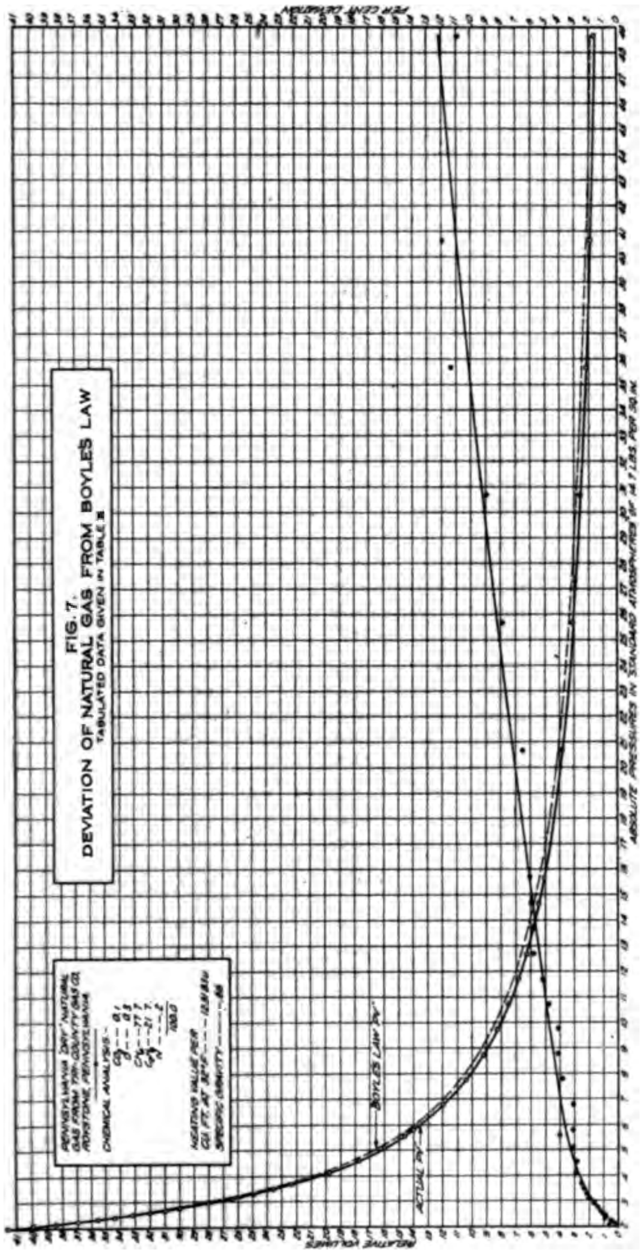


FIG. 7 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 4 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

PENNSYLVANIA "WET" NATURAL GAS FROM THE PENNSYLVANIA GAS COMPANY'S "EAST BRANCH" INTAKE. GRAPHICAL DATA FOR THIS ARE SHOWN IN FIG. 8

P Absolute Pressure in Standard Atmospheres	*V* Relative Volume Observed	Boyle's Law Volume	*PV* Product of *P* and *V* as "Constant"	Per Cent Deviation
1.936	42.03	42.03	81.37	0.00
2.013	40.31	40.42	81.14	0.28
2.092	38.72	38.89	81.00	0.45
2.172	37.21	37.46	80.82	0.68
2.254	35.80	36.10	80.69	0.84
2.336	34.48	34.83	80.55	1.02
2.420	33.26	33.62	80.49	1.09
2.505	32.08	32.48	80.36	1.25
2.590	30.96	31.42	80.19	1.47
2.676	29.92	30.41	80.07	1.62
2.763	28.92	29.45	79.91	1.83
2.940	27.12	27.68	79.73	2.06
3.119	25.52	26.09	79.60	2.22
3.300	24.07	24.66	79.43	2.44
3.482	22.78	23.37	79.32	2.58
3.667	21.60	22.19	79.21	2.73
4.134	19.09	19.68	78.92	3.23
4.607	17.07	17.66	78.64	3.47
5.086	15.42	15.99	78.42	3.75
5.837	13.41	13.94	78.27	3.96
6.811	11.42	11.95	77.78	4.61
7.791	9.91	10.44	77.21	5.39
8.776	8.74	9.27	76.70	6.09
9.764	7.84	8.33	76.55	6.30
10.754	7.06	7.57	75.92	7.18
11.745	6.42	6.81	75.40	7.92
13.732	5.45	5.93	74.83	8.74
15.723	4.72	5.18	74.21	9.65
18.214	4.02	4.47	73.22	11.12
20.706	3.47	3.93	71.85	13.24
25.696	2.70	3.17	69.38	17.28
30.689	2.19	2.65	67.21	21.07
35.685	1.86	2.28	66.37	22.60
40.681	1.57	2.00	63.87	27.40
48.677	1.27	1.67	61.82	31.62

September 2, 1915. Barometer 29.6 in. Temperature at start 68.0 deg. fahr., temperature at close 68.7 deg. fahr.

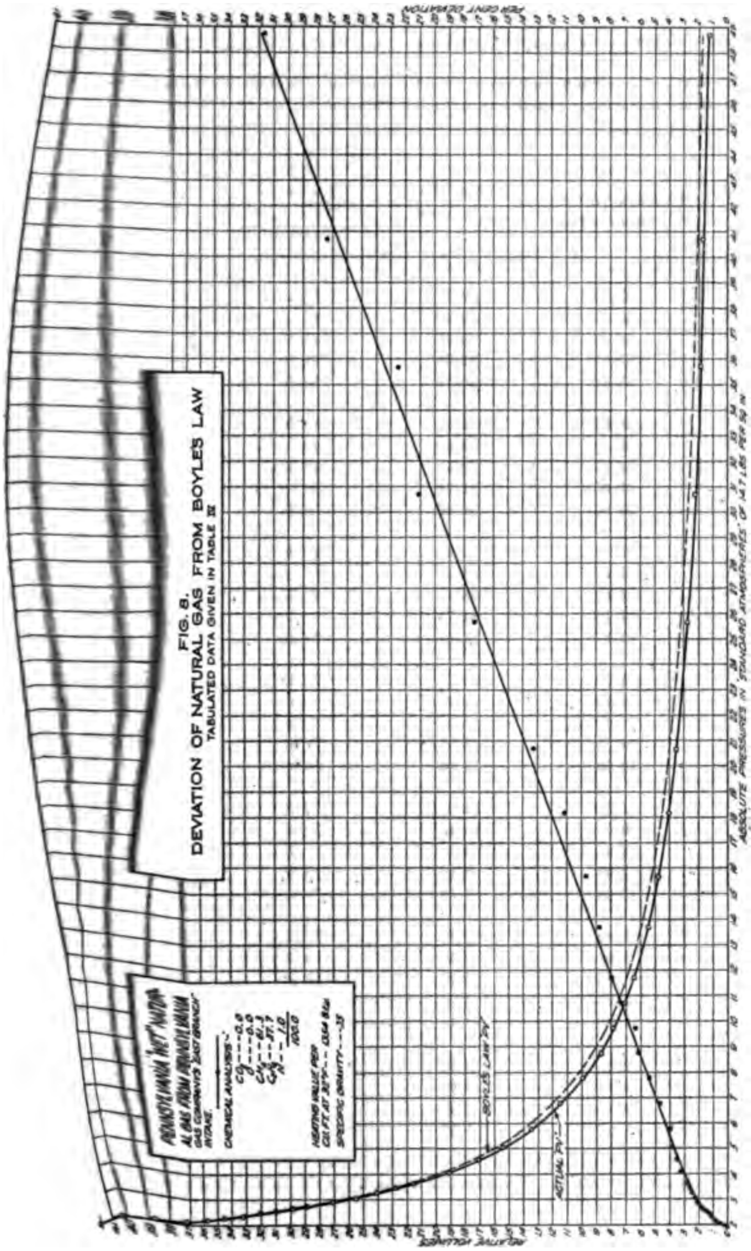


FIG. 8 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 5 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

Pennsylvania natural gas from Equitable Gas Company at Trafford City, practically a pure methane gas. Graphical data for this are shown in Fig. 9

P Absolute Pressure in Standard Atmospheres	*V* Relative Volume Observed	Boyle's Law Volume	*PV* Product of *P* and *V* as "Constant"	Per Cent Deviation
1.936	42.10	42.10	81.51	0.00
2.013	40.41	40.49	81.35	0.19
2.092	38.79	38.96	81.15	0.44
2.173	37.32	37.51	81.10	0.63
2.253	35.82	36.18	80.70	1.04
2.337	34.57	34.81	80.79	0.89
2.421	33.40	33.66	80.86	0.804
2.506	32.17	32.54	80.59	1.02
2.591	31.06	31.46	80.48	1.28
2.676	30.00	30.46	80.28	1.53
2.764	29.05	29.49	80.29	1.52
2.940	27.20	27.72	79.97	1.92
3.119	25.60	26.13	79.84	2.09
3.300	24.18	24.70	79.79	2.15
3.483	22.87	23.40	79.66	2.32
3.667	21.70	22.23	79.57	2.44
4.134	19.20	19.72	79.37	2.69
4.621	17.20	17.64	79.48	2.56
5.086	15.55	16.02	79.09	3.06
5.838	13.53	13.96	78.99	3.19
6.812	11.57	11.96	78.81	3.42
7.792	10.09	10.46	78.62	3.67
8.777	8.94	9.29	78.47	3.87
9.765	7.99	8.35	78.03	4.46
10.755	7.23	7.58	77.76	4.84
13.236	5.83	6.16	77.17	5.62
15.724	4.88	5.18	76.73	6.23
18.215	4.20	4.47	76.50	6.55
20.708	3.67	3.93	76.00	7.25
25.698	2.90	3.17	74.52	9.38
30.691	2.41	2.66	73.97	10.19
35.687	2.10	2.28	74.94	8.77
40.684	1.82	2.00	74.04	10.00
48.679	1.52	1.67	73.99	10.16

September 2, 1915. Barometer 29.6 in. Temperature at start 67.6 deg fahr., temperature at close 67.6 deg. fahr.

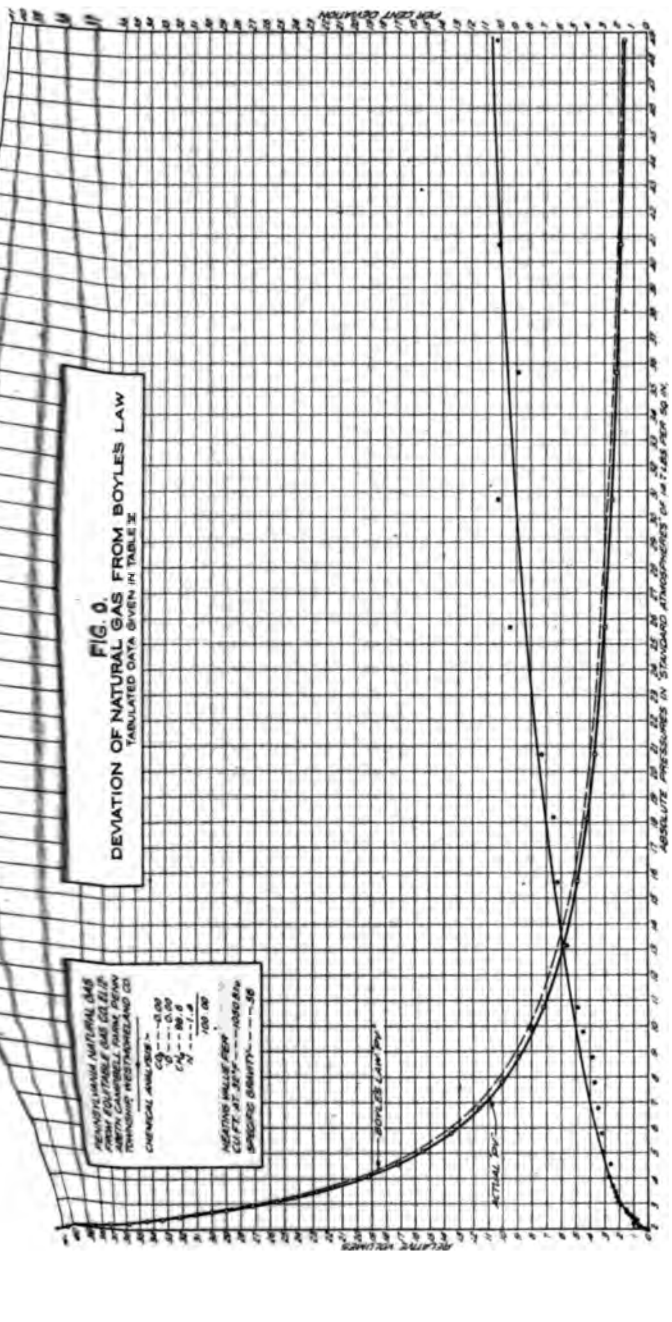


FIG. 9 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 6 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TEXAS NATURAL GAS, FROM LONE STAR GAS COMPANY AT PETROLIA, TEXAS. GRAPHICAL DATA FOR THIS ARE SHOWN IN FIG. 10

P Absolute Pressure in Standard Atmospheres	V Relative Volume Observed	Boyle's Law Volume	PV Product of P and V as "Constant"	Per Cent Deviation
2.018	40.98	40.98	82.70	0.00
2.097	39.39	39.44	82.60	0.12
2.177	37.85	37.99	82.40	0.36
2.258	36.42	36.62	82.24	0.56
2.340	35.09	35.34	82.11	0.72
2.424	33.84	34.12	82.03	0.82
2.508	32.65	32.97	81.89	0.99
2.593	31.52	31.89	81.73	1.19
2.679	30.47	30.87	81.63	1.31
2.767	29.50	29.89	81.63	1.31
2.943	27.66	28.10	81.40	1.59
3.121	26.02	26.50	81.21	1.83
3.302	24.57	25.04	81.13	1.93
3.484	23.22	23.74	80.90	2.22
3.669	22.03	22.54	80.83	2.31
4.135	19.47	20.00	80.51	2.72
4.609	17.49	17.94	80.61	2.59
5.087	15.79	16.26	80.32	2.96
5.568	14.40	14.85	80.18	3.14
6.053	13.24	13.66	80.14	3.19
5.837	13.70	14.17	79.97	3.41
6.810	11.69	12.14	79.61	3.88
7.791	10.21	10.61	79.55	3.95
8.776	9.05	9.42	79.42	4.13
9.763	8.12	8.47	79.28	4.31
10.753	7.32	7.69	78.71	5.07
11.745	6.69	7.04	78.57	5.26
12.738	6.15	6.49	78.34	5.57
13.732	5.72	6.02	78.55	5.28
14.746	5.31	5.61	78.30	5.62
15.722	4.97	5.26	78.14	5.83
16.718	4.69	4.94	78.41	5.47
17.714	4.41	4.67	78.12	5.86
18.712	4.18	4.42	78.22	5.73
19.708	3.94	4.20	77.65	6.503
20.706	3.75	3.99	77.65	6.503
25.696	3.01	3.22	77.34	6.93
30.690	2.54	2.69	77.95	6.09
35.685	2.19	2.32	78.15	5.82
40.682	1.90	2.03	77.32	6.96
48.677	1.59	1.70	77.40	6.85

August 14, 1915. Barometer 29.45 in. Temperature at start 73.4 deg. fahr., temperature at clo[se] 4.3 deg. fahr.

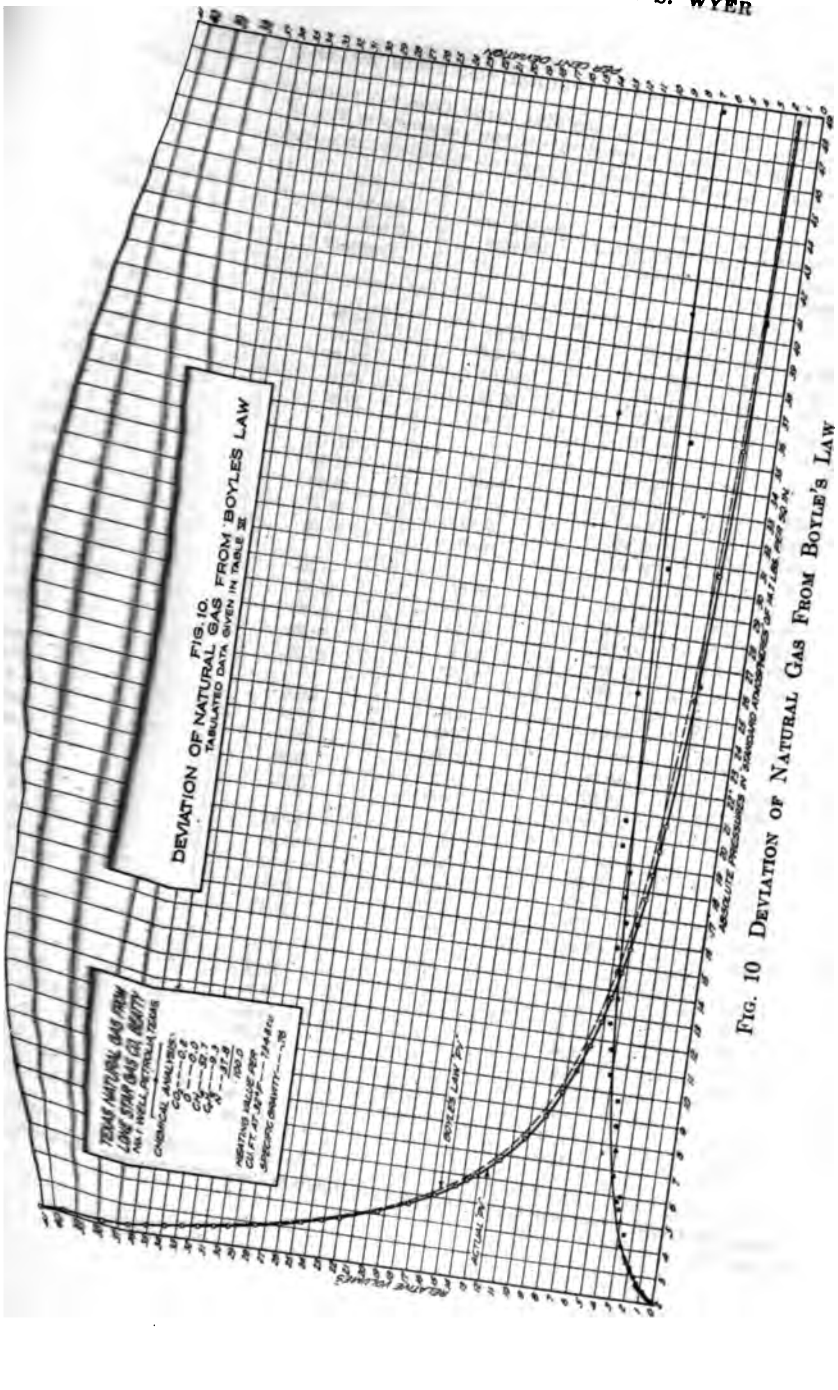


FIG. 10 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 7 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

OHIO NATURAL GAS TAKEN FROM THE OHIO FUEL SUPPLY COMPANY'S INTAKE LINES AT SUGAR GROVE, OHIO. GRAPHICAL DATA OF THIS ARE SHOWN IN FIG. 11

P Absolute Pressure in Standard Atmospheres	*V* Relative Volume Observed	Boyle's Law Volume	*PV* Product of *P* and *V* as "Constant"	Per Cent Deviation
2.022	41.07	41.07	83.04	0.00
2.101	39.42	39.53	82.82	0.27
2.180	37.90	38.09	82.62	0.52
2.261	36.45	36.72	82.41	0.76
2.344	35.10	35.42	82.27	0.93
2.427	33.83	34.21	82.11	1.13
2.511	32.65	33.07	81.98	1.29
2.596	31.52	31.98	81.83	1.49
2.683	30.48	30.95	81.78	1.54
2.769	29.45	29.99	81.55	1.82
2.945	27.60	28.19	81.28	2.16
3.123	25.96	26.59	81.07	2.43
3.304	24.52	25.13	81.01	2.51
3.487	23.20	23.81	80.90	2.64
3.671	22.00	22.62	80.76	2.82
4.138	19.47	20.07	80.57	3.06
4.611	17.43	18.01	80.37	3.32
5.089	15.78	16.32	80.30	3.41
5.840	13.69	14.22	79.95	3.86
6.814	11.70	12.19	79.72	4.16
7.794	10.19	10.65	79.42	4.56
8.779	9.05	9.46	79.45	4.53
9.766	8.07	8.50	78.81	5.36
10.756	7.29	7.72	78.41	5.90
13.237	5.90	6.27	78.10	6.32
15.724	4.90	5.28	77.05	7.77
18.215	4.20	4.56	76.50	8.55
20.708	3.70	4.01	76.62	8.38
25.699	2.96	3.23	76.07	9.16
30.692	2.45	2.71	75.20	10.42
35.687	2.08	2.33	74.23	11.87
40.684	1.84	2.04	74.86	10.93
48.679	1.52	1.71	73.99	12.23

September 3, 1915. Barometer 29.6 in. Temperature at start 67.3 deg. fahr., temperature at close 66.9 deg. fahr.

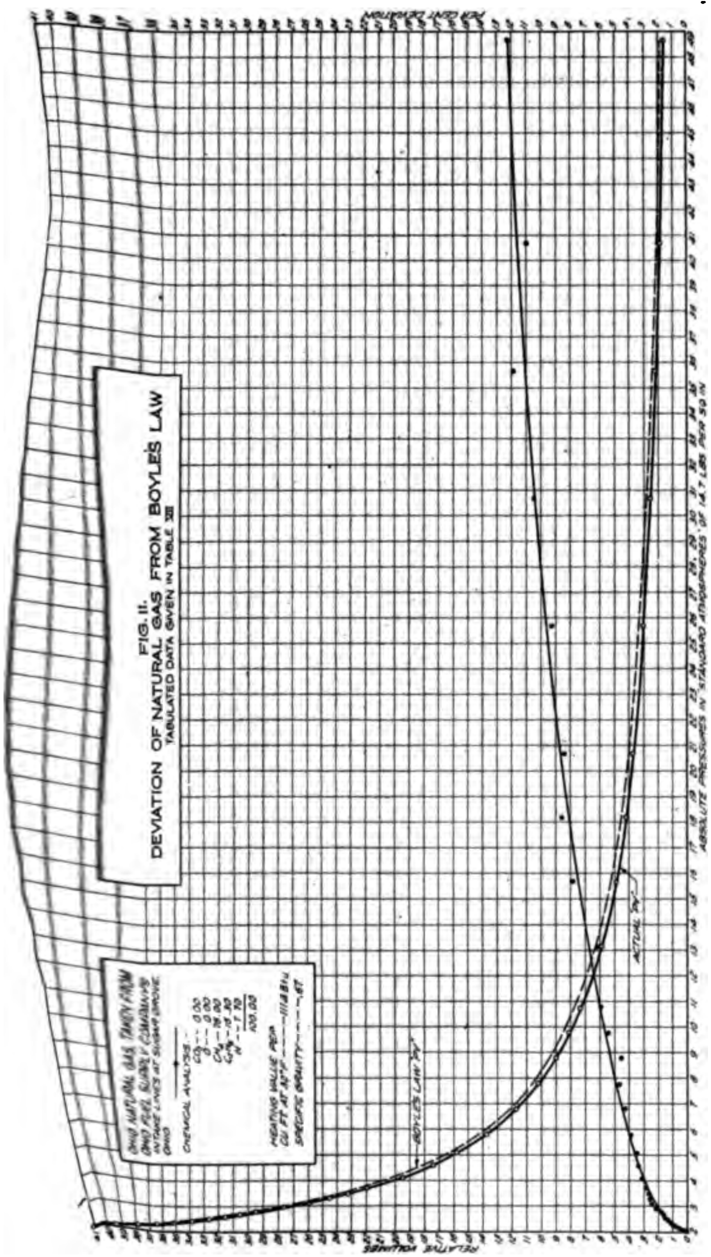


FIG. 11 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

TABLE 8 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

MIXTURE OF OHIO AND WEST VIRGINIA NATURAL GAS TAKEN FROM THE ST. MARY'S MEASURING STATION OF THE OHIO FUEL SUPPLY COMPANY AT COLUMBUS, OHIO.
GRAPHICAL DATA FOR THIS ARE SHOWN IN FIG. 12

P Absolute Pressure in Standard Atmospheres	V Relative Volume Observed	Boyle's Law Volume	PV Product of P and V as "Constant"	Per Cent Deviation
1.884	38.12	38.12	71.82	0.00
1.963	36.47	36.59	71.59	0.32
2.043	34.95	35.15	71.40	0.59
2.124	33.56	33.81	71.28	0.76
2.207	32.22	32.54	71.11	0.99
2.291	31.01	31.34	71.04	1.07
2.375	29.82	30.24	70.82	1.41
2.461	28.74	29.18	70.73	1.55
2.547	27.72	28.19	70.60	1.68
2.635	26.77	27.25	70.54	1.81
2.723	25.85	26.37	70.39	2.03
2.901	24.21	24.75	70.23	2.105
3.082	22.74	23.30	69.97	2.64
3.265	21.42	21.99	69.96	2.60
3.449	20.24	20.82	69.81	2.88
3.635	19.19	19.75	69.76	2.95
4.106	16.94	17.49	69.56	3.25
4.582	15.11	15.67	69.23	3.74
5.062	13.64	14.18	69.05	4.02
5.546	12.44	12.94	68.99	4.10
5.816	11.85	12.34	68.92	4.21
6.793	10.09	10.57	68.54	4.78
7.776	8.77	9.23	68.30	5.15
8.763	7.77	8.19	68.09	5.48
9.752	6.92	7.36	67.48	6.43
10.743	6.26	6.69	67.25	6.79
12.729	5.22	5.64	66.45	8.08
14.719	4.47	4.88	65.79	9.16
16.712	3.90	4.29	65.18	10.18
18.706	3.45	3.83	64.54	11.28
20.701	3.10	3.47	64.17	12.39
25.693	2.43	2.80	62.43	15.04
30.687	2.02	2.34	61.99	15.86
35.683	1.72	2.01	61.37	17.03
40.680	1.48	1.77	60.21	19.28
48.677	1.22	1.48	59.39	20.93

August 31, 1915. Barometer 29.6 in. Temperature at start 66.2 deg. fahr., temperature at clc
66.9 deg. fahr.

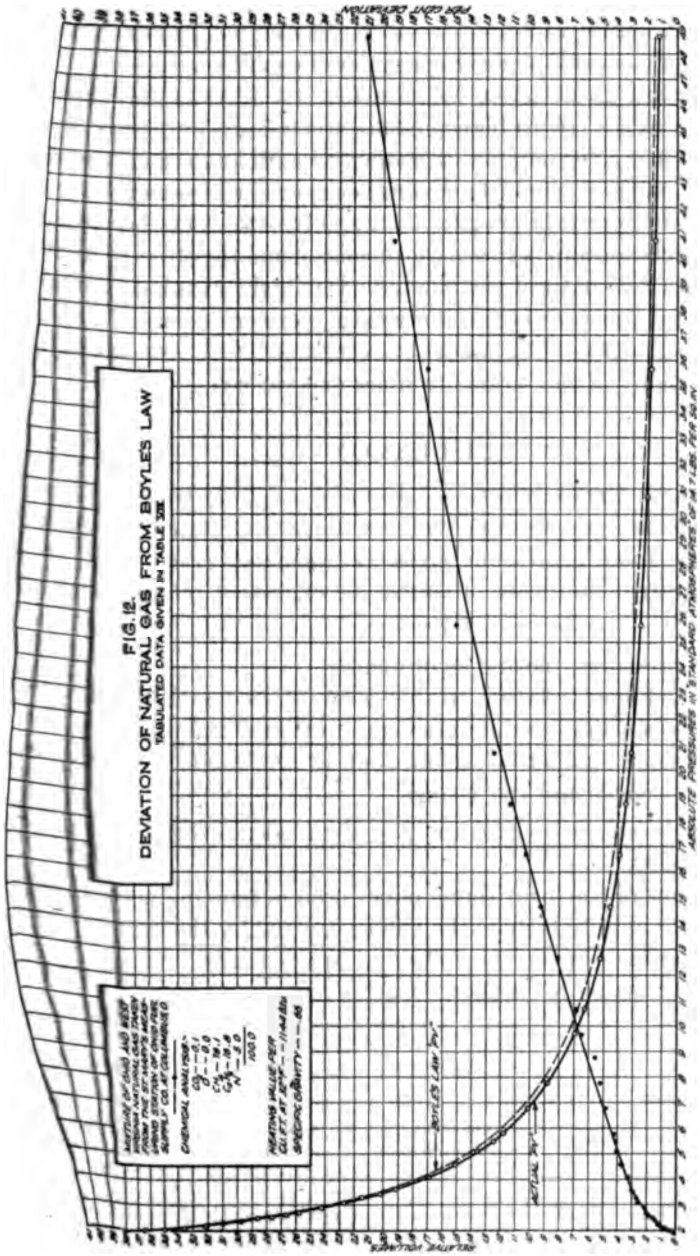


FIG. 12 DEVIATION OF NATURAL GAS FROM BOYLE'S LAW

It is important to note that all of the deviations are in one direction, and in favor of the buying company where gas is measured under high-pressure conditions.

The data given herein indicate that the direct application of Boyle's law to high-pressure natural-gas measuring problems gives only an exceedingly rough approximation.

The different gases have marked peculiarities and marked differences of deviation. Furthermore, there is no direct relationship between the deviation and the ethane content.

In practice natural gas is handled at high pressures and at a temperature below the critical point of some of its constituents. The failure to recognize this is responsible for many of the serious troubles experienced.

DISCUSSION

Charles N. Cross (written). In order to study better the relationship between pressure and volume, I have plotted the authors' observations upon logarithmic cross-section paper (Figs. 13 to 17), where all straight lines can, of course, be expressed by the equation

$$PV^n = c$$

where P = pressures, plotted as ordinates
V = volumes (relative, in this case), plotted as abscissæ
c = a constant = the intercept on the P-axis when $V = 1$
n = some number = the tangent of the angle between the curve and the V-axis.

For Boyle's law, n is equal to 1, or the angle of the line and the V-axis is 45 deg.

Limitations of the logarithmic paper used in plotting necessitated three separate lines to include the values covered by the experiments. Each line varies between the following approximate limits. Curve B is omitted because of its short length.

	Curve A	Curve B	Curve C
P (atmospheres)	1 to about 7	About 7 to 10	10 to about 70
V (relative)	About 80 to 10	10 to about 8	About 8 to 1

In Table 9 are given the values of n, together with the percentages of ethane and nitrogen in the gases. As was expected, n for each of of the eight gases is a number less than unity.

As shown in Table 9, the values of n for curves A differ materially from the values for the corresponding C curves, excepting for gas of Table 5, the practically pure methane gas. The curves A for gases of Tables 3, 4, and 7, when drawn as two straight lines of different slopes, pass through nearly every point. A slightly curved line drawn through the points, possibly, might represent the observations more correctly. There are not sufficient data to settle this point definitely.

TABLE 9 VALUES OF n

Gas of	Value of n		Per cent of C_2H_6	Per cent of N
	Curve A	Curve C		
Table 1	0.969	0.942	19.1	1.9
Table 2	0.975	0.949	18.8	1.2
Table 3	0.955 and 0.990	0.960	21.7	0.2
Table 4	0.940 and 0.970	variable	37.7	1.0
Table 5	0.969	0.969	0.0	1.4
Table 6	0.980	0.987	9.3	37.7
Table 7	0.958 and 0.974	0.958	16.3	7.7
Table 8	0.968	0.923	16.8	5.0

If the contract price for gas per cubic foot is based upon a pressure of about 2 atmos., then the percentage deviations as given in the paper are correct. But, as is probably more usual, this pressure is very close to one atmosphere, then the deviation from Boyle's law should be zero at that pressure. Also, in the absence of positive evidence to the contrary and from our general knowledge of the laws of gases, it is hardly to be expected that n would change abruptly at about 2 atmos. to a new value larger than a value between 2 and 8 atmos.

I have searched diligently through the literature for experiments upon gases at low pressures, say from 1 to 2 atmos., and have been unable to find any results upon so-called permanent gases giving evidence that below 2 atmos. n should differ from its value between 2 to 10 atmos. Amagat spent considerable time and thought upon gases at very low pressures and was unable to arrive at any conclusion in this matter because of the magnitude of experimental errors, particularly in the measurement of pressures.[1]

By extending the straight lines on the logarithmic paper back to a pressure of one atmosphere and reading off the volume, a new

[1] See *Annales de Chimie et de Physique*, 1883, Série 55, vol. 28, pp. 480–499.

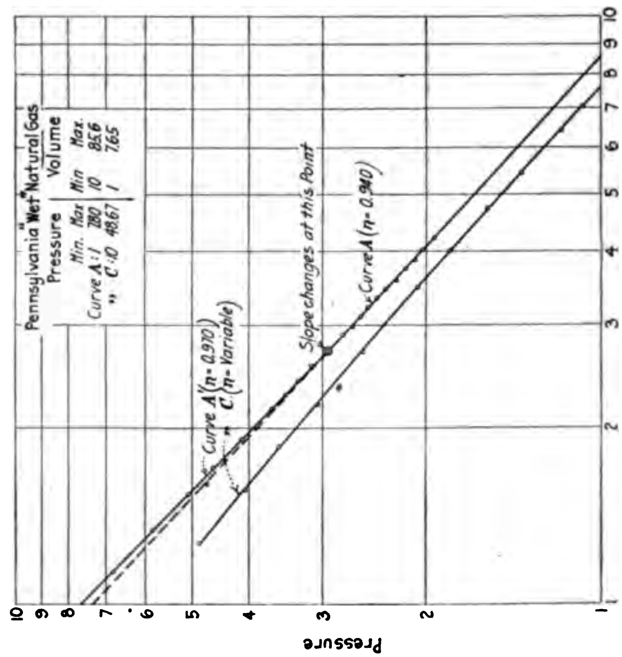


FIG. 14 LOGARITHMIC PLOT OF RESULTS IN TABLE 4

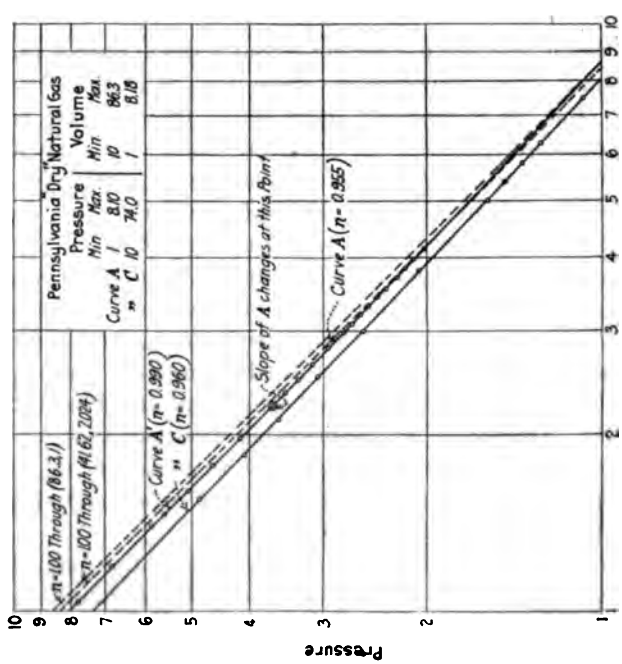


FIG. 13 LOGARITHMIC PLOT OF RESULTS IN TABLE 3

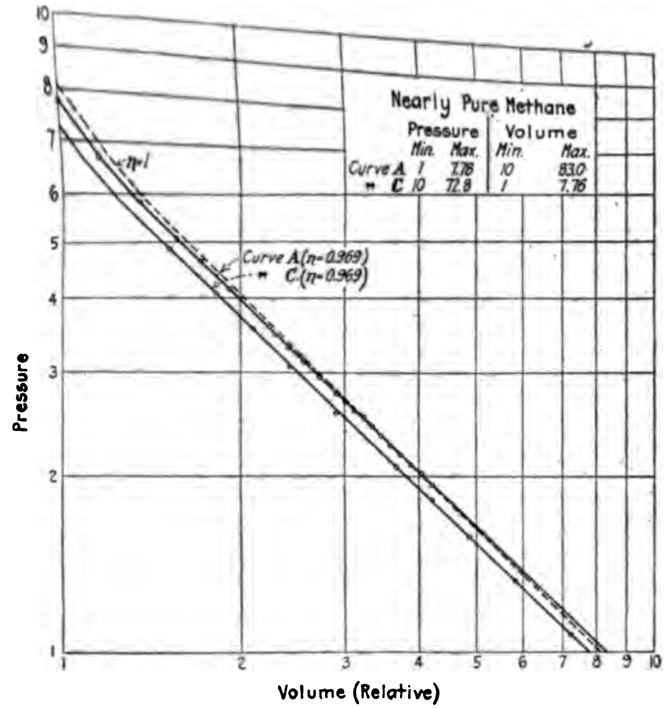


FIG. 15 LOGARITHMIC PLOT OF RESULTS IN TABLE 5

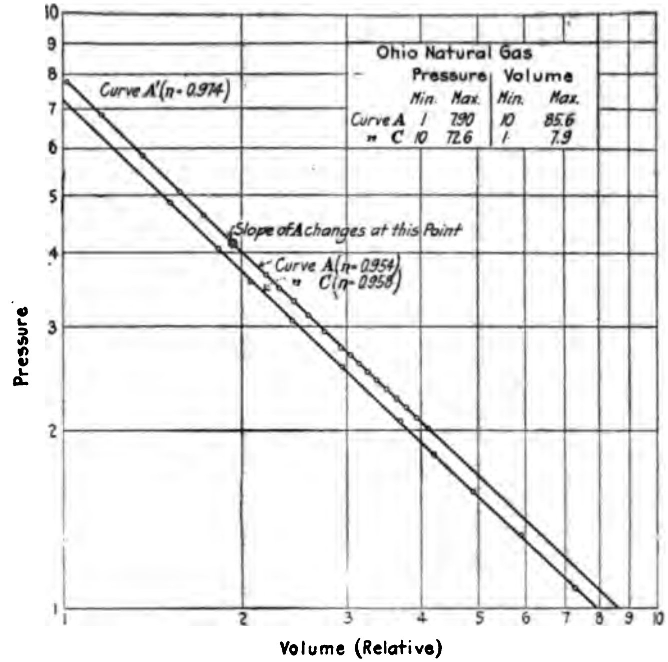


FIG. 16 LOGARITHMIC PLOT OF RESULTS IN TABLE 7

PV product for the initial, slightly different in every case from that found by the authors at about 2 atmos., was determined, and using this as a starting point, Table 4 was abridged and recomputed, as shown in Table 10. The percentage deviation is in every case appreciably larger. The deviation at 5 to 10 atmos. is increased 40 to 100 per cent. Hence the importance of locating correctly the point of zero deviation is clearly shown.

TABLE 10 (TABLE 4 REVISED)

Pennsylvania "Wet" Natural Gas

P Absolute pressure in standard atmospheres	V_o Relative volume observed	V_b Boyle's-law volume	Per cent deviation between V_o and V_b New	Per cent deviation between V_o and V_b Earhart & Wyer
1.00	85.6 [1]	85.6 [1]	0.00 [1]	....
1.936	42.03	44.25	5.27	0.00 [1]
2.254	35.80	38.01	6.18	0.84
2.676	29.92	32.02	7.02	1.62
3.482	22.78	24.55	7.78	2.58
5.837	13.41	14.70	9.62	3.96
10.754	7.06	7.96	12.72	7.18
20.706	3.47	4.14	19.30	13.24
25.696	2.70	3.33	23.30	17.28
48 677	1.27	1.76	38.60	31.62

[1] From logarithmic curve.

Across Fig. 13 are drawn 45-deg. reference lines through the initial observed point and through the point where the extrapolated curve A cuts the V-axis at a pressure of 1 atmosphere. Hence, at any pressure along curve A, the deviations can be seen at a glance.

Figs. 18 and 19 have been plotted from the observations of E. H. Amagat upon methane, hydrogen and ethylene at high pressures (30 to 420 atmos.), taken from The Laws of Gases, edited by Carl Barus, 1899. These curves may aid in understanding some of the deviations shown by the natural-gas curves. They illustrate briefly the facts as shown by Van der Waals' equation that $(P + a/v^2)(V - b) = C$ for constant temperature, where a/v^2 = the cohesion pressure between the molecules of the gas, and b = the volume of the gas when it is compressed to the utmost.

As a further aid, the following critical temperatures and pressures from the Smithsonian Physical Tables are inserted here.

	H	N	CH_4	C_2H_4	C_2H_6
Critical temperature, deg. fahr.....	−390.5	−220	−115	48.5	95.0
Critical pressure, atmospheres.....	20	35	54.9	58	45.2

It is of interest to compare the values of n from Amagat's data for methane with that for the practically pure methane gas in Table 5 of the authors' paper. See Figs. 15 and 18.

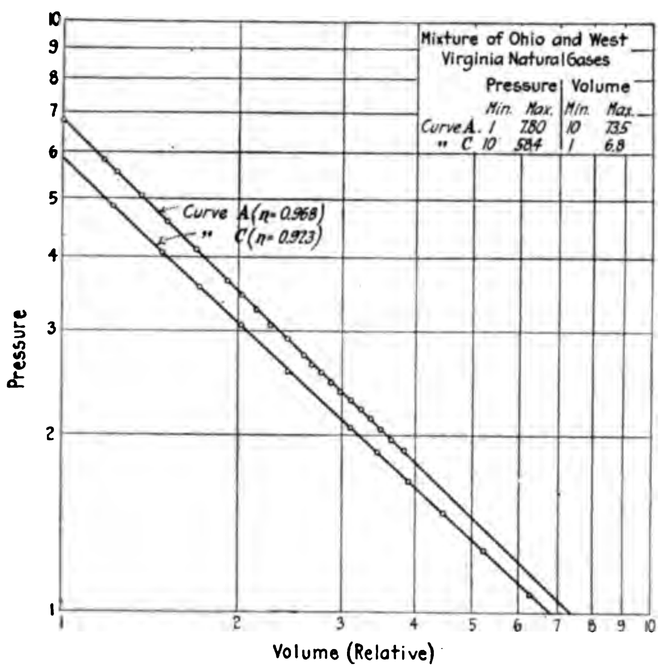


FIG. 17 LOGARITHMIC PLOT OF RESULTS IN TABLE 8

	Temperature, deg. fahr.	n (from straight portion of curve)
Methane (Amagat).....................................	58.5	0.905
Methane (Amagat).....................................	176.0	0.966
Methane gas (Earhart and Wyer)	67.6	0.968

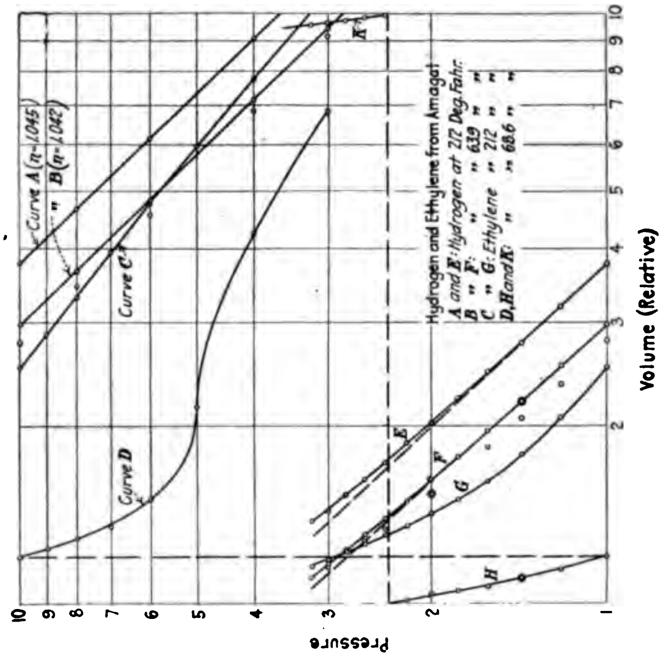


FIG. 19 LOGARITHMIC PLOT OF AMAGAT'S OBSERVATIONS

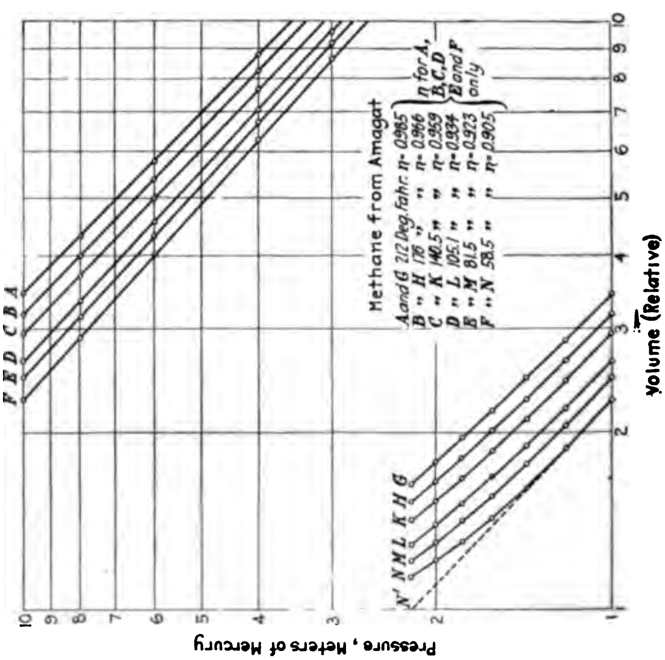


FIG. 18 LOGARITHMIC PLOT OF AMAGAT'S OBSERVATIONS

So great a difference for the value of n would hardly be expected. It has been alleged that the hydrogen used by Amagat was not pure, but contained as much as 20 per cent carbon monoxide. (See Study of Gases, by M. W. Travers, 1901 ed., p. 165.) Since much more is known about the analysis of gases now than in 1875 to 1885, it is barely possible that the methane experimented upon by Amagat was not pure.

It is shown in Figs. 18 and 19 that gases at temperatures far above the critical temperature plot as straight lines until the pressure becomes very high, when the lines become convex to the V-axis, showing that the value of n increases. For gases below the critical temperature the curves may be straight for a short distance, but soon become concave to the V-axis. Then, with an increase of pressure, the curvature is reversed and the lines become convex, as is characteristic of the curves of the so-called permanent gases. This explains why the "wet" gas of Table 4 and Fig. 14, containing 37 per cent ethane, is first straight and then concave to the V-axis.

The writer is unable to explain why the slope of the A curves should increase with the pressure in the case of natural gas.

E. O. Hickstein (written). It would be interesting to know if any precautions were taken in this work to eliminate water vapor present in the gas. If not, is there any probability of the presence of such vapor having an appreciable effect?

Some tests made by the writer about a year ago suggest a method of test which perhaps has not previously been used in this connection. The possible sources of error in this method are so few that the results may be interesting.

In tests made at Wann, Oklahoma, on orifice-meter disks, a known volume of pipe line (one mile of 12-in.) was filled with high-pressure gas and closed off from the source of supply. It was then passed through different types of meters at a constant rate of discharge into the atmosphere. The length of time during which gas escaped from the reservoir was taken by stop watch, and from this reading, together with initial and final pressures in the reservoir, and its volume, the rate of discharge could be calculated, assuming that the gas acted in accordance with Boyle's law. The figures presented here were obtained from these tests.

On January 24, 1915, four tests were made without refilling the pipe-line reservoir. The rate of discharge was kept constant, the flow being regulated by an operator who slowly opened a gate through which the gas was expanded to a low pressure and then discharged to the atmosphere through a meter prover. The readings taken during the four runs are reported in Table 11, which is self-explanatory.

From these data was calculated the number of cubic feet of gas at atmospheric pressure passing per minute through a square inch of orifice area, reduced to standard conditions of 60 deg. fahr. flow temperature, 0.675 gravity and 2.70 in. of water pressure in prover. The results appear in Column 2 of Table 12.

TABLE 11 DATA TAKEN DURING WANN TESTS, JANUARY 24, 1915, TO INVESTIGATE *PV* DEVIATION

(Barometer reading, 29.4 in. Gravity of gas, 0.672)

Run No.	Initial and final pressures of reservoir, lb. per sq. in., gage	Duration of tests, seconds	U-tube reading of prover, in. of H_2O	Temperature of gas at prover outlet, deg. fahr.	Temperature of storage line, deg. fahr.
1	200.5 147.0 ——— 53.5	1310	2.81	32	35
2	146.5 96.0 ——— 50.5	1274	2.81	33	35
3	96.0 47.0 ——— 49.0	1239	2.84	33	35
4	47.0 6.75 ——— 40.25	1237	2.82	39	35

As the prover was working at all times under identical conditions, it is clear that the discrepancy between these four values must be due to the fact that the calculation of the rate of flow, based on Boyle's law, was incorrect. If, now, it is assumed that Boyle's law is true between 50 and 0 lb. per sq. in. pressure, that is, that the fourth test is correct as figured, 32.43 is the correct constant for the prover orifice; and the ratio of this figure to the other values in Column 2 of Table 12 gives the correction factor for the other tests as recorded in Column 3 of the same table.

The excess of gas, over Boyle's law, passing out during each of the first three tests, is expressed in atmospheres and in cubic feet of the gas in Columns 4 and 5 of Table 12. The accumulative excess over Boyle's law is shown in columns 6 and 7 of the same table. Fig. 20 indicates how it is possible to correct for the assumption made that the fourth test and Boyle's law give identical discharges from the reservoir, by showing that, in reality, the excess at 50 lb.

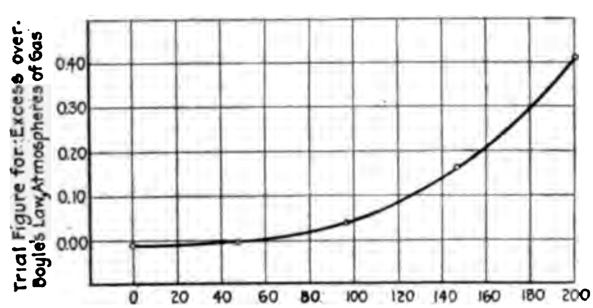


FIG. 20 METHOD OF CORRECTING FOR ASSUMPTION THAT BOYLE'S LAW IS CORRECT BETWEEN 0 AND 50 LB. PER SQ. IN.

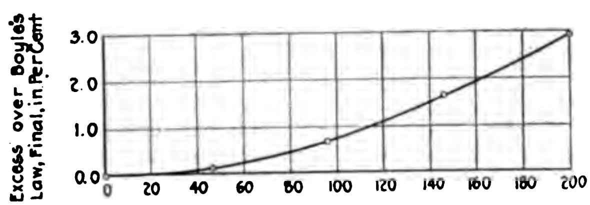


FIG. 21 FINAL *PV* DISCREPANCY CURVE

per sq. in. is 0.010 atmos. (2.74 cu. ft.) over Boyle's law. Correcting for this 0.23 per cent of the theoretical quantity between 47 and 0 lb. per sq. in., the true constant for the prover orifice is 32.50 (= 1.0023 × 32.43). These corrected figures are given in Table 13.

The *PV* deviation expressed in per cent is as follows:

At 200.5 lb. per sq. in. (14.9 atmos.), 2.96 per cent
146.5 lb. per sq. in. (11.2 atmos.), 1.66 per cent
96.0 lb. per sq. in. (7.7 atmos.), 0.70 per cent
47.0 lb. per sq. in. (4.2 atmos.), 0.14 per cent

These percentages will be found plotted in Fig. 21.

While these figures are given more for their interest as a mathematical analysis, the method used appears to possess advantages. It relies for its accuracy on the correctness of a stop watch, pressure gage and thermometer, and is absolutely independent of the reservoir volume, prover calibration, or gravity of the gas.

TABLE 12 PRELIMINARY FIGURES, BASED ON ASSUMPTION THAT BOYLE'S LAW IS TRUE UP TO 50 LB. PER SQ. IN.

Run No.	Calculated constant for prover orifice, based on Boyle's-law calculation of gas quantity	Ratio of 32.43 (the constant assumed as correct) to value in previous column	Excess over Boyle's law for each run		Accumulative excess over Boyle's law (initial pressure to zero pounds)	
			Atmospheres	Cu. ft.	Atmospheres	Cu. ft.
1	30.39	1.0672	0.249	68.3	0.412	113.0
2	31.33	1.0352	0.123	33.7	0.163	44.7
3	32.05	1.0119	0.040	11.0	0.040	11.0
4	32.43	1.0000	0.000	0.0	0.000	0.0

The gas tested was taken out of a line supplying Oklahoma gas, which was a mixture from the Cushing and Pawhuska fields and had a gravity of 0.672.

TABLE 13 FINAL FIGURES, CORRECTED FOR ORIGINAL ASSUMPTION THAT BOYLE'S LAW IS TRUE UP TO 50 LB. PER SQ. IN.

Run No.	Calculated constant for prover orifice, based on Boyle's-law calculation of gas quantity	Ratio of32.50 (the correct constant, calculated from curve, Fig. 20)to value in previous column	Excess over Boyle's law for each run		Accumulative excess over Boyle's law (initial pressure to zero pounds)	
			Atmospheres	Cu. ft.	Atmospheres	Cu. ft.
1	30.39	1.0695	0.258	70.8	0.443	121.4
2	31.33	1.0375	0 131	35.9	0.185	50.6
3	32.05	1.0142	0.048	13.1	0.054	14.7
4	32.43	1.0023	0.006	1.6	0.006	1.6

It will be noted that the magnitude of the deviation as found in these tests is considerably smaller than any found by the authors and that the shape of the curve is quite different, but as the samples tested by the authors did not come from Oklahoma, this may not be significant.

G. A. Burrell [1] and I. W. Robertson [2] (written). The Bureau of Mines has developed a formula by means of which the deviation of any natural gas, at any pressure, can be determined from a simple analysis of the gas. The advantage lies in the ease with which analyses may be made. See Bulletins 15, 42 and 88 and Technical Paper 109, Bureau of Mines, covering analysis of natural gas and other characteristics.

In order to develop a formula by means of which the compressibility of natural gas (deviation from Boyle's law) could be determined

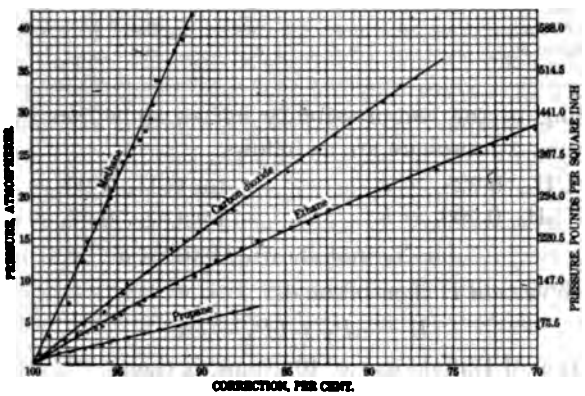


Fig. 22 Compressibility of Gases Found in Natural Gas

from the analysis, the compressibility of the gases found in natural gas, principally methane, ethane, propane, and carbon dioxide, were first determined. The compressibility of nitrogen has been determined by others. This preliminary information is given in the form of curves in Fig. 22.

In Table 14 are the analyses of nine samples of natural gas.

The formula was developed as follows: Suppose it is desired to determine the compressibility of the natural gas used in Buffalo at 40 atmos. pressure. Buffalo gas has the following percentage com-

[1] In charge, Laboratory of Gas Investigations, Bureau of Mines.
[2] Junior chemist, Bureau of Mines. Pittsburgh, Pa.

position: CH_4, 88.1; C_2H_6, 11.5; N_2, 0.4. At 40 atmos. the partial pressure of the

$$\begin{aligned} CH_4 &= 0.881 \times 40 = 35.2 = P_1 \\ C_2H_6 &= 0.115 \times 40 = 4.6 = P_2 \\ N_2 &= 0.004 \times 40 = 0.2 = P_3 \end{aligned}$$

TABLE 14 PERCENTAGE ANALYSES OF SAMPLES OF NATURAL GAS FOUND IN NINE CITIES IN THE UNITED STATES

City	CO_2	CH_4	C_2H_6	N_2	Total
Little Rock, Ark.	1.0	96.7	0.0	2.3	100
Bartlesville, Okla.	2.8	89.1	4.2	3.9	100
Cleveland, Ohio	0.0	83.6	15.7	0.7	100
Los Angeles, Cal.	25.1	59.1	14.5	1.3	100
Palestine, Ill.	0.5	95.6	0.0	3.9	100
Buffalo, N. Y.	0.0	88.1	11.5	0.4	100
Kansas City, Mo.	0.8	84.1	6.7	8.4	100
Charlestown, W. Va.	0.0	76.8	22.5	0.7	100
Olney, Ill. (H_2S=1.2%)	0.0	37.5	59.6	1.7	100

According to the compressibility curves shown in Fig. 22, the deviations per atmosphere are as follows:

for CH_4, 0.228, $= a$ for C_3H_8, 1.90, $= c$
for C_2H_6, 0.900, $= b$ for CO_2, 0.67, $= d$
for N_2 (not shown), 0.01 $= e$

The total deviation D then becomes

$$D = aP_1 + bP_2 + \text{etc.}$$

The deviation of Buffalo gas at 40 atmos. is then

$$\begin{aligned} D &= (0.228 \times 35.2) + (0.90 \times 4.6) \\ &= 8.03 + 4.14 = 12.17 \text{ per cent.} \end{aligned}$$

The calculated and determined deviations at a pressure of 20

TABLE 15 CALCULATED AND DETERMINED DEVIATIONS FROM BOYLE'S LAW OF DIFFERENT NATURAL GASES AT A PRESSURE OF 20 ATMOSPHERES

	Per cent deviation			Per cent deviation	
	observed	calculated		observed	calculated
Little Rock	5.8	4.4	Buffalo	6.0	6.1
Bartlesville	4.8	4.8	Kansas City	4.0	4.4
Cleveland	4.3	4.5	Charleston	6.4	7.5
Los Angeles	7.8	7.6	Olney	12.3	12.4
Palestine	4.2	4.4	.....	.....	.....

atmos. for different natural gases experimented with by the writers are recorded in Table 15; and the results there given show that it is possible to calculate the compressibility of different natural gases, having given the analysis of the gas.

THOMAS R. WEYMOUTH (written). At the suggestion of Mr. Wyer, I made a test of a gas from Roystone Station, where the samples indicated in Tables 3 and 4 were obtained, by a method originally proposed by Mr. E. B. Rosa of the Bureau of Standards. This consisted in filling a small tank of known capacity with the gas at high pressure, which, after its temperature and pressure had been observed, was expanded into a 10-cu.-ft. meter prover. The increase in volume, as indicated by the lift of the prover drum, added to the volume of the original container, gave the total volume of the gas at the pressure and temperature in the prover. In order to maintain the temperature of the gas as nearly constant as possible, it was expanded very slowly, at a rate of about one-half a cubic foot per minute. After the expansion was complete and observations of the lift and pressure of the prover had been made, the connection to the container was broken and the gas was allowed to flow from the prover through the connecting hose with a thermometer in its path, which immediately settled to the value given in the data and remained there. This temperature was the same as that obtained on similar observations of air, made just previous to the run on gas. This, therefore, is considered as giving the true temperature of the expanded gas.

The high pressure of the gas in the tank was measured by means of a sensitive bourdon spring test gage, calibrated against a dead-weight tester at the time of the test. The temperature of the gas in the tank was observed by means of a thermometer placed in contact with the exterior of the tank and heavily lagged. Three observations of pressure and temperature were made after having allowed the container to remain in rooms of different temperatures long enough to become constant in each instance, and from these observations the pressure existing in the tank at the temperature of the expanded gas was computed. The average of the three results was then used in the final computations. The data obtained were as follows:

V = volume of container at 60 deg. fahr. = 0.3938 cu. ft.
B = barometer at time of test = 29.08 in. = 14.249 lb. per sq. in.

Pressure in gas container =

$p' = 277.15$ lb. per sq. in. gage at 72.5 deg. fahr.
$p'' = 270.50$ lb. per sq. in. gage at 61.4 deg. fahr.
$p''' = 271.00$ lb. per sq. in. gage at 62.0 deg. fahr.
or $P' = 283.85$ lb. per sq. in. absolute at 58.7 deg. fahr.
$P'' = 283.28$ lb. per sq. in. absolute at 58.7 deg. fahr.
$P''' = 283.45$ lb. per sq. in. absolute at 58.7 deg. fahr.
Avg. $P = 283.53$ lb. per sq. in. absolute at 58.7 deg. fahr.
h = pressure in prover through test = 2.07 in. water
P_0 = absolute pressure of expanded gas = $B + 0.036\,h$ = 1[illegible].324 lb. per sq. in.
L = increase in volume = lift of prover = 8.07 cu. ft.
N = actual ratio of total expanded gas volume to the volume of high-pressure gas = $(L + V)/V = 21.492$
N_0 = ratio of volumes according to Boyle's law = P/P_0 = 19.794

Deviation factor = $N/N_0 = 1.0858$.

TABLE 16 COMPUTED VALUES OF n FROM $P^nV=KT$

Gas of	P_1 obs.	V_1 obs.	T obs.	P_2 obs.	V_2 obs.	n computed	Gravity
Fig. 5	2.029	41.31	534.8	20.706	3.69	1.040	[illegible].66
Fig. 6	2.020	41.35	526.2	20.709	3.73	1.033	[illegible].65
Fig. 7	2.024	41.62	526.1	20.705	3.82	1.027	[illegible].66
Fig. 8	1.936	42.03	528.4	18.214	4.02	1.047	[illegible].75
Fig. 9	1.936	42.10	527.6	20.708	3.67	1.030	[illegible].56
Fig. 10	2.018	40.98	533.9	20.700	3.75	1.027	0.76
Fig. 11	2.022	41.07	527.1	20.708	3.70	1.033	0.67
Fig. 12	1.884	38.12	526.6	20.701	3.10	1.047	0.66
T. R. W.	14.320	8.464	518.7	283.530	0.3938	1.028	0.76
Average	....	....	....	....	....	1.035	....

In other words, the percentage error of this gas is 8.58 at a pressure of 283.53 lb. per sq. in. absolute, or 19.29 standard atmospheres. The analysis of the gas showed it to contain 56.0 per cent methane, 36.2 per cent ethane, and 7.8 per cent nitrogen.

The error introduced by the assumption of the validity of Boyle's law in cases where actual volumes at high pressure are measured by a volumetric meter and reduced to standard conditions by applying direct-pressure ratios, is double the error in the measurement by rate-flow meters, such as the pitot tube and orifice, wherein the pressure factor enters under the radical sign.

In order to apply the information given in the paper to actual measurement problems, I have assumed an empirical equation of the form $P^n V = KT$, wherein the characters have the usual significance. Applying in this formula the original observations at about 2 and 20 atmos., the results are shown in Table 16, from which it is seen that the exponent of P is fairly constant, considering the wide variation in the constitution of the gases tested. From this fact I believe that much closer approximations to the true measurement of natural gas can be obtained by the use of the formula given than is now the case with the assumption of Boyle's law.

In order to determine how closely the empirical formula adheres to the actual expansion curve given by the authors, I have computed from the test of Table 3 the volume corresponding to the observed pressures, using the formula $P^{1.027}\ V = 85.86$. The deviation of the computed volume from the actual is less than 1 per cent up to 30 atmos., and by slightly changing the exponent of P and the value of K this deviation can be made much smaller.

In order to employ the equation in the transformation to standard conditions of measurement made by volumetric apparatus, it is merely necessary to multiply the measured volume by the nth power of the absolute pressure ratios.

In the case of measurements with rate-flow meters, such as the orifice or pitot tube, the pressure factor under the radical in the usual formula of such instruments (see Journal Am.Soc.M.E., Dec., 1912, p. 1645) is used with the exponent n, and the numerical coefficient is changed to correspond with the proper value of K in the empirical formula.

W. C. Baxter (written). The tests show the expected deviation from Boyle's law without proving it to be a commercial factor and without suggesting a means for correcting for this deviation. The authors point out the impracticability of attempts to correct for the deviation by saying the different gases have marked differences of deviation, and that no relationship can be established, since the exact ethane content cannot be determined, any higher hydrocarbon present being shown as ethane in the usual analysis. Furthermore, any appreciable amount of the higher hydrocarbons would probably have a greater influence than a high percentage of ethane.

Refinements in the measurements of natural gas are to be welcomed by purchaser and selling company, but the statement that "the direct application of Boyle's law to high-pressure gas measurements gives an exceedingly rough approximation" is very alarming, if not altogether misleading, since it is made without regard to other considerations that enter into the measurement.

Any sand, oil or water passing through a meter would not be affected by deviations from Boyle's law, but their presence in any considerable quantity results in an overmeasurement. In practically all field measurements, large quantities of water vapor and oil vapor are condensed after having passed the meter, causing a shrinkage or loss to the purchasing company. The difficulty of removing such vapors at or near the fields can be appreciated only by those having actual field experience.

The remarkable feature of the test results is that the percentage deviation curve has a point of inflection at 2 atmos. pressure, when the tests at lower pressure, not published, show no deviation. This would indicate that the roughest approximation would be made by applying Boyle's law at or near 2 atmos., introducing an uncertainty as to which party is favored. The upper parts of the curves indicate that the percentage factor might become constant or decrease with a sufficiently high pressure. However, very little natural gas is measured above 400 lb. per sq. in. gage, and an extension of the curve would be of no practical value.

FRANCIS P. FISHER (written). Because of the disagreement in numerical results obtained from experiments on the deviation of natural gas from Boyle's law when slightly different methods have been used upon similar gases, the writer is led to examine in detail some factors of the methods used which might have led to erroneous conclusions in any or all of the experiments to be quoted.

In the work done in our laboratories, the method used by Mr. Baxter was that of expanding small increments of gas from a metal container of known volume into a second similar container, and observing by means of pressure increase in the second container the volume of the expanded gas delivered in order to effect a given reduction in pressure in the original container. This roughly reproduced the conditions under which the gas is expanded in actual industrial application. The experiments were originally attempted by measuring the expanded gas in a precision gasometer of small bore and long travel, but the results were unsatisfactory because of the very difficult element of vapor tension in measuring over water, which showed the necessity of dry measurement.

In the work done at the laboratory of Kansas University at Lawrence by Professor Walker and Professor Garver, a mercury column was used as a source of pressure, and the method of compression was very similar to that in the experiments of Earhart and Wyer, with the exception that the precaution was taken to guard against the presence of water vapor in the gas sample by passing the gas carefully through chloride drying tubes before entering the apparatus. The importance of this was pointed out by our own experience in measuring over water. I do not find reference to any such precaution in the paper under discussion. The pressure in the Kansas University experiments was limited by the available height of the mercury column to slightly over 200 lb., but the apparatus admitted of very accurate work in the very important range between 1 and 2 atmospheres.

I would confine the present discussion to pointing out some possible causes for the difference between the factors determined by these different methods of investigation. Our own work has pointed out the great importance of freeing the gas from water vapor in order to get significant results. From this I would assume that the difference between the two compression methods, where similar samples were compressed in carefully calibrated tubes, would lie largely in the fact that the gas was carefully freed from water vapor in one of the tests.

I would suggest the following explanation as a possible source of disagreement between the facts calculated by means of comparison of a small volume in a graduated glass tube, and by method of expansion of increments of high-pressure gas from one dry container into another.

Our knowledge of oils, gasolines and fluids of the paraffin series would tend to show that such liquids will wet glass surfaces freely, whereas glass strongly repels mercury, giving a high and inverted meniscus as compared with gasoline. Any fluid formed would therefore have a strong tendency to wet the glass surface and remain below the upper surface of the mercury as compression was carried to the higher point, giving a false observation as to the true reduced volume by not including all of the substance of the original gas above the mercury. Any fluid thus trapped would represent a relatively large volume of the original gas, so that the slightest tendency in this direction would materially affect the value of the final observations.

It is necessary in filling a good barometer tube to boil the mercury for some time in all parts of the tube to remove the air from the surface of the glass. I would therefore infer that there is considerable probability that in these experiments of compression over mercury, irrespective of the condensation of small quantities of fluid, a volume of the gas itself is attached to the surface of the glass as the mercury column ascends, giving an unobserved deduction from the true quantity of substance and consequently a false reading of the volume.

In the concluding paragraph Mr. Wyer refers to experiments with a different apparatus in the range between 1 and 2 atmos., in

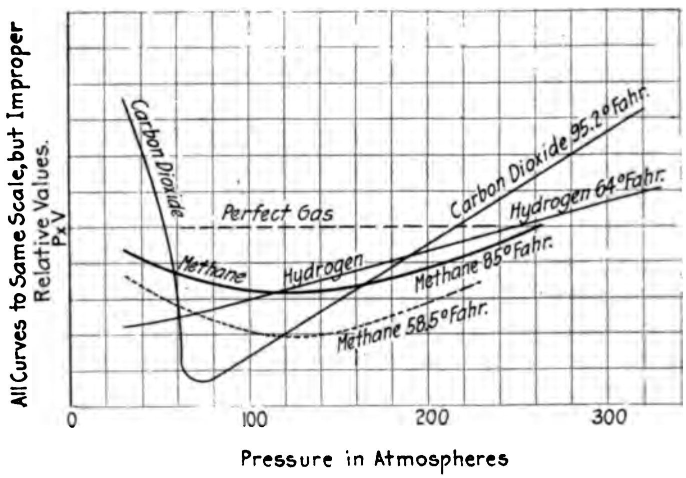


FIG. 23 CHARACTERISTIC VARIATIONS OF GASES FROM BOYLE'S LAW

which there was no perceptible change from Boyle's law within those limits. This result seems to be contradictory and irreconcilable with the shape of the curve of per cent deviation in the charts given, which show a maximum divergence of the curve from a horizontal line in the range immediately beyond 2 atmospheres. It would not seem reasonable or consistent with other experiments along the same line that gas should be compressed from absolute pressure of 1 atmos. to 2 atmos. "with no perceptible deviation from the law" and begin sharply at 2 atmos. to attain the maximum rate of deviation from this same law, for a given increase in pressure, falling off to a minimum deviation at a pressure higher than 4 or 5 atmospheres.

P. F. Walker (written). I have taken great interest in the matter treated in this paper, which bears upon the work of several scientists of note, particularly upon that of Amagat.

Amagat set up a mercury column with which to measure pressures in a mine shaft where he could secure high pressures, and for a gas-measuring tube employed a closed tube similar to that employed by the authors. After developing the characteristic curve for hydrogen shown in Fig. 23, he proceeded to investigate several gases not far removed from their critical states at ordinary temperatures, such as carbon dioxide, ethylene, and methane. The form of characteristic curve for the first two is that shown in Fig. 23 and marked

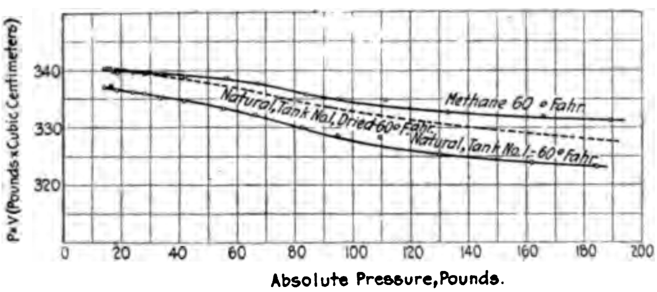


Fig. 24 Low-pressure Results on Methane and Natural Gas

carbon dioxide. The critical point, where the gas changes from diminishing to increasing values of the product, PV, is sharply marked. For higher temperatures this is less sharply defined, as represented for methane. At the higher temperatures and pressures the curves all appear to approach the form of that for hydrogen.

On the low-pressure side the gases close to saturation pass promptly from the critical point to a form of curve convex toward the horizontal line indicating the perfect gas, which convexity continues to the lowest pressures. At the lowest pressures the rate at which the gas deviates from the perfect gas with changing pressure becomes small, the curve being asymptotic to the horizontal. As these gaseous vapors become more like true gases this convexity diminishes, it being as though the methane curves of Fig. 23 were continued to the left without reversal of curvature. Whether methane reverses or not has not been shown absolutely as yet, but there is evidence to show that at the lower temperatures the curve does reverse.

Fig. 24 shows the definite character of the curve for methane at 60 deg. fahr. for pressures down to atmospheric, as determined with the apparatus in use in the Kansas laboratory. Alongside the methane curve is shown, also, the curve for a sample of natural gas containing about 87 per cent methane drawn from the Kansas-Oklahoma gas field. The effect of drying this gas before its admission to the measuring tube of the apparatus is shown by the broken curve to be but slight. The undried gas indicates a greater total deviation from the perfect gas, and both indicate greater deviation than does pure methane.

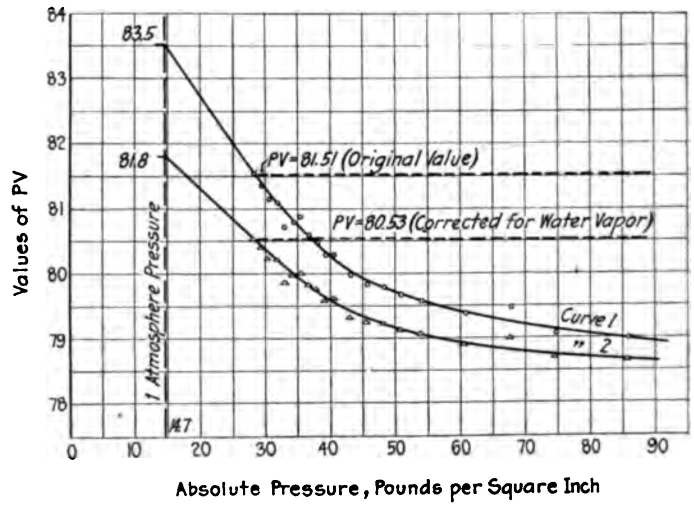


FIG. 25 AUTHORS' VALUES IN TABLE 5 REPLOTTED AND EXTENDED

These curves are in sharp contrast to the form of curve given by the data supplied by the authors. Fig. 25 is constructed from the data taken from Table 5 in the paper, it being for the natural gas which contains 98.6 per cent methane, with the remainder nitrogen. If the sharp curvature secured by the authors for all of their gases at the low pressures were due to the presence of condensable vapors, we would expect this curve for almost pure methane to be distinctly different in form. The only difference is in magnitude, however, which really is the result to be expected. The curve on original data in Fig. 25 has been extended to meet the atmospheric pressure line at a value of PV of about 83.5. If percentages of variation were to be calculated from this value, and in the gas business we want volume referred to atmospheric pressure, they would be considerably greater than those given by the authors.

As to the form of the curves at low pressures, it is clear that the sharp variation is such as would be produced by a constant error in the observed pressures. Such an error, the amount being added to the true pressure, would be of maximum significance at low pressure where the added quantity would be large in proportion to the true value, and would produce an effect on the form of the curve as noted. In order to determine if the presence of water vapor in the gas would account for the peculiarity, I have recalculated the values of PV after deducting from P the partial pressure which would be produced by vapor in the saturated state at the temperature of 67.6 deg. fahr. This partial pressure is 0.337 lb. per sq. in., or 0.023 atmos. The result is shown in Curve 2 of Fig. 25. Clearly this is not enough to change the form vitally, although it acts in that direction.

Gardner T. Voorhees said that he had made an investigation with carbon dioxide and had suggested that "at the critical point, and for some distance above it, the substance might be a chemical mixture of a liquid in solution in the gas and not a *true* gas in the present accepted sense of the word *gas*."

The Author. In answer to Mr. Hickstein's query regarding water vapor, we made some observations with dried gas, but the results obtained indicated that the tests would be of more practical value if the various gases were tested in their natural state, just as they pass through a gas-measuring device.

Replying to Mr. Baxter's stricture of our conclusions, we do not believe that our statements are alarming, misleading, or in error, but the facts are precisely as stated in the paper.

No. 1538

EXPERIMENTS ON WATER FLOW THROUGH PIPE ORIFICES

By Horace Judd, Columbus, Ohio

Member of the Society

During the past ten years papers upon the flow of fluids have appeared from time to time in the Transactions of the Society;[1] most of these have dealt with the flow of air and gas, although a few have treated of the flow of steam. The methods set forth in these papers have for the most part centered around the venturi meter and the pitot tube.

2 During this period, also, many forms of meters have been brought out for the measurement of air, gas and steam, as well as of water, all more or less successful in operation, and of these, also, a great many have made use of the principles of the venturi meter or the pitot tube.

3 The pipe orifice, consisting of a diaphragm in which there is an orifice inserted in a pipe, has not hitherto met with much favor for the measurement of the flow of steam or water in a pipe, although it is coming into use for air and gas measurement and has been used with certain types of steam meters of German make and also in Bailey flow meters, developed quite recently by E. G. Bailey, Mem. Am.Soc.M.E.

4 The chief objections to the pipe orifice have seemed to be that a special form of orifice flange was needed and that considerable uncertainty prevailed concerning the pressure conditions adjacent to the orifice, so that the working coefficients for an orifice in a pipe were not so reliable as those of the venturi meter or the pitot tube.

5 About two years ago, while employing the Bailey meter to measure the steam used by an engine, the writer became interested

[1] See references on page 363.

Presented at the Spring Meeting, New Orleans, La., April, 1916, of The American Society of Mechanical Engineers.

in the orifice as a pipe-flow measuring device. It was important to know at what point near the orifice the attachment should be made in order to measure correctly the pressure drop of the steam flowing through the orifice. It was thought at the time both by Mr. Bailey and the writer that an investigation would reveal many valuable points concerning the flow conditions through the orifice in a pipe. The present line of work is the final outcome of this thought and has been carried on in connection with water flow, since water could be more easily handled than steam, the intention being to continue later with a study of the flow conditions in a steam line.

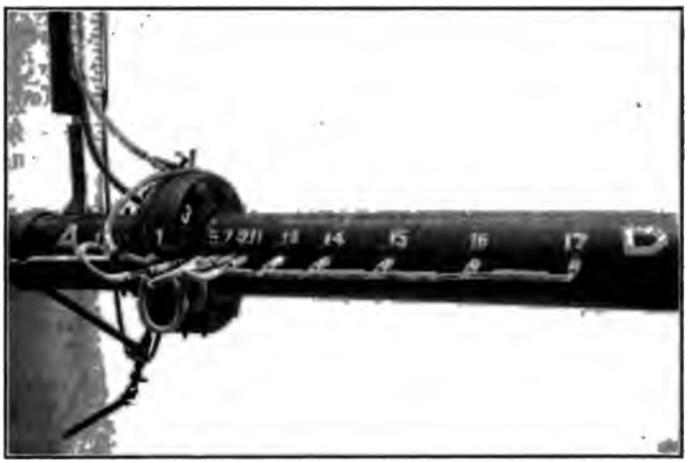


FIG. 1 MANIFOLD AND MANOMETER CONNECTIONS AT THE DIAPHRAGM FLANGE

6 The paper is really a record of progress, for only a few points have been touched upon, such as the pressure changes in the vicinity of the diaphragm, or orifice; the probable location of the least section, or zone of greatest velocity of the water jet; and the working coefficients of the diaphragms; leaving many other points of equal interest and importance to be studied later. The experiments were confined to one size of water pipe (5-in.) and the pressure drop through the diaphragm did not run much in excess of 6 ft. of water, which was considered to be the probable maximum drop desirable to use in connection with any automatic registering device. For the largest diaphragm this range of pressure drop gave a maximum velocity of 17.5 ft. per sec.

PURPOSE OF THE TESTS

7 In general, the purpose of the tests was

a To discover the pressure relations existing immediately above and below an orifice diaphragm in a pipe in order to determine the most suitable points for making the pressure connections (for the manometer).

b To note the position of greatest pressure drop due to flow through the orifice, or the probable point of greatest velocity, corresponding to the point of least section of a jet of water flowing freely into the air.

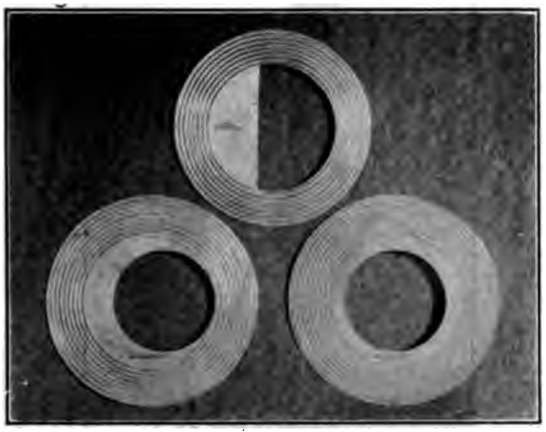


FIG. 2 MONEL METAL DIAPHRAGMS, 80% SIZES USED

c To obtain reliable coefficients of discharge for orifice diaphragms of different diameters.

THE EQUIPMENT

8 For these experiments a horizontal line of 5-in. standard pipe was used, approximately 42 ft. long, in which was located an extra-heavy flanged coupling for holding the orifice diaphragms. The heavy coupling was not required for strength, but was employed to allow the use of diaphragms adapted for high-pressure pipe and fittings. In Fig. 1 is a view of the pipe, flanges and manometer connections, and from the diagram in Fig. 4 it will be noted that the location of the diaphragm is 21 ft. 11 in. below an ell and 20 ft. 2 in. from the discharge end of the pipe, on which is screwed a cap having

a discharge orifice. Water was supplied to the pipe line by a 6-in. centrifugal pump direct-connected to a steam turbine. The water was measured by means of several large measuring bays.

9 The diaphragms were all formed from plates of Monel metal $\frac{1}{32}$ in. thick, with the outer rings of their surfaces, above 5.1 in. diameter, corrugated to form a tight joint between the flanges, without the use of gaskets. The orifices were bored out accurately to size and were of three types: (1) Concentric with the rim of the diaphragm; (2) eccentric with the rim; and (3) segmental.

FIG. 3 ORIFICE AT END OF 5-IN. PIPE WITH MANOMETER OR GAGE GLASS CONNECTION

10 There were eight concentric diaphragms and two each of the eccentric and segmental diaphragms. The concentric diaphragms had orifices ranging from 4.5 in. diameter to 1 in. diameter, by steps of $\frac{1}{2}$ in., making a series in terms of pipe diameter of 90, 80, 70, 60, 50 40, 30 and 20 per cent diameter sizes. The eccentric and segmental diaphragms were in two sizes, having orifices equal in area to the 90 per cent and 80 per cent concentric sizes mentioned above.

11 The eccentric diaphragms were made with the edge of the orifice just flush on one side with the inner surface of the pipe. The orifices of the segmental diaphragms were of the same diameter as the

inside of the pipe, but had strips inserted which were so located as to produce segmental openings equal in area to the 90 per cent and 80 per cent sizes of the concentric diaphragms. The three types of diaphragms are shown in Fig. 2. These diaphragms were furnished by E. G. Bailey and are the standard form of orifice plate used in the Bailey fluid meters.

PIPE CAP ORIFICES

12 Pipe cap orifices were placed on the end of the 5-in. pipe (Fig. 3), to provide a means for regulating the water contents of the pipe below the diaphragm and so control more easily the pressure drop through the diaphragm. These orifices also gave an opportunity to secure data for computing the coefficients of discharge for this form of orifice. The pipe cap orifices varied in size, ranging from 4-in. diam-

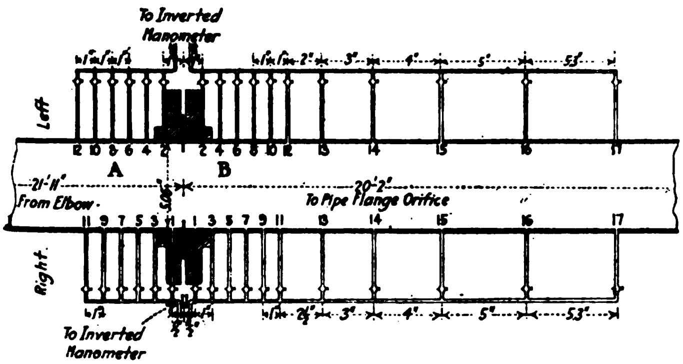


FIG. 4 POSITION OF DIAPHRAGM FLANGE AND POINTS OF PRESSURE CONNECTIONS

eter to 1-in. diameter by ½-in. intervals. No special refinements were practiced in finishing the caps except to bore and ream the orifices accurately to size and to smooth up the inside surfaces of the caps.

PRESSURE CONNECTIONS

13 Fig. 4 shows the locations of the pressure connections for the manometer, which are spaced along lines on diametrically opposite sides of the pipe in a horizontal plane with its axis.

14 Above the diaphragm, on the upstream side, there are six connections on each side of the pipe, and below the diaphragm there are 11 connections. It will be seen that on the right side of the pipe,

looking downstream, the pressure connections begin at a poi located $\frac{1}{2}$ in. from the plane of the diaphragm and that the first s connections are spaced 1 in. apart; while the remaining five poin on the downstream side are spaced respectively $2\frac{1}{2}$, 3, 4, 5 and $5\frac{3}{16}$ i apart, making the last one, No. 17, five pipe diameters from tl diaphragm plane.

15 The connections on the left side are similarly placed, exce that those nearest the diaphragm are located 1 in. from it, instead $\frac{1}{2}$ in., and that the five connections farthest away from the diaphrag are placed opposite those on the right side. These connections a designated by odd numbers on the right side and even numbers on tl left side, except that the five points farthest away from the diaphrag on both sides of the pipe are given numbers in order from 13 to 1 inclusive.

16 On the downstream side the connections were made by drillin and tapping holes for $\frac{1}{8}$-in. brass air cocks which were screwed into tl pipe until their ends were flush with the inside. These air cocks hav openings $\frac{3}{16}$ in. in diameter. A manifold formed with glass tees an heavy rubber tubing led from these pressure connections to one sid of a water manometer 7 ft. in length, the bottom of which is shown i Fig. 1. The glass connections made it possible to see whether th tubes were free from air pockets.

17 On the upstream side air cocks of the same size were use but the nipple connections leading to the air cocks were plugged an then drilled with holes of only $\frac{1}{8}$ in. diameter. The manifold for th upstream side was formed from a rigid brass bar to which the ai cocks from the pipe were connected by nipples and from which rubber tube led to the other side of the inverted water manomet mentioned above.

18 A second water gage glass (shown in Fig. 1 with a rubbe tubing leading to a connection at the bottom side of the pipe) wa used to indicate the head on the diaphragm and to aid in maintainin a more constant water head.

19 The complete manifold arrangement (see Fig. 4) made i possible to maintain a pressure difference, or drop, between any poin of connection above and any point below the diaphragm without dis turbing the gage glass which indicated the head on the upstream sid of the pipe.

20 The static head at the cap orifice was read at a point 7.5 ir from the plane of the orifice on a water column whose zero scale read ing was on a level with the center of the pipe.

PRESSURE TRAVERSES

21 A series of explorations of pressures was made along the horizontal element of the pipe on each side of the diaphragm, going upstream one pipe diameter and downstream five pipe diameters. In making these explorations, or longitudinal traverses, those on the downstream or B side were all made with the upstream connection open at A-5, or one-half pipe diameter above the diaphragm (excepting for the 90 per cent concentric diaphragm, when A-10 was open). When traversing the upstream or A side, the downstream connection was open at B-14 (excepting for the 90 per cent concentric diaphragm when B-10 was open). In this way the pressure changes in the vicinity of the diaphragm could be noted. Such traverses were made for four different pressure drops (approximately ½ ft., 2 ft., 4 ft. and 6 ft.), for the 90 to 50 per cent diaphragms inclusive; and for a 6-ft. pressure drop for the 40, 30 and 20 per cent diaphragms.

22 In general, the pressure-head conditions could be maintained with little variation by means of the centrifugal pump. Such variation was watched and controlled by a man stationed at a valve in the supply pipe line some 50 ft. distant from the diaphragm.

23 During the calibration of the diaphragms and orifices the water was measured volumetrically in a calibrated cistern, or bay, of about 200 sq. ft. cross-section and 12,000 gal. capacity. A series of deflector pipes was used so that the water could be shifted into or from the measuring bay as desired.

PROCEDURE IN MAKING A TRAVERSE

24 The usual procedure was to bring the static pressure above the diaphragm to the desired height and when conditions had reached a constant state and the manifold connections were found to be free from air a traverse was made of the pressure drop through the diaphragm between the fixed point A-5 (½ pipe diameter upstream) and each point in succession on the downstream side, taking the odd numbered points first and then following with the even numbered pressure points.

25 A fixed reference point B-14 was then taken on the downstream side, and the upstream reading was explored, to get the pressure variation above the diaphragm. Each time a reading was taken the inverted manometer column was allowed to assume its state of equilibrium. The height of the water in the two lengths

of the manometer was read simultaneously by two observers at a given signal. The whole traverse required from 10 to 15 minutes.

CALIBRATION OF ORIFICES

26 The flow of water was also measured, using these same diaphragms and orifices for the same pressure conditions as were maintained during the so-called traverses. In most cases, though not always, a calibration either followed a traverse or immediately preceded it so as to have flow conditions the same. The length of the runs for the calibrations ranged from 9 to 30 minutes, depending on the quantity of water discharged by the orifices. The readings were taken in many cases every minute of the pressure drop between A-5 and B-5, A-1 and B-1, and also of the head at the pipe orifice. These readings were taken in rotation and formed entirely independent observations. For the large-sized diaphragms more time was needed to take each individual reading, hence fewer readings in all could be taken. The time readings were all taken on a stop watch whose accuracy was checked.

DATA SECURED FROM TESTS

27 Only one set of data is included in this paper, since the same scheme was pursued in getting the other sets of readings. Table 1 shows the data taken for the 4-in. diaphragms, or the 80 per cent sizes. This includes, besides the data for the concentric orifice, or diaphragm, those for the eccentric and segmental diaphragms. Table 2 shows the observations during the calibrations of these same diaphragms, taken for the quantity of water discharged at the same or nearly the same rates of flow. In this table are included the pressure drops between points A-5 and B-5; between A-1 and B-1 and also the head readings at the pipe orifice; and the cubic feet of water discharged per run, together with the quantity in cubic feet per second.

28 The data so taken were made use of in two ways: (*a*) the observations taken during a traverse were plotted on coördinate paper to show the pressure distribution at the diaphragm, and (*b*) from the calibration data the coefficients of discharge for the diaphragms and pipe cap orifices were computed.

PRESSURE DROP CURVES

29 The curves plotted in Figs. 5 to 11 inclusive show the pressure drop variation for all the diaphragms. The horizontal scale was

TABLE 1 OBSERVED DATA

Run No.	Static Gage at A	Drop in Pressure, Feet Water, at Position Shown									
		At A	At B	Manometer			At A	At B	Manometer		
				left	right	total ft.			left	right	total ft.
		4-in. Concentric Diaphragm with 4-in. Pipe Orifice									
18	6.0	A5	B1	−2.23	+1.32	3.55	A5	B2	−2.33	+1.36	3.69
			3	2.30	1.33	3.63		4	2.33	1.36	3.69
			5	2.24	1.31	3.55		6	2.22	1.28	3.50
			7	2.10	1.20	3.30		8	1.94	1.07	3.01
			9	1.79	1.00	2.79		10	1.71	0.90	2.61
			11	1.58	0.82	2.40		12	1.55	0.77	2.32
			13	1.28	0.58	1.86		13	1.30	0.58	1.88
			14	1.15	0.46	1.61		14	1.15	0.45	1.60
			15	1.06	0.40	1.46		15	1.07	0.38	1.45
			16	1.05	0.38	1.43		16	1.01	0.34	1.35
			17	−1.04	+0.40	1.44		17	−1.03	+0.35	1.38
	6.0										
	6.0	A1	B14	−1.27	+0.57	1.84	A2	B14	−1.22	+0.52	1.74
		3		1.18	0.47	1.65	4		1.17	0.46	1.63
		5		1.16	0.43	1.59	6		1.15	0.44	1.59
		7		1.15	0.42	1.57	8		1.15	0.45	1.60
		9		1.12	0.39	1.51	10		1.16	0.46	1.62
	6.0	11		−1.14	+0.42	1.56	12		−1.13	+0.45	1.58
		4-in. Eccentric Diaphragm with 4-in. Pipe Orifice									
22	6.0	A5	B1	−1.98	+1.96	3.94	A5	2	−1.33	+1.43	2.76
			3	2.04	2.01	4.05		4	1.58	1.65	3.23
			5	2.12	2.06	4.18		6	1.60	1.65	3.25
			7	2.06	2.01	4.07		8	1.72	1.78	3.50
			9	1.96	1.93	3.89		10	1.80	1.67	3.47
			11	1.82	1.86	3.68		12	1.58	1.68	3.26
			13	1.51	1.54	3.05		13	1.50	1.56	3.06
			14	1.20	1.32	2.52		14	1.18	1.32	2.48
			15	1.01	1.16	2.17		15	1.02	1.08	2.10
			16	0.86	1.05	1.91		16	1.88	1.07	1.95
	6.0		17	−0.83	+1.03	1.86		17	−0.83	+1.06	1.89
	6.0	A1	B14	−1.21	+1.35	2.56	A2	B14	−0.80	+0.81	1.61
		3		1.20	1.35	2.55	4		0.92	1.01	1.93
		5		1.11	1.23	2.34	6		0.92	1.01	1.93
		7		1.10	1.16	2.26	8		1.07	1.17	2.24
		9		1.04	1.12	2.16	10		1.07	1.16	2.23
	6.0	11		−1.05	+1.16	2.21	12		−1.13	+1.21	2.34
		4-in. Segmental Diaphragms with 4-in. Pipe Orifice									
26		A5	B1	−2.40	+1.97	4.37	A5	B2	−1.53	+1.35	2.88
			3	2.44	2.00	4.44		4	1.92	1.63	3.55
			5	2.49	2.05	4.54		6	2.01	1.67	3.68
			7	2.47	2.05	4.54		8	2.14	1.81	3.95
			9	2.50	2.07	4.57		10	2.12	1.79	3.91
			11	2.45	2.00	4.45		12	2.16	1.80	3.96
			13	2.13	1.80	3.93		13	2.05	1.70	3.75
			14	1.69	1.42	3.11		14	1.66	1.43	3.09
			15	1.36	1.22	2.38		15	1.40	1.25	2.65
			16	1.21	1.09	2.30		16	1.20	1.05	2.25
			17	−1.15	+1.03	2.28		17	−1.17	+1.02	2.19
		A1	B14	−1.56	+1.35	2.91	A2	B14	−1.02	+0.60	1.62
		3		1.63	1.44	3.07	4		1.26	0.95	2.21
		5		1.64	1.40	3.04	6		1.30	1.02	2.32
		7		1.60	0.32	1.92	8		1.38	1.12	2.50
		9		1.55	1.23	2.78	10		1.41	1.18	2.59
		11		−1.55	+1.25	2.80	12		−1.41	+1.18	2.59

laid off in pipe diameter distances and is also marked at the proper points showing the positions along the pipe where the pressure readings were taken. The vertical scale is laid off to represent pressure drop through the diaphragm in feet.

TABLE 2 OBSERVED DATA

Calibration of 4-in. Concentric Diaphragm and 4-in. Pipe Orifice

Run No.	Time, Min.	Head At orifice, ft.	Head At A, ft.	Manometer A5 to B5 left	Manometer A5 to B5 right	Manometer A5 to B5 total	Manometer A1 to B1 left	Manometer A1 to B1 right	Manometer A1 to B1 total	Volume Scale	Cu. Ft.	c.f.s.
16	0	3.77	6.15	−2.36	+1.10	3.46	−2.36	+1.10	3.46	0.8	−23	
		3.78	6.15	2.40	1.12	3.52	2.38	1.08	3.46			
	3	3.78	6.15	2.41	1.09	3.50	2.36	1.06	3.42	1.84		
				2.38	1.05	3.43	2.37	1.02	3.49			
	6	3.71	6.10	2.43	1.07	3.50	2.39	1.01	3.40	2.90		
				2.43	1.07	3.50	2.41	1.03	3.44			
	9	3.71	6.10	−2.50	+1.03	3.53	−2.41	+0.96	3.37	2.95	574	
		3.77										
		3.748				3:50			3.47	total	597	1.105
Calibration of 4-in. Eccentric Diaphragm and 4-in. Pipe Orifice												
20	0	3.60	6.00	−2.07	+2.03	4.10	−2.00	+1.95	3.95	4.93	778	
				2.02	2.00	4.02	2.00	2.00	4.00			
				2.04	1.99	4.03	2.02	1.98	4.00	5.95		
	3	3.52	6.00	2.13	2.01	4.14	2.04	1.96	4.00			
				2.02	1.92	3.94	2.01	1.94	3.95	6.98		
	6	3.55	5.95	2.10	1.95	4.05	2.04	1.93	3.97			
	8.58	3.55	5.95	−2.15	+1.97	4.12	−2.00	+1.90	3.90	8.00	1351	
	average	3.555				4.057			3.967	total	573	1.065
Calibration of 4-in. Segmental Diaphragm and 4-in. Pipe Orifice												
24	0	3.48	5.90	−2.58	+2.01	4.59	−2.40	+1.83	4.23	0.70	−42	
				2.60	2.00	4.60	2.38	1.82	4.20			
	3	3.44	5.90	2.55	1.95	4.50	2.36	1.83	4.19	1.70		
		3.52	6.00	2.55	1.97	4.52	2.38	1.84	4.22			
	6	3.50	5.95	2.60	1.99	4.59	2.38	1.85	4.23	2.725		
		3.50	5.95	2.54	1.96	4.50	−2.41	+1.87	4.28			
	9	3.52	6.00	−2.56	+2.00	4.56				3.74	535	
	average	3.49				4.55			4.22	total	577	1.068

30 To illustrate: Run No. 18, Table No. 1 (4.022-in. concentric diaphragm) gives the readings from which curve No. 18 (Fig. 5) was

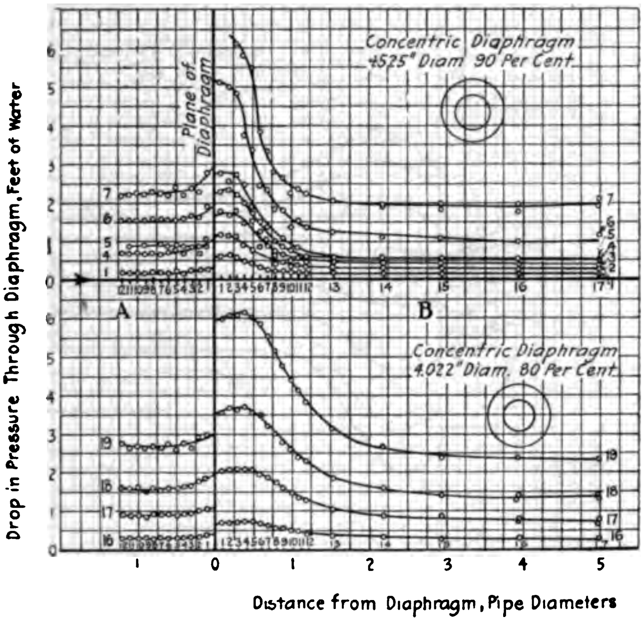


FIG. 5 80 AND 90% CONCENTRIC DIAPHRAGMS

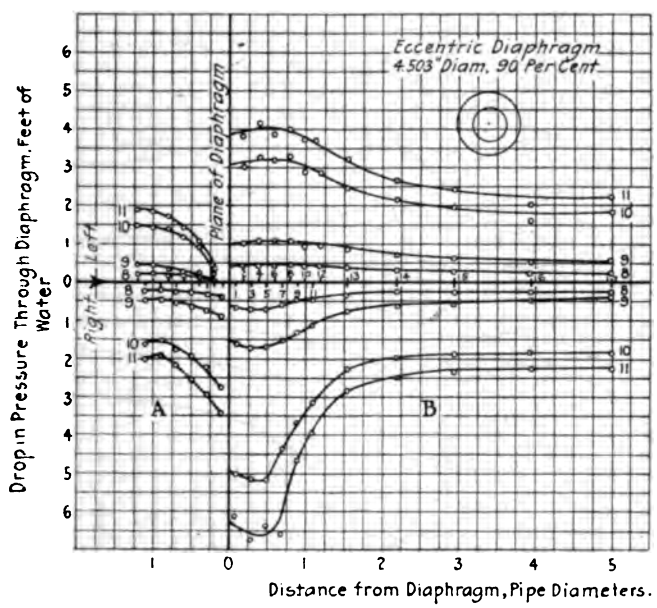


FIG. 6 90% ECCENTRIC DIAPHRAGM

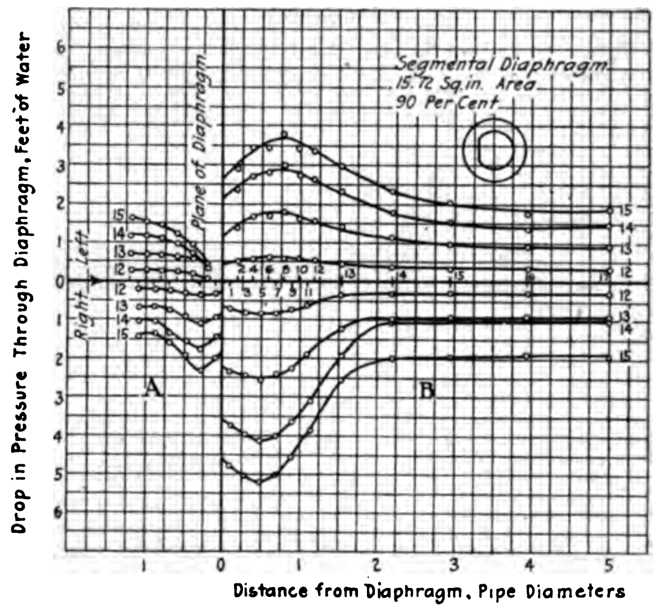
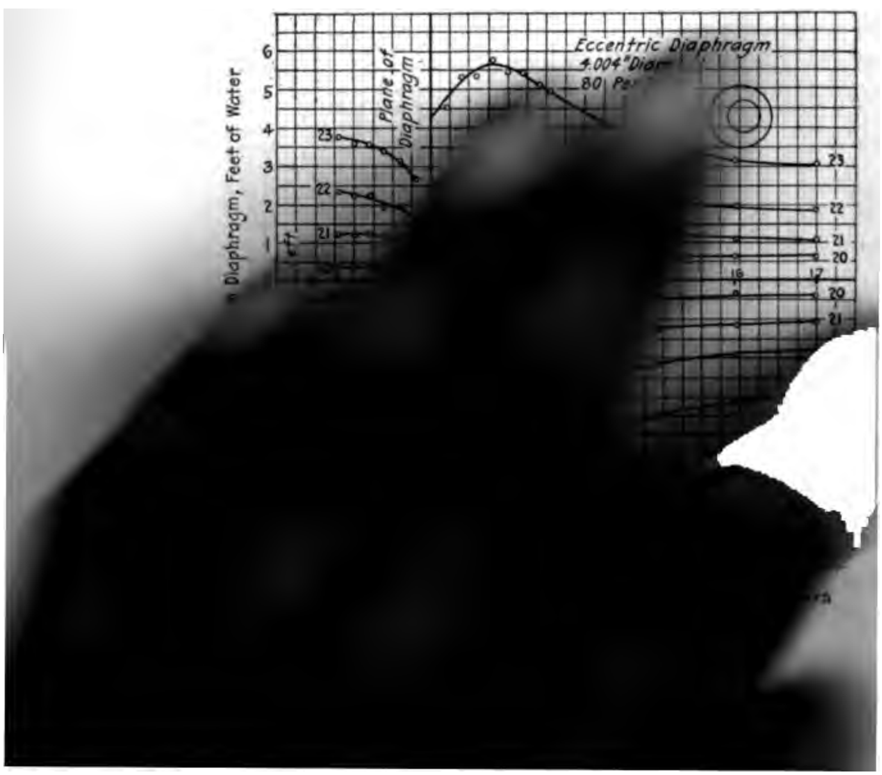


FIG. 7 90% SEGMENTAL DIAPHRAGM



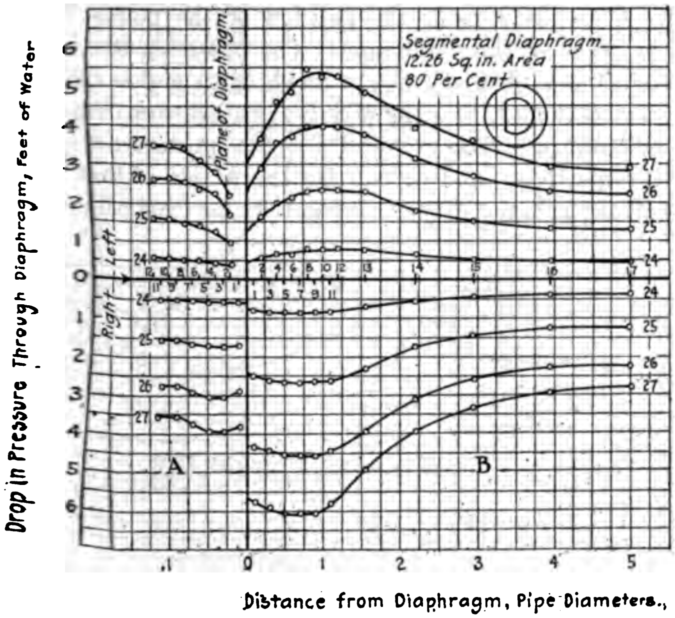


FIG. 9 80% SEGMENTAL DIAPHRAGM

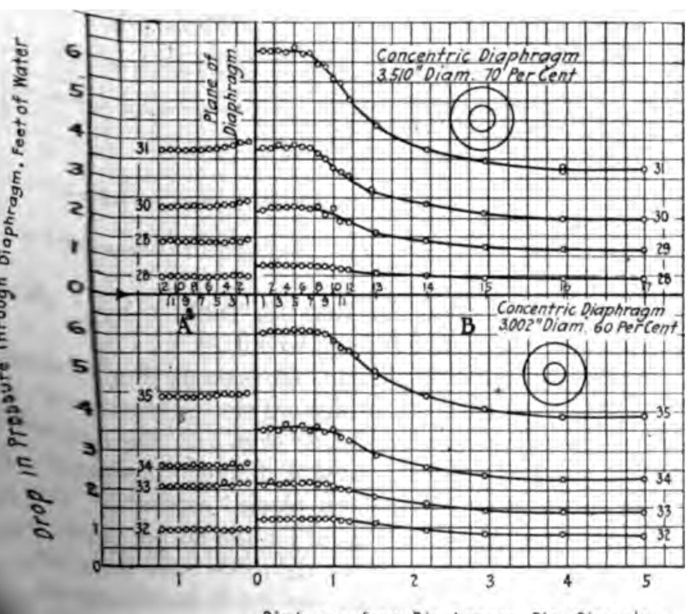


FIG 10 60 AND 70% CONCENTRIC DIAPHRAGMS

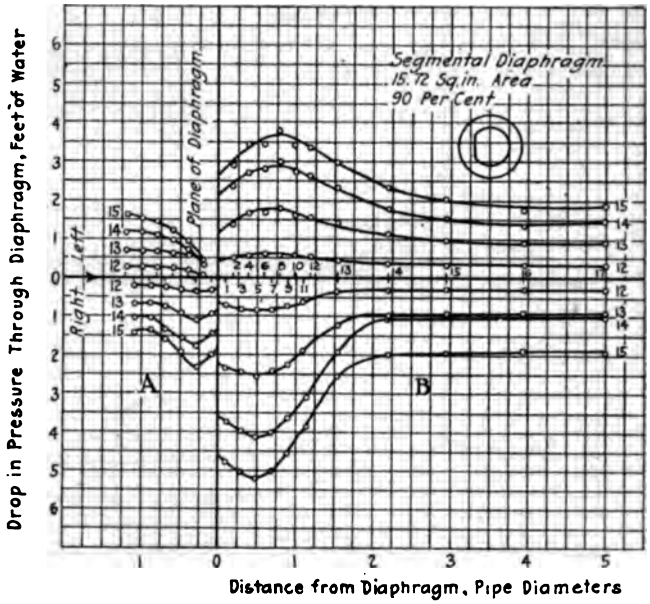


FIG. 7 90% SEGMENTAL DIAPHRAGM

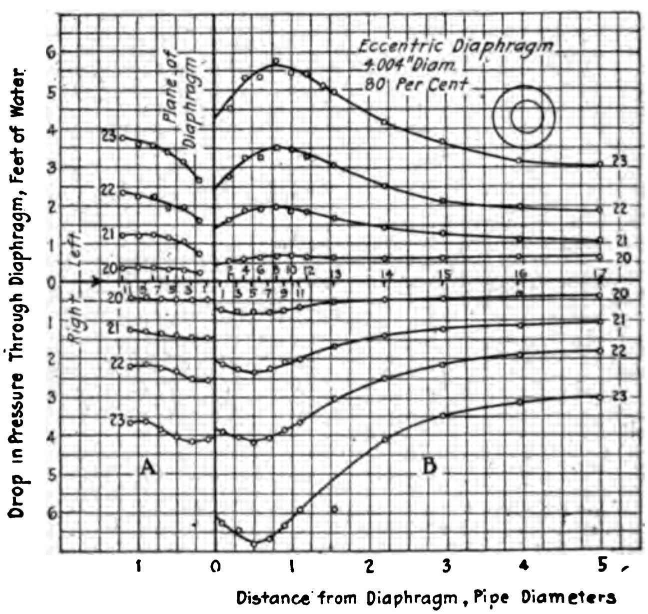


FIG. 8 80% ECCENTRIC DIAPHRAGM

TABLE 1 OBSERVED DATA

Run No.	Static Gage at A	Drop in Pressure, Feet Water, at Position Shown									
		At A	At B	Manometer left	Manometer right	Manometer total ft.	At A	At B	Manometer left	Manometer right	Manometer total ft.
		4-in. Concentric Diaphragm with 4-in. Pipe Orifice									
18	6.0	A5	B1	−2.23	+1.32	3.55	A5	B2	−2.33	+1.36	3.69
			3	2.30	1.33	3.63		4	2.33	1.36	3.69
			5	2.24	1.31	3.55		6	2.22	1.28	3.50
			7	2.10	1.20	3.30		8	1.94	1.07	3.01
			9	1.79	1.00	2.79		10	1.71	0.90	2.61
			11	1.58	0.82	2.40		12	1.55	0.77	2.32
			13	1.28	0.58	1.86		13	1.30	0.58	1.88
			14	1.15	0.46	1.61		14	1.15	0.45	1.60
			15	1.06	0.40	1.46		15	1.07	0.38	1.45
			16	1.05	0.38	1.43		16	1.01	0.34	1.35
			17	−1.04	+0.40	1.44		17	−1.03	+0.35	1.38
	6.0										
	6.0	A1	B14	−1.27	+0.57	1.84	A2	B14	−1.22	+0.52	1.74
		3		1.18	0.47	1.65	4		1.17	0.46	1.63
		5		1.16	0.43	1.59	6		1.15	0.44	1.59
		7		1.15	0.42	1.57	8		1.15	0.45	1.60
		9		1.12	0.39	1.51	10		1.16	0.46	1.62
	6.0	11		−1.14	+0.42	1.56	12		−1.13	+0.45	1.58
		4-in. Eccentric Diaphragm with 4-in. Pipe Orifice									
22	6.0	A5	B1	−1.98	+1.96	3.94	A5	2	−1.33	+1.43	2.76
			3	2.04	2.01	4.05		4	1.58	1.65	3.23
			5	2.12	2.06	4.18		6	1.60	1.65	3.25
			7	2.06	2.01	4.07		8	1.72	1.78	3.50
			9	1.96	1.93	3.89		10	1.80	1.67	3.47
			11	1.82	1.86	3.68		12	1.58	1.68	3.26
			13	1.51	1.54	3.05		13	1.50	1.56	3.06
			14	1.20	1.32	2.52		14	1.18	1.32	2.48
			15	1.01	1.16	2.17		15	1.02	1.08	2.10
			16	0.86	1.05	1.91		16	1.88	1.07	1.95
	6.0		17	−0.83	+1.03	1.86		17	−0.83	+1.06	1.89
	6.0	A1	B14	−1.21	+1.35	2.56	A2	B14	−0.80	+0.81	1.61
		3		1.20	1.35	2.55	4		0.92	1.01	1.93
		5		1.11	1.23	2.34	6		0.92	1.01	1.93
		7		1.10	1.16	2.26	8		1.07	1.17	2.24
		9		1.04	1.12	2.16	10		1.07	1.16	2.23
	6.0	11		−1.05	+1.16	2.21	12		−1.13	+1.21	2.34
		4-in. Segmental Diaphragms with 4-in. Pipe Orifice									
26		A5	B1	−2.40	+1.97	4.37	A5	B2	−1.53	+1.35	2.88
			3	2.44	2.00	4.44		4	1.92	1.63	3.55
			5	2.49	2.05	4.54		6	2.01	1.67	3.68
			7	2.47	2.05	4.54		8	2.14	1.81	3.95
			9	2.50	2.07	4.57		10	2.12	1.79	3.91
			11	2.45	2.00	4.45		12	2.16	1.80	3.96
			13	2.13	1.80	3.93		13	2.05	1.70	3.75
			14	1.69	1.42	3.11		14	1.66	1.43	3.09
			15	1.36	1.22	2.38		15	1.40	1.25	2.65
			16	1.21	1.09	2.30		16	1.20	1.05	2.25
			17	−1.15	+1.03	2.28		17	−1.17	+1.02	2.19
		A1	B14	−1.56	+1.35	2.91	A2	B14	−1.02	+0.60	1.62
		3		1.63	1.44	3.07	4		1.26	0.95	2.21
		5		1.64	1.40	3.04	6		1.30	1.02	2.32
		7		1.60	0.32	1.92	8		1.38	1.12	2.50
		9		1.55	1.23	2.78	10		1.41	1.18	2.59
		11		−1.55	+1.25	2.80	12		−1.41	+1.18	2.59

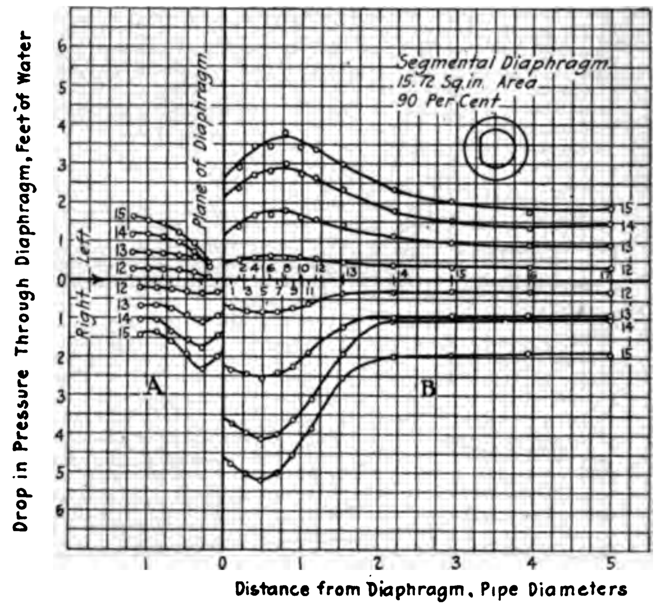


FIG. 7 90% SEGMENTAL DIAPHRAGM

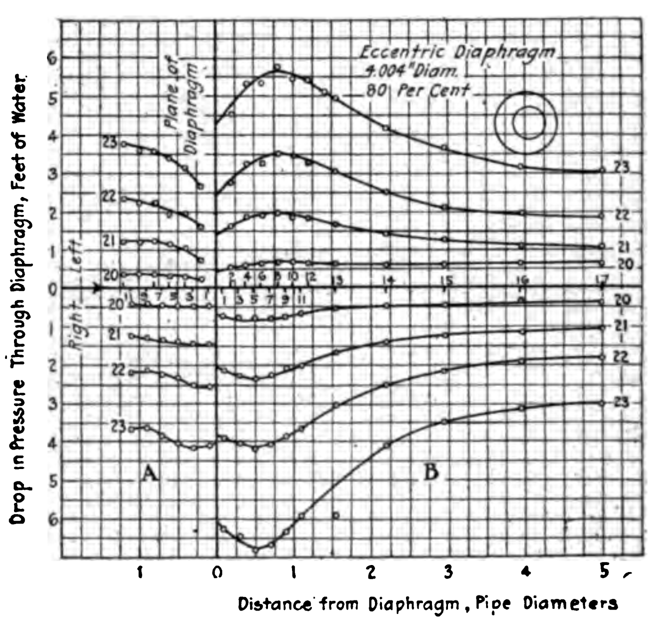


FIG. 8 80% ECCENTRIC DIAPHRAGM

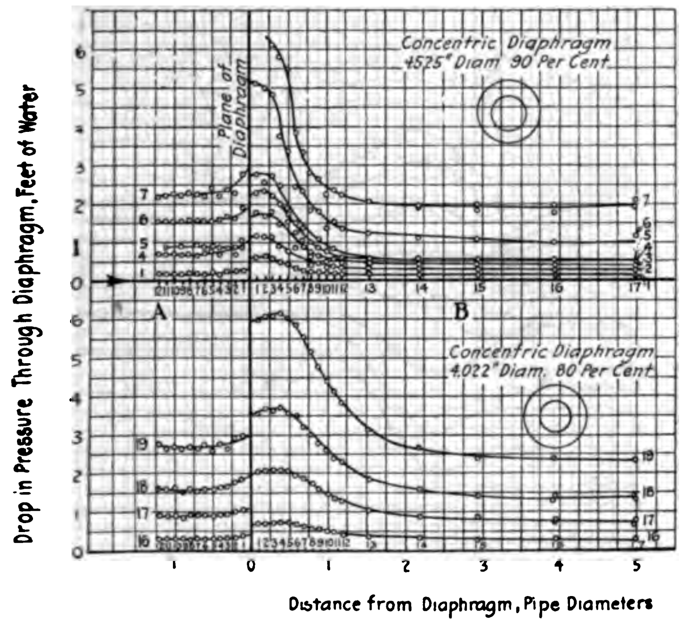


Fig. 5 80 and 90% Concentric Diaphragms

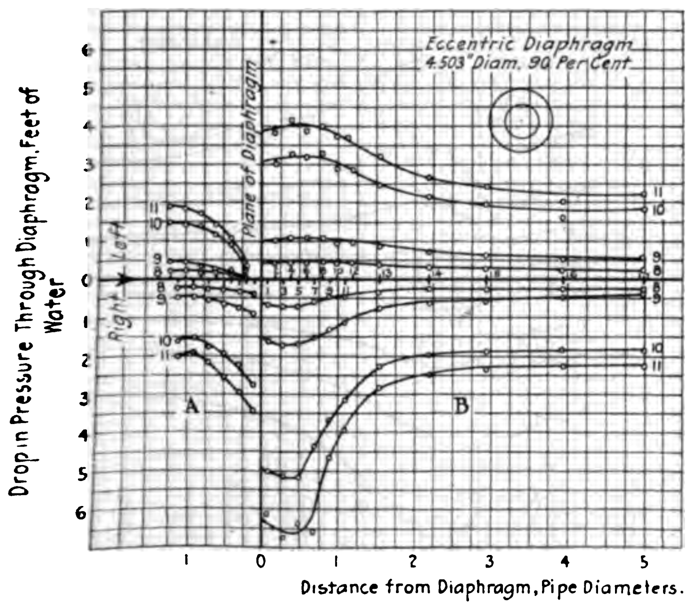


Fig. 6 90% Eccentric Diaphragm

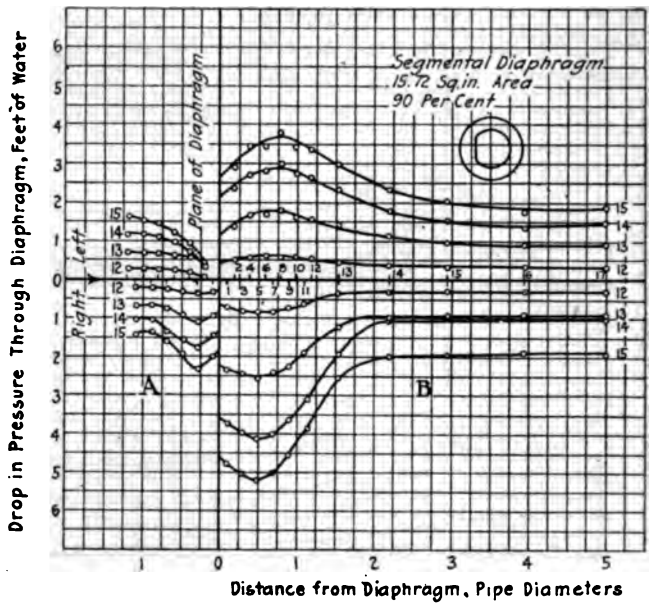


FIG. 7 90% SEGMENTAL DIAPHRAGM

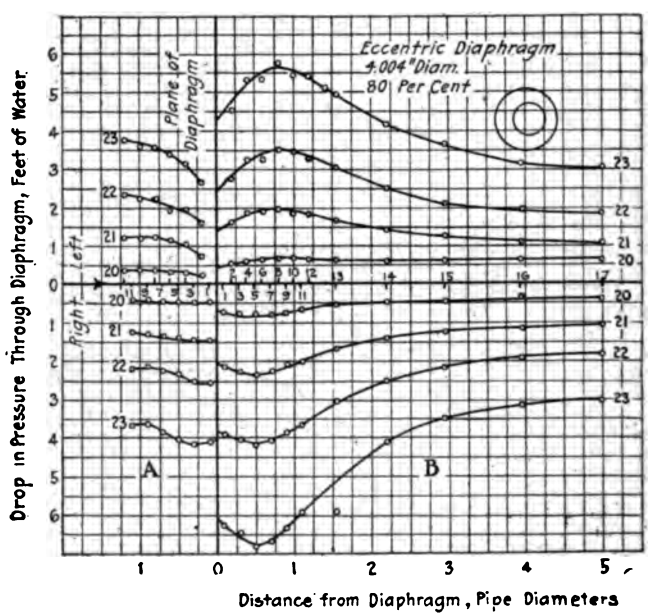


FIG. 8 80% ECCENTRIC DIAPHRAGM

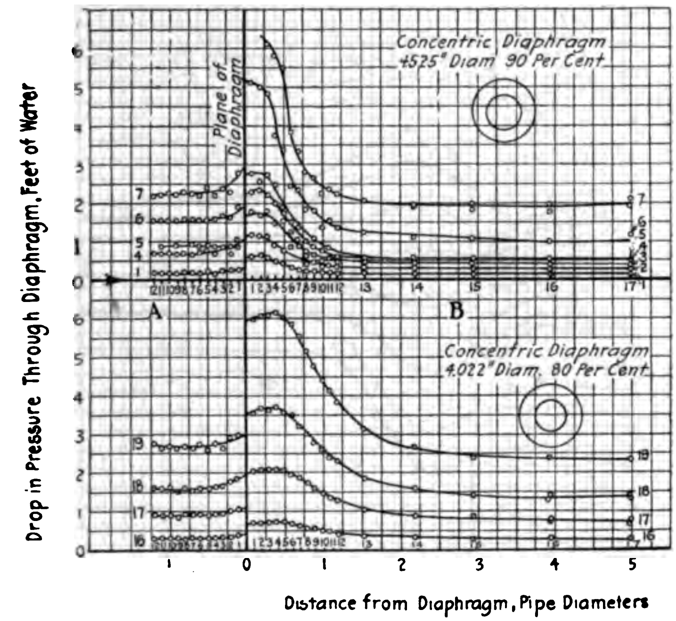


Fig. 5 80 and 90% Concentric Diaphragms

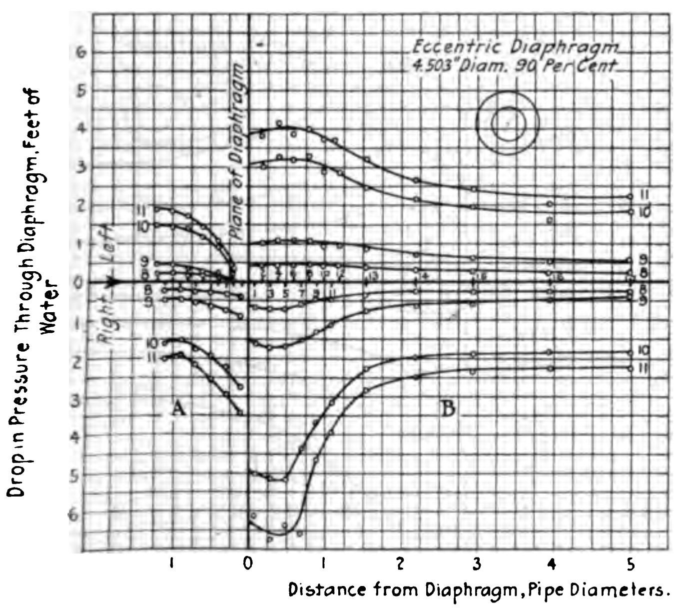


Fig. 6 90% Eccentric Diaphragm

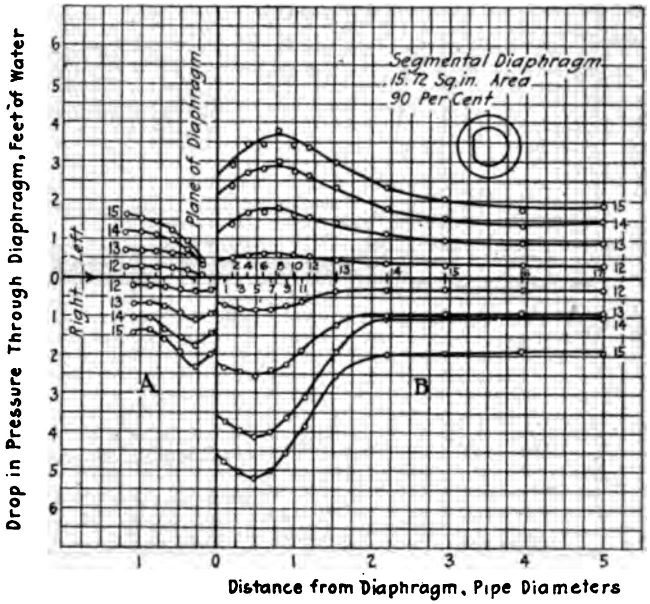


FIG. 7 90% SEGMENTAL DIAPHRAGM

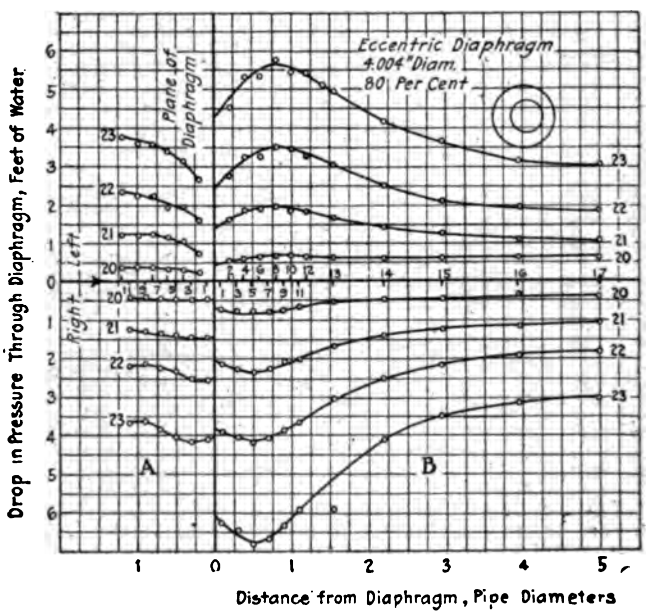


FIG. 8 80% ECCENTRIC DIAPHRAGM

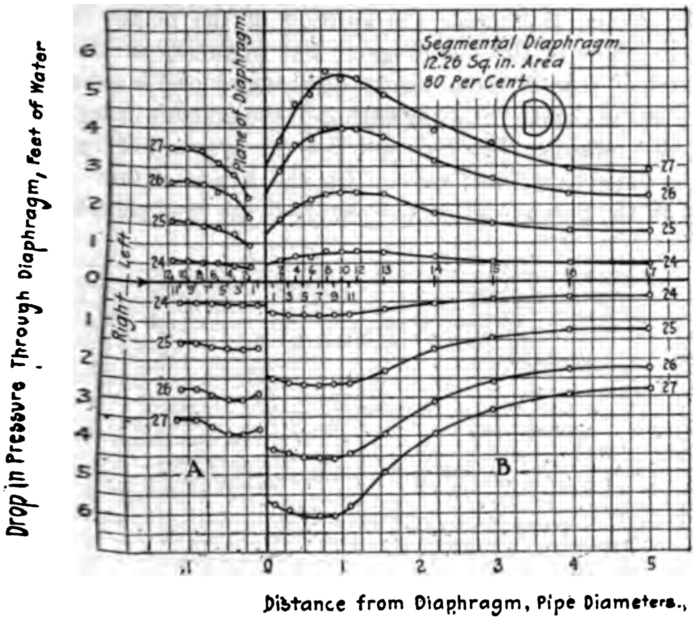


FIG. 9 80% SEGMENTAL DIAPHRAGM

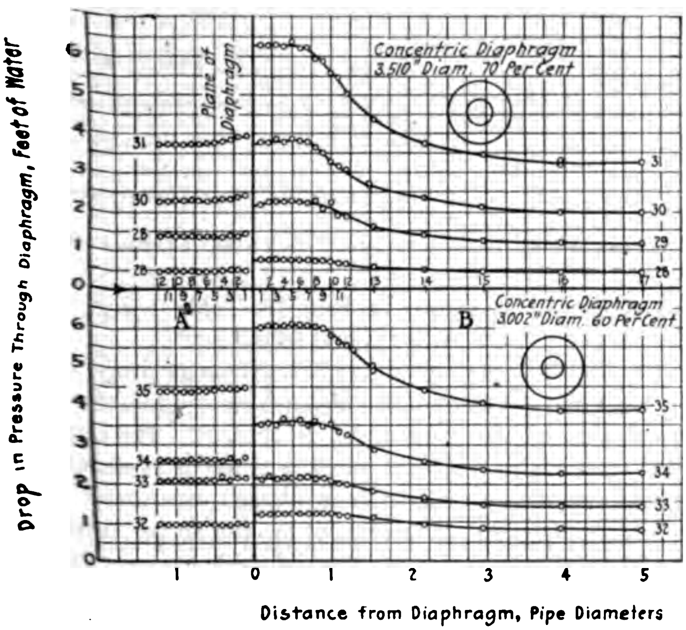


FIG 10 60 AND 70% CONCENTRIC DIAPHRAGMS

plotted. A and B indicate the location of the points above ε
below the plane of the diaphragm. The upstream connection ·
A-5 (one-half pipe diameter above diaphragm) and is considerec
a zero pressure point. Starting with B-1, the pressure drop of 3.5ξ
is laid off upwards. For B-3 and B-5 the pressure drops are ξ
and 3.55 respectively, and so on. On the A side of the diaphra
are laid off the pressure drops between B-14 (as fixed point) and e

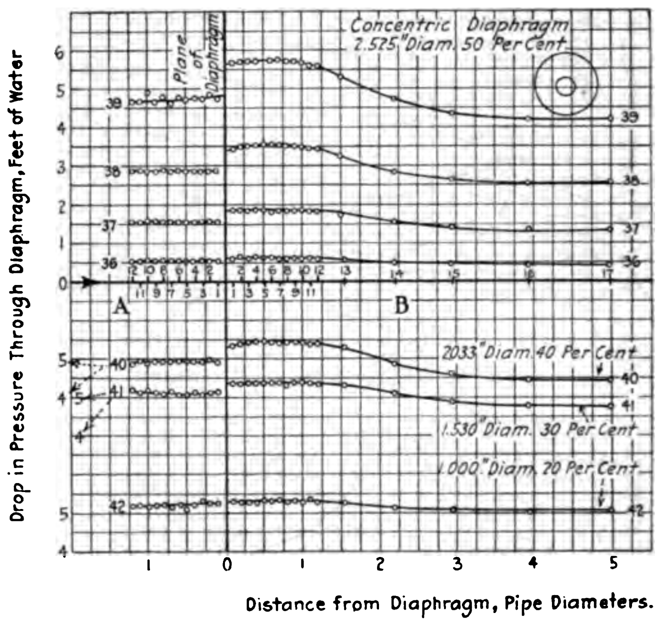


FIG. 11 50, 40, 30 AND 20% CONCENTRIC DIAPHRAGMS

point in succession as shown in Run 18, Table 1, by A-1 = 1
ft., A-3 = 1.65 ft., A-5 = 1.59 ft., and so on.

31 For the concentric diaphragms the pressure drop readings
both the right and left sides of the pipe are plotted together and fc
a single curve, as shown by curve No. 18 (Fig. 5). The eccentric ε
segmental diaphragms, however, have a different pressure drop
each side of the pipe and have been plotted separately as may
seen in curves Nos. 20 to 23 (Fig. 8) for the 80 per cent eccen
diaphragms, where the curves for the right side have been placed
low and those for the left side of the pipe are placed above the cer
line of the curve sheet.

32 The curves for the A side of the diaphragm have been plotted as read from the data, and, since A-5 was taken as a zero point for the

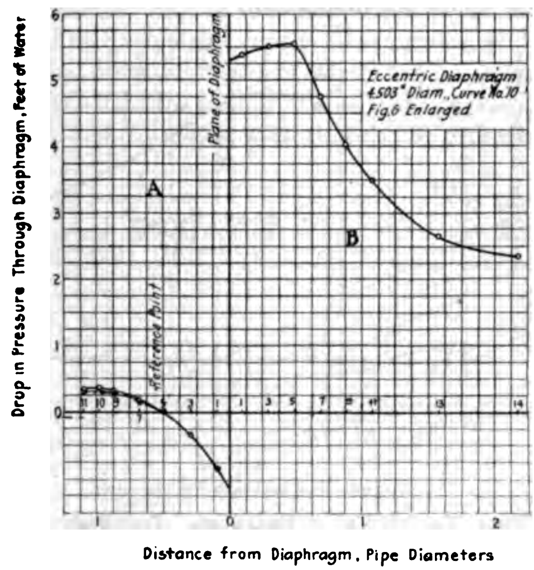


FIG. 12 FIG. 6 ENLARGED. ECCENTRIC DIAPHRAGM

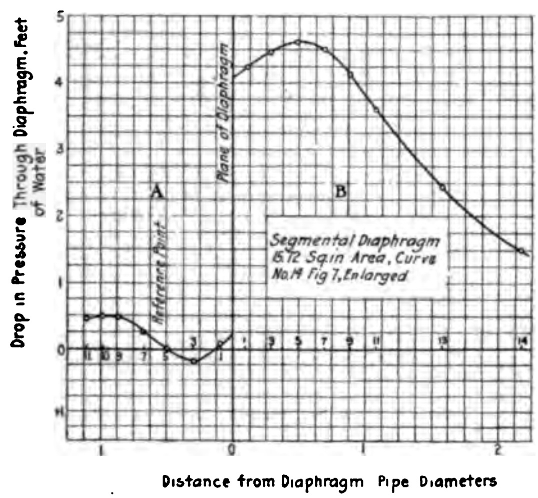


FIG. 13 FIG. 7 ENLARGED. ECCENTRIC DIAPHRAGM

downstream side of the diaphragm, all points on the A side should start at A-5 as zero; but if so plotted, it was feared that a confusion of curves would result, hence they are separated on the diagram and are intended to show only a relative change in pressure and not the absolute change above the diaphragm.

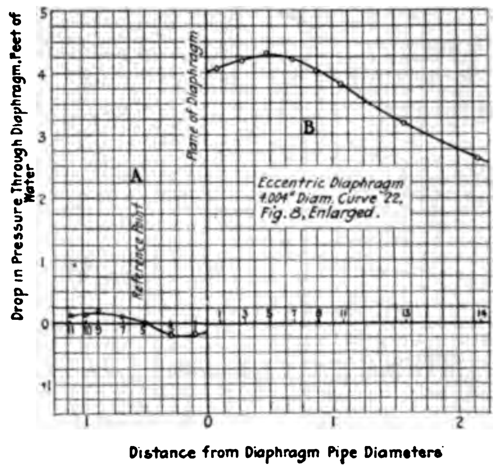


FIG. 14 CURVE NO. 22, FIG. 8 ENLARGED. ECCENTRIC DIAPHRAGM

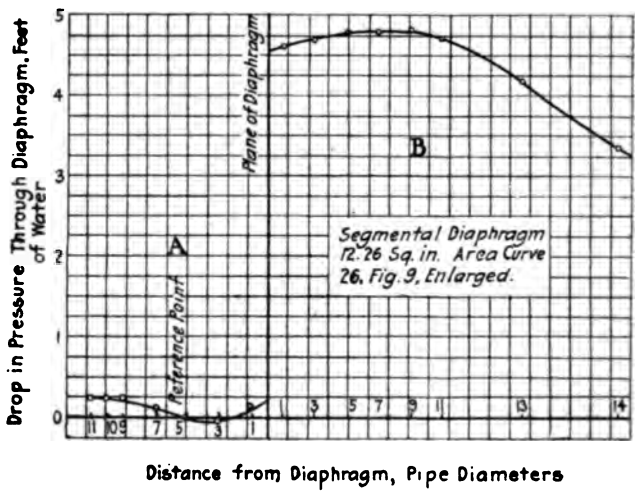


FIG. 15 CURVE NO. 26, FIG. 9 ENLARGED. SEGMENTAL DIAPHR

33 For the special diaphragms, viz. the 80 and 90 p eccentric and segmental, enlarged curves have been plotted fr observed data (Figs. 12 to 15), on which the readings on th have been corrected so as to start from A-5 as the zero

point and the ordinates on these curves will show the actual pressure difference between any two points in the vicinity of the diaphragm.

34 For example: Fig. 14 (plotted from Run 22, Table 1), giving curve No. 22 (Fig. 8) enlarged, for the 80 per cent eccentric diaphragm, shows the true pressure relations when A-5 is taken as the zero point. From the data for Run 22, Table 1, the pressure drop between A-5 and B-5 is 4.18 ft. which is plotted as the ordinate at B-5 with

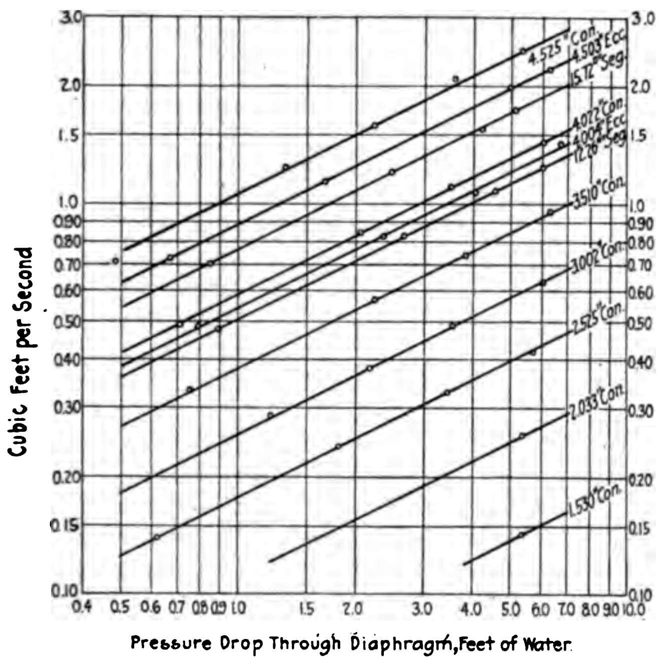


FIG. 16 DISCHARGE CURVES IN CU. FT. PER SEC. FOR DIAPHRAGM IN 5-IN. WATER PIPE

the point A-5 at zero. The other points on the B side are plotted directly from the observed readings and the remaining points on the A side are plotted as differences between the observed readings and the reading A-5.

FORMULA FOR COEFFICIENT OF DISCHARGE

35 In computing the coefficients of discharge, the pipe orifices were considered as frictionless or thin plate orifices having a velocity of approach. Hence the total head acting on the diaphragm will be:

$$\text{Total head} = \text{Static head} + \text{velocity head}$$

$$h = \frac{V^2}{2g} = H + \frac{v^2}{2g} = H + \frac{V^2}{2g}\left(\frac{A_2}{A_1}\right)^2 \quad \ldots\ldots$$

Since $Q = A_1 v = A_2 V$

$$v^2 = \left(\frac{A_2}{A_1}\right)^2 V^2$$

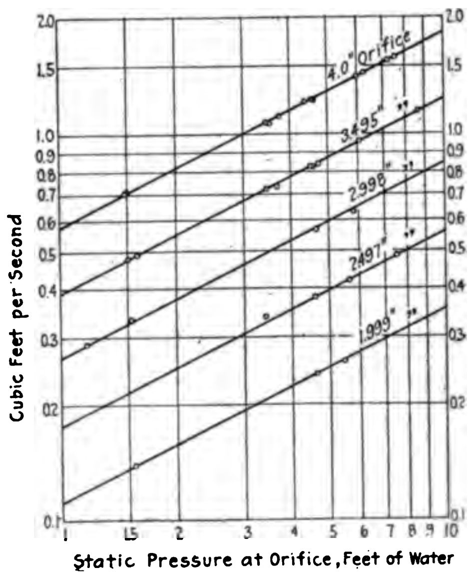


FIG. 17 DISCHARGE CURVES IN CU. FT. PER SEC. FOR ORIFICES 5-IN. WATER PIPE

Then from [1]

$$\left[1 - \left(\frac{A_2}{A_1}\right)^2\right] V^2 = 2gH$$

$$V = \sqrt{2gH} \div \sqrt{1 - \left(\frac{A_2}{A_1}\right)^2} \quad \ldots\ldots$$

Introducing coefficient of discharge K

$$q = \frac{KA_2\sqrt{2gH}}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}}$$

$$K = \frac{q\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}}{A_2\sqrt{2gH}} = \frac{q}{A_2\sqrt{2gH}}F$$

In the above formulæ:

h = total acting head on the diaphragm
V = velocity through diaphragm
v = velocity through pipe
A_2 = area of diaphragm
A_1 = area of pipe
q = actual discharge through diaphragm
Q = theoretical discharge through diaphragm
K = coefficient of discharge through diaphragm
F = factor of correction for velocity of approach

$$= \sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}$$

H = drop through diaphragm or orifice.

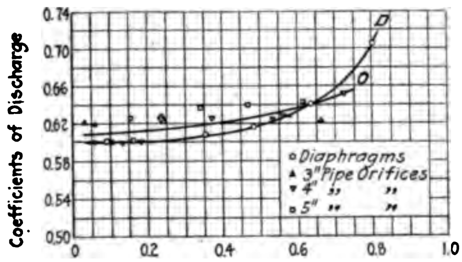


FIG. 18 COEFFICIENTS OF DISCHARGE FOR PIPE ORIFICES

36 No attempt in the formula has been made to correct for the contraction of the stream as has been done by some experimenters[1] in hydraulics, since it is believed that the simpler formula will best serve the purpose in finding the coefficients and in their application.

37 *The factor of correction* in the formula may be read directly from the factor of correction curve for velocity of approach in Fig. 20.

38 For the diaphragm the drop in pressure was taken at two points, the first at A-5, B-5; the second at A-1, B-1. This was done so as to compare coefficients of discharge for a point close to the diaphragm with a point some distance away from the diaphragm and which is a better point at which to make an attachment.

39 Curves have also been plotted on logarithmic coördinate paper between the cubic-feet-second discharge and the pressure drop through the diaphragms (see Fig. 16). Similar curves are also given for the 5-in. pipe cap orifices (see Fig. 17).

[1] Proc. Inst. Civil Engineers, vol. 197, p. 243, "The Diaphragm Method of Measuring the Velocity of Fluid-Flow in Pipes," by Holbrook Gaskell, Jr.

40 *The coefficients of discharge* for the diaphragms and the 5-in. pipe cap orifices are to be found in Tables 3 and 4.

41 Other coefficients of discharge for pipe cap orifices are given in Tables 5 and 6. These coefficients are computed from results obtained from experiments made by Prof. R. S. King, Mem. Am. Soc.M.E., University of Arizona, and the writer some years ago. They are given for the purpose of comparison with the results obtained from the 5-in. pipe orifices.

42 These results, which were obtained from the calibrations of orifices attached to 3-in. and 4-in. water pipes, have been used re-

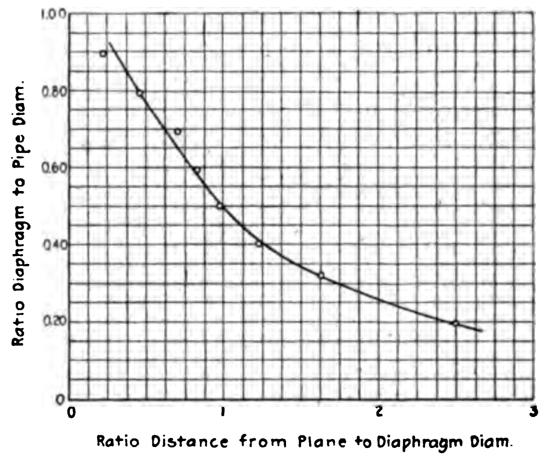


FIG. 19 PROBABLE POSITION OF LEAST SECTION AS SHOWN BY LOCATION OF GREATEST PRESSURE DROP THROUGH CONCENTRIC DIAPHRAGMS

peatedly since then and together with the discharge curve sheets have been found of great value in measuring quantities of water up to 1000 gal. per min. The curve sheets are given in Figs. 21 and 22 with the thought that other members of the Society may also find them useful.

43 A comparison of the coefficients of discharge for the series of pipe orifices, including the diaphragms and the pipe cap orifices, can be noted in Fig. 18, and also in Table 7. The coefficients for the 5-in. pipe orifices, including both the diaphragm orifices and the pipe orifice, show a more uniform variation through the range of head maintained than is found for the coefficients as computed for the 3-in. and 4-in. pipe cap orifices. This uniform variation is partly accounted for by the fact that more runs were made for the

TABLE 8 COEFFICIENTS OF DISCHARGE FOR DIAPHRAGMS

Run			Diaphragm					Pressure Drop at			Coefficient $K = \frac{q\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}}{A_2\sqrt{2gH}}$		
No.	Length, Min.	No. of Readings Taken	Kind	Diam. In.	A_2, Area, Sq. Ft.	Ratio, A_2/A_1	$\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}$	H_5 Pressure Drop A-5, B-5 Ft.	H_1 Pressure Drop A-1, B-1 Ft.	Cu. Ft. Sec. q	$K5$	$K1$	K Computed from Curves
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	15	8	con-	4.525	0.1117	0.800	0.600	0.48	0.69	0.708	0.683	0.578	
2	9	4	cen-	4.525	0.1117	0.800	0.600	1.32	2.00	1.226	0.714	0.580	
3	12	6	tric	4.525	0.1117	0.800	0.600	2.24	3.40	1.573	0.704	0.571	
4	11	5		4.525	0.1117	0.800	0.600	3.62	5.46	2.085	0.733	0.597	
5	9	7		4.525	0.1117	0.800	0.600	5.34		2.446	0.718		
										av.	0.710	0.582	0.706
6	15	9	eccen-	4.503	0.1106	0.792	0.611	0.67	0.73	0.725	0.615	0.584	
7	20	10	tric	4.503	0.1106	0.792	0.611	1.66	1.87	1.136	0.608	0.572	
8	9	7		4.503	0.1106	0.792	0.611	4.98	5.64	1.970	0.607	0.571	
9	10	5		4.503	0.1106	0.792	0.611	6.26	6.89	2.195	0.603	0.575	
										av.	0.608	0.576	0.609
10	12	5	seg-		0.1092	0.783	0.622	0.85	0.78	0.701	0.540	0.562	
11	12	5	men-		0.1092	0.783	0.622	2.48	2.306	1.208	0.544	0.564	
12	15	8	tal		0.1092	0.783	0.622	4.21	3.73	1.554	0.537	0.570	
13	12	5			0.1092	0.783	0.622	5.11	4.72	1.728	0.541	0.564	
										av.	0.541	0.565	0.542
14	15	10	con-	4.022	0.0882	0.632	0.775	0.707	0.744	0.489	0.637	0.622	
15	12	10	cen-	4.022	0.0882	0.632	0.775	2.055	2.177	0.837	0.639	0.622	
16	9	8	tric	4.022	0.0882	0.632	0.775	3.500	3.466	1.104	0.647	0.651	
17	9	7		4.022	0.0882	0.632	0.775	6.074	6.380	1.440	0.640	0.626	
										av.	0.641	0.631	0.640
18	12	10	eccen-	4.004	0.0875	0.627	0.779	0.790	0.760	0.484	0.604	0.615	
19	12	9	tric	4.004	0.0875	0.627	0.779	2.361	2.279	0.824	0.594	0.605	
20	9	7		4.004	0.0875	0.627	0 779	4.057	3.967	1.065	0.586	0.593	
21	9	7		4.004	0.0875	0.627	0.779	6.720	6.570	1.410	0.601	0.609	
										av.	0.596	0.606	0.598
22	12	10	seg-		0.0851	0.610	0.792	0.880	0.838	0.476	0.588	0.603	
23	12	10	men-		0.0851	0.610	0.792	2.646	2.474	0.823	0.587	0.606	
24	9	7	tal		0.0851	0.610	0.792	4.545	4.224	1.068	0.585	0.603	
25	9	8			0.0851	0.610	0.792	6.061	5.693	1.233	0.581	0.600	
										av.	0.585	0.603	0.585
26	15	21	con-	3.510	0.0672	0.482	0.877	0.752	0.752	0.332	0.622	0.622	
27	15	16	cen-	3.510	0.0672	0.482	0.877	2.248	2.258	0.566	0.614	0.612	
28	12	16	tric	3.510	0.0672	0.482	0.877	3.824	3.852	0.737	0.613	0.612	
29	15	19		3.510	0.0672	0.482	0.877	6.335	6.413	0.955	0.614	0.607	
										av.	0.616	0.613	0.611
30	15	18	con-	3.002	0.0492	0.352	0.936	1.217	1.210	0.288	0.618	0.619	
31	15	18	cen-	3.002	0.0492	0.352	0.936	2.166	2.166	0.379	0.610	0.610	
32	15	13	tric	3.002	0.0492	0.352	0.936	3.525	3.533	0.484	0.612	0.609	
33	18	13		3.002	0.0492	0.352	0.936	6.052	6.081	0.629	0.607	0.605	
										av.	0.612	0.611	0.609
34	24	21	con-	2.525	0.0348	0.249	0.969	0.619	0.620	0.138	0.608	0.607	
35	21	21	cen-	2.525	0.0348	0.249	0.969	1.818	1.819	0.239	0.615	0.615	
36	21	21	tric	2.525	0.0348	0.249	0.969	3.459	3.462	0.327	0.610	0.610	
37	21	21		2.525	0.0348	0.249	0.969	5.727	5.721	0.419	0.606	0.607	
										av.	0.610	0.610	0.611
38	27	25		2.033	0.0225	0.161	0.987	5.378	5.371	0.257	0.606	0.606	0.602
pipe at					A1								
diaphragm				5.059	0.1396								

TABLE 4 COEFFICIENTS OF DISCHARGE FOR ORIFICE ON 5-IN. PIPE

Run			Orifice			$\sqrt{1-(A_2/A_1)^2}$	H Static Head at Orifice	Cu. Ft. Sec. q	Coefficient $K = \frac{q\sqrt{1-(A_2/A_1)^2}}{A_2\sqrt{2gH}}$	
No	Length Min.	Readings	Diam., In.	A_2, Area, Sq. Ft.	Ratio, A_2/A_1				K	K (average)
1	2	3	4	5	6	7	8	9	10	11 (from curves)
10	12	5	4.000	0.0875	0.613	0.790	1.48	0.701	0.647	
1	15	8	4.000	0.0875	0.613	0.790	1.50	0.708	0.652	
24	9	7	4.000	0.0875	0.613	0.790	3.49	1.068	0.644	
20	9	7	4.000	0.0875	0.613	0.790	3.56	1.065	0.635	
16	9	8	4.000	0.0875	0.613	0.790	3.75	1.104	0.642	
11	12	5	4.000	0.0875	0.613	0.790	4.38	1.208	0.648	
2	9	4	4.000	0.0875	0.613	0.790	4.54	1.226	0.648	
25	9	8	4.000	0.0875	0.613	0.790	4.61	1.233	0.647	
21	9	7	4.000	0.0875	0.613	0.790	5.99	1.410	0.649	
17	9	7	4.000	0.0875	0.613	0.790	6.24	1.440	0.650	
12	15	8	4.000	0.0875	0.613	0.790	7.29	1.554	0.648	
3	12	6	4.000	0.0875	0.613	0.790	7.54	1.573	0.644	
								average	0.646	0.646
22	12	10	3.495	0.0666	0.467	0.884	1.51	0.476	0.640	
18	12	10	3.495	0.0666	0.467	0.884	1.55	0.484	0.642	
14	15	10	3.495	0.0666	0.467	0.884	1.58	0.489	0.643	
6	15	9	3.495	0.0666	0.467	0.884	3.46	0.725	0.644	
28	12	16	3.495	0.0666	0.467	0.884	3.69	0.737	0.634	
23	12	10	3.495	0.0666	0.467	0.884	4.53	0.823	0.639	
19	12	9	3.495	0.0666	0.467	0.884	4.60	0.824	0.635	
15	12	10	3.495	0.0666	0.467	0.884	4.75	0.837	0.635	
29	15	19	3.495	0.0666	0.467	0.884	6.04	0.955	0.642	
7	14	10	3.495	0.0666	0.467	0.884	8.63	1.136	0.638	
								average	0.639	0.639
30	15	18	2.998	0.0487	0.341	0.940	1.17	0.233	0.640	
26	15	21	2.998	0.0487	0.341	0.940	1.54	0.332	0.644	
27	15	16	2.998	0.0487	0.341	0.940	4.66	0.566	0.631	
33	18	17	2.998	0.0487	0.341	0.940	5.85	0.629	0.626	
								average	0.635	0.636
36	21	21	2.497	0.0340	0.239	0.971	3.42	0.337	0.628	
31	15	18	2.497	0.0340	0.239	0.971	4.62	0.379	0.628	
37	21	21	2.497	0.0340	0.239	0.971	5.63	0.419	0.627	
32	15	13	2.497	0.0340	0.239	0.971	7.57	0.484	0.626	
								average	0.627	0.634
34	24	21	1.999	0.0218	0.153	0.988	1.55	0.188	0.625	
35	21	21	1.999	0.0218	0.153	0.988	4.67	0.239	0.624	
38	27	25	1.999	0.0218	0.153	0.988	5.49	0.257	0.619	
								average	0.623	0.626
	pipe at orifice		5.113	A1 0.1426						

TABLE 5 COEFFICIENTS OF DISCHARGE FOR ORIFICE ON 3-IN. PIPE COMPUTED FROM THE CURVES DETERMINED BY JUDD AND KING, 1907

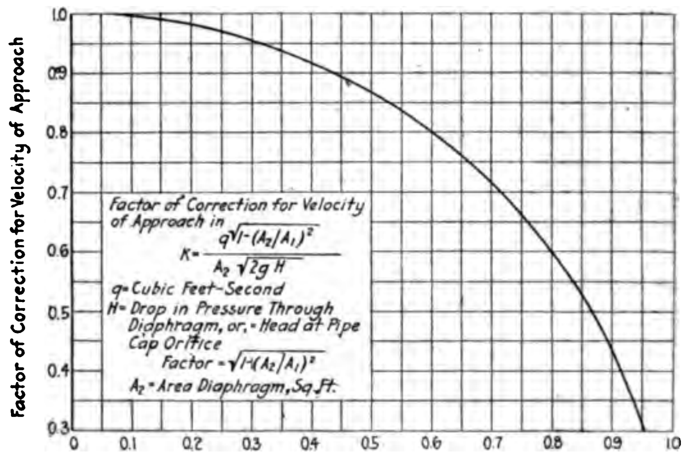
No.	Diam., In.	Area of Orifice, A_2	Ratio, A_2/A_1	$\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}$	H Static Head at Orifice	Cu. Ft. Sec. q	Coefficient $K=\frac{q\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}}{A_2\sqrt{2gH}}$	Average Coefficient
	1	2	3	4	5	6	7	8
1	0.50	0.00135	0.0266	0.999	4.0	0.0136	0.622	
2	0.50	0.00135	0.0266	0.999	9.0	0.0203	0.619	0.621
3	0.75	0.00301	0.0586	0.998	4.0	0.0299	0.618	
4	0.75	0.00301	0.0586	0.998	9.0	0.0449	0.620	0.619
5	1.00	0.00545	0.1060	0.995	4.0	0.0534	0.608	
6	1.00	0.00545	0.1060	0.995	9.0	0.0800	0.607	0.608
7	1.50	0.0123	0.2380	0.971	4.0	0.1276	0.628	
8	1.50	0.0123	0.2380	0.971	9.0	0.1915	0.628	0.628
9	2.00	0.0218	0.4238	0.905	4.0	0.2410	0.624	
10	2.00	0.0218	0.4238	0.905	9.0	0.3600	0.619	0.619
11	2.50	0.0341	0.6630	0.748	4.0	0.4580	0.628	
12	2.50	0.0341	0.6630	0.748	9.0	0.6840	0.626	0.627
pipe at orifice	3.072	area of pipe A_1 0.0514						

TABLE 6 COEFFICIENTS OF DISCHARGE FOR ORIFICES ON 4-IN. PIPE COMPUTED FROM THE CURVES DETERMINED BY H. JUDD, 1913

No.	Diam., In.	Area of Orifice, A_2	Ratio, A_2/A_1	$\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}$	H Static Head at Orifice	Cu. Ft. Sec. q	Coefficient $K=\frac{q\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}}{A_2\sqrt{2gH}}$	Average Coefficient
	1	2	3	4	5	6	7	8
1	0.998	0.00545	0.0595	0.998	4.0	0.0533	0.609	
2	0.998	0.00545	0.0595	0.998	9.0	0.0798	0.608	0.608
3	1.495	0.0122	0.1332	0.992	4.0	0.1175	0.594	
4	1.495	0.0122	0.1332	0.992	9.0	0.1765	0.595	0.595
5	1.75	0.0167	0.1820	0.984	4.0	0.1640	0.602	
6	1.75	0.0167	0.1820	0.984	9.0	0.2450	0.600	0.601
7	2.00	0.0218	0.2380	0.971	4.0	0.2175	0.603	
8	2.00	0.0218	0.2380	0.971	9.0	0.3240	0.599	0.602
9	2.495	0.0339	0.3700	0.929	4.0	0.3680	0.629	
10	2.495	0.0339	0.3700	0.929	9.0	0.5450	0.622	0.625
11	2.993	0.0480	0.5330	0.848	4.0	0.5700	0.628	
12	2.993	0.0480	0.5330	0.848	9.0	0.8480	0.622	0 625
13	3.497	0.0665	0.726	0.688	4.0	0.9910	0.639	
14	3.497	0.0665	0.726	0.688	9.0	1.4800	0.635	0.637
pipe at orifice	4.10	area of pipe A_1 0.0917						

low range of head with the 5-in. pipe orifices than for the 3-in. and 4-in. cap orifices.

44 The average diaphragm coefficients range from 0.602 to 0.706 while the average coefficients for the 5-in. cap orifices range from 0.610 to 0.637 for the 40 to 90 per cent orifices respectively. For the 80 per cent 5-in. pipe cap orifice the coefficients agree very closely (Table 4) with an average value of 0.646 as compared with a probable average value of 0.629 for the 3-in. and 4-in. cap orifices for 80 per cent orifices. For the 5-in. pipe orifices the coefficients for the orifices below the 80 per cent size average 0.608 for the diaphragms and 0.622 for all the cap orifices, or a difference of about 2.3 per cent, which is relatively a small variation and would seem to indicate



Ratio Orifice Area to Pipe Area $= \frac{A_2}{A_1}$

FIG. 20 FACTOR OF CORRECTION FOR VELOCITY OF APPROACH

that the behavior of water in passing through the diaphragm is very similar to that of a jet of water issuing from a thin-plate, or frictionless, orifice into the air.

45 The variation of the coefficients of discharge may also be seen from Fig. 18, where O is the average curve for the cap orifices and D is the average curve for the diaphragms. It will be seen that the average curve for the cap orifices coincides with the curve for the diaphragms at about 0.640, which is equivalent to 80 per cent pipe diameter.

COEFFICIENTS FOR SPECIAL ORIFICES

46 For the special orifices (80 and 90 per cent eccentric and segmental diaphragms) the coefficients, of course, run lower than

for the concentric diaphragms, since the pressure drop through the diaphragm is magnified by the offset at one side. For the 90 per cent size the variation is about 14 per cent for the eccentric and 23 per cent for the segmental diaphragm.

TABLE 7 VARIATION OF COEFFICIENTS OF DISCHARGE FOR PIPE ORIFICES, TAKEN FROM TESTS

Kind of Orifice	Orifice Diameter Expressed in Per Cent Pipe Diameter						
	40%	50%	60%	70%	80%	90%	Av.
3-in. cap	0.608	0.628	....	0.622	0.627	....	0.621
4-in. cap	0.595	0.602	0.625	0.625	0.631	0.637	0.619
5-in. cap	0.626	0.624	0.636	0.639	0.646	....	0.634
Average	0.610	0.618	0.631	0.629	0.635	0.637	0.627
Concentric Diaphragm	0.602	0.611	0.609	0.611	0.640	0.706	0.630

47 For the 80 per cent size, as might be expected, the variation is not so large, being 7 per cent for the eccentric and about 9 per cent for the segmental diaphragm (Table 8). Figuring the coefficient for the 80 per cent eccentric and segmental diaphragms on the basis of the average of the two pressure drops for both sides of the pipe at points ½ pipe diameter downstream, and comparing with the coefficient for the 80 per cent concentric diaphragm the results would be as shown in Table 8.

TABLE 8 COMPARISON OF COEFFICIENTS, 80 PER CENT DIAPHRAGMS, 4-FT PRESSURE DROP

Kind of Diaphragm	Coefficients	Comparison of Coefficients with that for Concentric Diaphragm
Concentric	0.641 From Table 3	
Eccentric	0.596 From Table 3	7 per cent less
Segmental	0.585 From Table 3	8.7 per cent less
Eccentric	0.629 From average head for both sides of pipe	1.8 per cent less
Segmental	0.616 From average head for both sides of pipe	3.9 per cent less

48 When figured on average head, the eccentric diaphragm gives a coefficient only 1.8 per cent less than the concentric diaphragm, and the segmental diaphragm gives on the average head a coefficient 3.9

per cent less than the concentric diaphragm. This indicates that the eccentric diaphragm, especially, has very little distorting effect on the stream flow. The curves also show that for the 80 per cent eccentric diaphragm, for a point $\frac{1}{2}$ pipe diameter downstream, the pressure drop is 18.8 per cent greater on one side than on the other, or a probable average magnification over the concentric diaphragm of 9.4 per cent; and for the segmental diaphragm there is a 24 per cent magnification, with a probable average of 12 per cent over the concentric diaphragm.

49 The coefficients of discharge for points A-1, B-1 ($\frac{1}{10}$ pipe diameter from diaphragm) for all sizes of diaphragms except the 90 per cent size, agree fairly well with those computed from points A-5, B-5 (see Table 3) running above in some cases and below in others. The coefficients run practically constant for each size of orifice for the full range of 6 ft. pressure drop.

COEFFICIENT OF DISCHARGE OF POINT A-10, B-5

50 The points at which the pressure drop was taken for computing the coefficients of discharge referred to in Table 3 were A-1, B-1 ($\frac{1}{10}$ pipe diameter) and A-5, B-5 ($\frac{1}{2}$ pipe diameter) from the diaphragm. In many cases A-10 would have been a more desirable point of attachment. For the concentric diaphragms a change from that point A-5 to A-10 on the upstream side would make no appreciable change in the coefficient of discharge. For the special diaphragms there is more variation in pressure adjacent to the diaphragm on the upstream side so that a change of from two to six per cent is shown in the coefficient of discharge in changing from A-5 to A-10 on the upstream side. Table 9 gives a comparison of coefficients of discharge for points A-5, B-5 and A-10, B-5.

51 Since from formula [3] given above the coefficient of discharge varies inversely as the square root of the head:

$$K_{10} = K_5\sqrt{\frac{H_5}{H_{10}}} = K_5\sqrt{\frac{H_5}{H_5 + d}} = \frac{K_5}{\sqrt{1 + \frac{d}{H_5}}}$$

K_{10} = coefficient of discharge for points A-10, B-5

K_5 = coefficient of discharge for points A-5, B-5

H_5 = pressure drop between A-5, B-5 taken as average from pressure drop curves

d = pressure drop between A-10 and A-5 as read from enlarged pressure drop curves, or as read from the observed data.

Example: Referring to curve No. 22 (Fig. 14).

$H_5 = 4.15$

$d = 0.13$

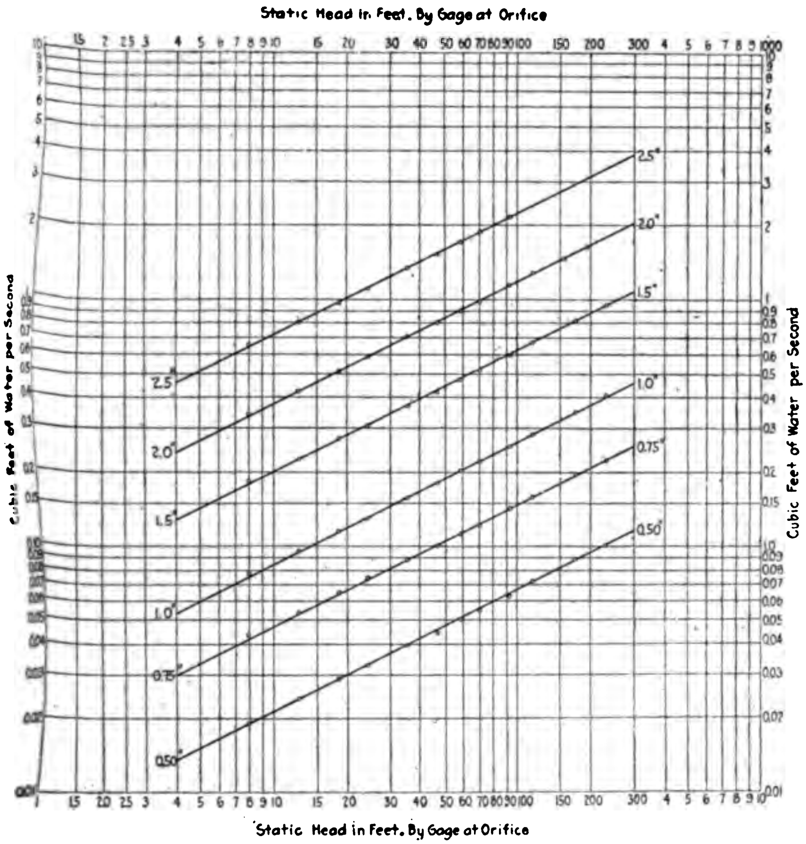


Fig. 21 DISCHARGE CURVES IN CU. FT. PER SEC. FOR FRICTIONLESS ORIFICES ATTACHED TO A 3-IN. WATER PIPE

$H_{10} = 4.02$ (also equal to ordinate A-10, B-10 on the curve)

$K_5 = 0.598$ (Table 3, 80 per cent eccentric diaphragm)

$$K_{10} = \frac{0.598}{\sqrt{1 - \frac{0.13}{4.15}}} = 0.607.$$

52 For the concentric diaphragms, one run for each size diaphragm was selected, having a head ranging from 3.0 to 5.0 ft., and

K_{10} was computed, noting the values of the pressure differences A-5, B-5, either from the observed data or from the curves plotte directly from the data taken. The values thus computed vary for K (Table 9) from the average values for K_5 by no greater than 0.5 pe

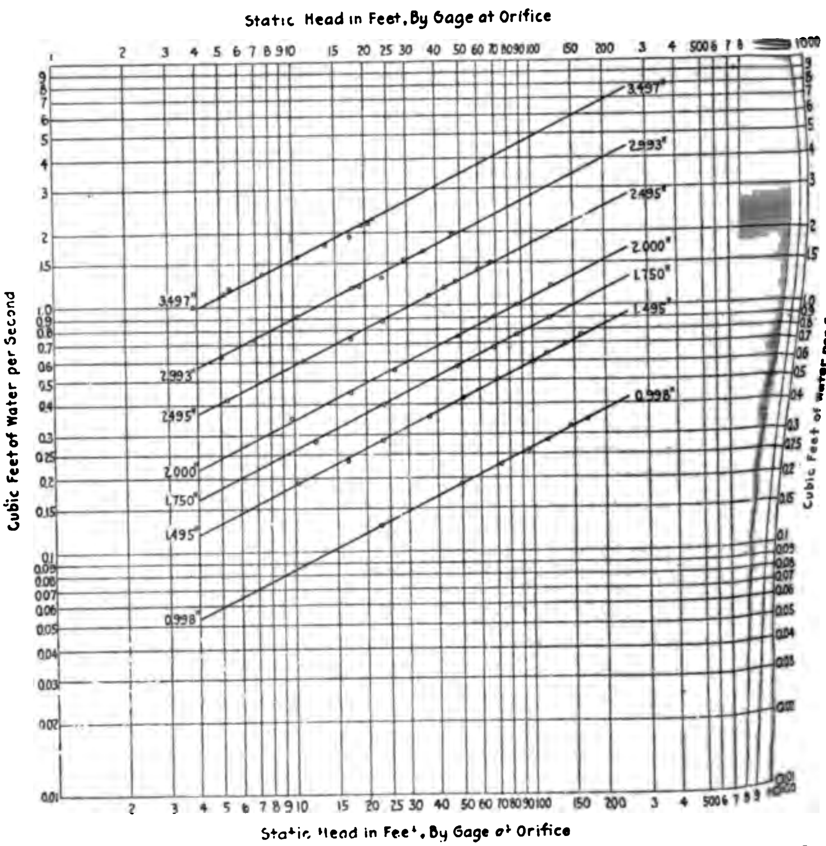


FIG 22 DISCHARGE CURVES IN CU. FT. PER SEC. FOR FRICTIONLESS ORIFICES ATTACHED TO A 4-IN. WATER PIPE

cent. For the 90 and 80 per cent eccentric and segmental diaphragms the coefficients referred to points A-10, B-5 run from 1.5 per cent to 6.1 per cent higher than the coefficients when referred to points A-5, B-5.

BEST POINT FOR PRESSURE CONNECTION

53 An inspection of Figs. 5, 10 and 11 shows that the point of greatest drop in pressure on the downstream side of the diaphragm

lies at a distance from the place of the diaphragm varying from 1 in. for the 90 per cent diaphragm to 2.5 in. for the diaphragms below the 70 per cent size. The curves for the smallest sizes and those for all the lowest heads on all the diaphragms below the 70 per cent size do not give this point of greatest pressure drop so clearly defined on the curves, although the observed data in nearly every case show a low point at 2.5 in. out from the plane of the diaphragm.

TABLE 9 COEFFICIENTS OF DISCHARGE FOR CONCENTRIC DIAPHRAGMS REFERRED TO A-10, B-5

Curve No.	Diameter, of Diaphragm, In.	Drop in Pressure, Ft.			Coefficient		Per Cent Difference
		From Curve A-5, B-5	From Data A-5, A-10	A-10, B-5	A-5, B-5	A-10, B-5	
1	2	3	4	5	6	7	8
			Concentric Diaphragms				
6	4.525	3.10	−0.01	3.09	0.706	0.706	0.0
18	4.022	3.55	−0.03	3.52	0.640	0.642	+0.3
30	3.510	3.82	−0.02	3.80	0.611	0.612	+0.2
34	3.002	3.59	−0.03	3.56	0.609	0.611	+0.3
38	2.525	3.53	0.0	3.53	0.611	0.611	0.0
40	2.033	5.44	+0.02	5.46	0.602	0.601	−0.1
41	1.530	5.35	+0.06	5.41	0.603	0.600	−0.5
			Eccentric Diaphragms				
10	4.503	5.15	−0.35	4.80	0.609	0.631	+3.5
22	4.004	4.15	0.13	4.02	0.598	0.607	+1.5
			Segmental Diaphragms				
	area						
14	15.72	4.13	−0.48	3.65	0.542	0.575	+6.1
26	12.26	4.53	+0.25	4.28	0.585	0.602	+3.0

54 These results are shown graphically in Fig. 19, in which the distance from the diaphragm is given in terms of the orifice diameter. The vertical scale gives the orifice diameters in per cent of the pipe diameter.

55 For nearly all of the diaphragms except the 90 per cent size a point of connection for pressure readings 2.5 in. from the plane on the downstream side would give pressure drop readings varying but little from the maximum pressure drop. For the diaphragms below the

70 per cent size the range of point of connection could easily be extended to one pipe diameter distance from the place without appreciable change for pressure drops for the concentric diaphragms. On the upstream side, for the concentric diaphragms, the point of pressure connection could be made at any point from $\frac{1}{2}$ pipe diameter to 1 pipe diameter from the plane of the diaphragm.

56 For the special orifices with one exception the point of greatest pressure drop lies surprisingly close to the $\frac{1}{2}$ pipe diameter distance from the plane of diaphragm on the right side of the pipe, which is the side of greatest offset. For the 80 per cent segmental diaphragm the point of greatest pressure drop occurs at 3.5 in. from the plane.

57 On the left side of the pipe, where the edge of the diaphragm is nearly in line with the inside of the pipe, there is found, of course, a less total pressure drop than on the other side. In all cases except for the 90 per cent eccentric the point of greatest pressure drop is moved downstream to about $\frac{8}{10}$ pipe diameter, or 60 per cent more than the corresponding point on the right side. For the 90 per cent eccentric diaphragm the point of greatest pressure drop seems to lie slightly within the $\frac{1}{2}$ pipe diameter distance from the plane.

58 On the upstream side, as might be expected with these special orifices, there is found to be a greatest pressure variation in the vicinity of the diaphragm. This difference seems to be more pronounced with the segmental orifices, due probably to the increase of swirling and eddying, as the water suddenly changes its direction sidewise on approaching these offset diaphragms, in addition to the sudden contraction it undergoes in passing through the orifice. In view of these disturbances the curves show that the upstream pressure connection should not approach the diaphragm nearer than 1 pipe diameter, to secure steadiness of pressure drop readings.

59 It is also plain to be seen that for the diaphragms above the 70 per cent size there is more need of care in making a pressure connection to keep close to the $\frac{1}{2}$ pipe diameter distance from the plane in order to avoid sudden pressure changes. It is also quite evident that for pressure drops through the diaphragm above 3 ft. the 90 per cent diaphragm would likely give unsteady readings, owing to rapid pressure changes, unless the downstream connection could be made as close as 1 in. from the plane of the diaphragm.

60 Some experiments along this same line have recently been made in England by Holbrook Gaskell, Jr.,[1] on water flow in 6-in. and

[1] Proc. Inst. Civil Engineers, vol. 197, p. 243, "The Diaphragm Method of Measuring the Velocity of Fluid-Flow in Pipes," by Holbrook Gaskell, Jr.

8-in. cast-iron water pipes. Mr. Gaskell states that the point of connection may be made at any point within 1½ in. from the plane of the diaphragm. It is not usually convenient to make a pressure connection even at 1½ in. distance from the diaphragm, hence it would seem advisable to use a diaphragm of not over 80 per cent size so as to allow the pressure connections to be made not nearer than ½ pipe diameter from the diaphragm. This would also insure more steady flow conditions.

ZONE OF MAXIMUM VELOCITY

61 Since the pressure drop must be largely due to increase of velocity, because of contraction of the jet of water on leaving the orifice, the point of least pressure would be at the zone of greatest contraction; that is, the zone where the water is moving most rapidly. The contour of the pressure drop curves would seem to indicate a jet of moving water whose shape must somewhat resemble the jet of water issuing into the air from an orifice. The curve in Fig. 19 is plotted to indicate the position of this so-called least action.

62 Its distance from the diaphragm, expressed in orifice diameter distances, is seen to increase as the diaphragm diameter decreases, although the actual distance does not vary much from ½ pipe diameter from the diaphragm plane. It is difficult to say whether or not the point of maximum pressure drop would be found at ½ pipe diameter distance for all diameters of pipes. The pressure drop curves show that the pressure has reached in nearly every case its maximum amount of restoration at about 4 pipe diameters from the diaphragm for all concentric diaphragms below the 70 per cent size. For these sizes the maximum pressure drop occurs at ½ pipe diameter from the diaphragm, which would seem to indicate that for area ratios of 50 per cent and under the pipe itself has more influence on the point of greatest pressure drop than the size of the diaphragm opening.

THE AMOUNT OF PRESSURE RESTORATION

63 The amount of pressure restoration after passing through the diaphragm will average as given in Table 10.

64 An example will serve to show how the per cent of pressure restoration was found. From curve No. 18, Fig. 5, 80 per cent concentric diaphragm, the maximum drop in pressure through the diaphragm = 3.7 ft.

The drop in pressure at B-16, point of greatest restoration = 1.36 ft.

Per cent of restoration $= \frac{3.7 - 1.36}{3.7} = 63.3$.

Average per cent for four heads = 62.0 (from Table 10).

TABLE 10 PER CENT PRESSURE RESTORATION

Size of diaphragm,	Per cent restoration
90 per cent concentric	77.0 initial pressure restored
80 per cent concentric	62.0 initial pressure restored
70 per cent concentric	48.0 initial pressure restored
60 per cent concentric	35.0 initial pressure restored
50 per cent concentric	30.0 initial pressure restored
40 per cent concentric	20.0 initial pressure restored
30 per cent concentric	11.0 initial pressure restored
20 per cent concentric	4.0 initial pressure restored
90 per cent eccentric	65.0 initial pressure restored
80 per cent eccentric	57.0 initial pressure restored
90 per cent segmental	65.0 initial pressure restored
80 per cent segmental	54.0 initial pressure restored

65 After the water has passed the contracted section of the jet its velocity is somewhat gradually reduced, which causes a restoration of pressure similar to the action in a venturi tube. For all the concentric diaphragms the point of maximum pressure restoration occurs at a distance of 4 pipe diameters, and for the special diaphragms the restoration of pressure is nearly if not quite completed at 5 pipe diameters distance from the plane of the diaphragm. The restoration of pressure appears to be nearly proportional to the area of the diaphragm.

CONCLUSIONS

66 *In conclusion*, the results of these experiments would seem to bring out the following points:

1 A thin plate orifice inserted in a pipe is as reliable for flow measurement as the thin-plate, or frictionless, orifice, as used ordinarily.

2 The shape of the pressure drop curves indicates a maximum point at about ½ pipe diameter from the diaphragm, which would seem to indicate a zone of maximum velocity of flow.

3 The eccentric and segmental diaphragms are advantageous in that they increase the drop reading, but more care is necessary in making the pressure connections, especially on the upstream side of the diaphragm.

4 The best point for the pressure connection is not less than ½ pipe diameter from the diaphragm on the downstream side, and not less than 1 pipe diameter from the diaphragm on the upstream side of the pipe.

5 The 80 per cent diameter size of diaphragm should not be exceeded to secure uniform flow conditions and to insure steady pressure drop readings.

6 The average coefficient of discharge for a 5-in. pipe diaphragm agrees very closely with the average coefficient of discharge for the 5-in. pipe cap orifice.

7 The coefficients for the pipe diaphragms vary more nearly in accord with the variation in size of diaphragm orifice than those for the pipe cap orifices.

8 The agreement is closer for the coefficients for all the pipe orifices at about 80 per cent diameter ratio.

9 The point of maximum velocity of flow is apparently a function of the pipe diameter and not of the diameter of the diaphragm orifice.

10 The restoration of pressure on the downstream side of the diaphragm is apparently a function of the area of the diaphragm.

ACKNOWLEDGMENT

The work was done in the hydraulic laboratory of The Ohio State University. Preliminary experiments on orifice diaphragms in the pipe formed the thesis of H. J. and W. W. Watson, of the class of 1915 in mechanical engineering, for whose able assistance within the past year on the work embodied in this paper, credit is hereby given.

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DISCUSSION

Sanford A. Moss (written). There is no doubt that orifices in thin plates or pipe diaphragms are accurate and valuable instruments in many cases. It must be borne in mind, however, that in some cases the venturi meter and orifice with gradual approach offer superior advantages. The most important one is the matter of calibration. A properly shaped convergent orifice or venturi meter with a smooth and gradual entrance and a proper parallel portion for sizing the jet has a velocity coefficient of about 0.99 for all cases whatever, whether for liquids, steam, air, or gas, and whether for relatively small differentials where the incompressible-fluid formula $V = \sqrt{(2\,gh)}$ can be used, or for relatively large differentials where the accurate thermodynamic formula has to be used. On the contrary, for thin-plate orifices, calibration has to be made for all of the different cases.

In the case of the measurement of flow of air and gas in pipes, there is often the problem of securing a differential of sufficient magnitude to give satisfactory readings. If a venturi meter is used the differential can be made properly large without causing any serious net pressure drop. The net pressure loss due to a good venturi meter with large differential and 3-to-1 ratio is in the neighborhood of $\frac{1}{5}$ of the differential. That is, 80 per cent of the differential is restored. This is to be compared with the figure of about 11 per cent given by Mr. Judd. For cases where the pipe pressure is high, so that a satisfactory differential can be secured with a total pressure drop which is a small percentage, the thin-plate orifice is satisfactory. With lower pressures, however, where pressure drop is more important, it is necessary to use a venturi meter because it gives very much less drop.

H. D. Fisher (written). A consideration of the results can lead one to no other conclusion than that the moving particles of fluid form the equivalent of a venturi tube, and that the form of this is as definite as though it were confined by a solid casing; for other experiments to which the writer has access show the same coefficients of discharge, or, if another form of expressing it is preferred, corresponding pressure drops for such different fluids as water, steam, air, and gas when the ratio of diameter of orifice to diameter of pipe and distances of connections are the same. This phenomenon is more obvious for the larger ratios of diameter of orifice to diameter of pipe, because for any given pressure drop the velocity of approach and energy of the fluid will be greater under these conditions. Also the surface of the moving stream and the amount of dead fluid compared to the volume of the jet will both be less, so that as the ratio of the diameter of the orifice to that of the pipe diminishes, the factors tending to break up and obscure this complete restoration of velocity head as static pressure increase very rapidly. It is found, however, with the velocities now common in practice that if it be desired to use one recorder for measurement with a number of orifices, the orifices show a comparatively narrow range of variation in ratio of diameter compared to diameter of pipe. This of course is reasonable, as fluid velocities are based on permissible pressure drop, which is a function of velocity and density, and of course varies with different classes of service.

The Author. In reply to Dr. Moss, I will say that I agree with him in that the venturi meter possesses a superior advantage in regard to the constancy of the coefficient and ability to get a high differential reading, so far as the concentric pipe diaphragms are concerned. The use of the segmental diaphragm, however, does not seem to distort the stream to a serious extent and affords also a means of securing a high differential reading.

In the author's opinion, the orifice diaphragm affords a distinct advantage in that it can be easily placed in an existing pipe line, in most cases, simply by breaking a flanged joint and inserting the diaphragm and gasket combined. While, on the other hand, the installation of a venturi meter requires the recutting and rearrangement of the pipe line.

Mr. Fisher has called the author's attention to the fact that, as the diaphragm was approached, there was a rise and not a drop in pressure as the author had first represented in the enlarged curves showing the pressure drop through the diaphragms (Fig. 14).

The rise in pressure is especially noticeable in the curves for 90 per cent and 80 per cent diaphragms, but can also be seen in the curves for the other diaphragms for the highest pressure drop through the diaphragms. For the eccentric and the segmental diaphragm the rise in pressure as shown by the curves for the right side is als seen to be accompanied by a corresponding drop in pressure on th left side of the pipe, which apparently indicates in the first place, a Mr. Fisher has pointed out, a restoration of static pressure due to th loss in the velocity of approach, and in the second place a loss static pressure on the opposite side due to increase in velocity as th stream swerves toward that side of the pipe on its way through th diaphragm.

No. 1539

DYNAMIC BALANCE

By N. W. Akimoff, Philadelphia, Pa.

Member of the Society

In this paper is described a machine devised by the author for correcting the condition of dynamic unbalance. The description is preceded by an explanation of the phenomenon of dynamic unbalance, made as elementary as possible by carefully excluding all references to products of inertia, momental ellipsoids, free and forced oscillations, etc. The subject of dynamic balance is not an involved one, and the author feels it is time it were placed on a purely rational basis.

2 For the general reader, a knowledge of the subject of dynamics of rotation as treated in Worthington's Dynamics of Rotation will suffice. To those who would like to develop a mathematical theory of the machine described, the author would recommend a study of Routh's Dynamics. Those interested in reciprocating balance are no doubt familiar with Dalby's well-known publication on the subject, and with Sharp's recent and extremely valuable Balancing of Engines; neither of these touches, however, in any way upon the subject of *dynamic balance* proper, in the sense in which it is generally understood and in which it will be considered in the present paper.

3 The balancing of reciprocating parts is purely a matter of calculations and of design; but the balancing of rotating parts, aside from the design, is a matter of trial and adjustment, because of its accidental nature.

4 In a theoretically perfect rotating body, symmetrical and made of homogeneous material, there cannot be any question of running balance — such a condition would be understood as a matter of course. But we know that in practice nearly all bodies rotating at high speeds show a certain amount of unbalance. The immediate consequences of this unbalance are vibrations in automobiles and turbines; defective commutation in electrical machinery; undue

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wear and strain on bearings; defective products in the cases of grinding disks for steel balls, woodworking machinery, etc.

5 As is well known to all, an unbalanced condition of a rotating body may be due to two distinct causes: lack of static balance, and lack of dynamic balance.

6 By static balance is understood the condition when the center of mass of the body lies somewhere on the axis of rotation. Such a condition is easily obtained on one of the static balancing machines of the knife-edge or of the roller type. In order to place the body into static balance, it is sufficient to drill one hole or to add one weight, although either of these might be split up into one or more components whenever desirable, all of which does not present any special difficulties.

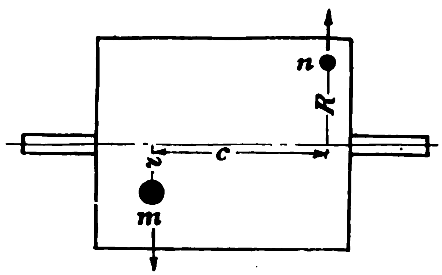

FIG. 1 CENTRIFUGAL COUPLE IN STATICALLY BALANCED BODY

7 Now, by dynamic balance is understood the condition when there is no so-called *centrifugal couple* in any axial plane. In a statically balanced body, Fig. 1, a centrifugal couple can only be due to *two* masses, m and n, on opposite sides of the shaft and located at a certain distance, c, axially, from each other. Such masses may be, for instance, the centers of gravity of corresponding congested regions. At any rate, in view of the static balance, (1) such masses must be in some axial plane, and (2) the products of each mass and its respective distance from the axis of rotation must be equal.

8 Such a couple is in general numerically equal to a certain coefficient multiplied by radius times weight times axial distance, that is, equal to $k.m.r.c$, where k involves the speed as well as other numerical constants. Now we shall choose the unit of speed in such a manner that k can be made equal to unity, so that the centrifugal couple is equal to $m.r.c$ or $n.R.c$ (since $mr = nR$).

9 Since the effect of a couple can only be counteracted by that of another couple, it will be seen that any effort to balance such a body

as in Fig. 1 by adding *one* weight, or drilling *one* hole, cannot fail to make matters worse. Such a method not only does not take care of the centrifugal couple, but also distorts the static balance.

10 It seems to be a "natural feeling" that the way to correct unbalance is to drill a hole at the "high spot," — the point where the marking tool touches the body. Indeed, it appears that various devices have been designed and are now on the market precisely for marking such "high spots," after which judicious removal of metal is supposed to secure the desired condition of balance. These devices are based upon the so-called floating bearing principle, Fig. 2, that is, are provided with bearings yielding in the horizontal plane to emphasize the running of the body out of true. Of course, all such devices can only serve to indicate that the body is out of balance; they can give neither the true axial plane of the disturbing centrifugal couple, nor the numerical value of it. Also, drilling in any one place,

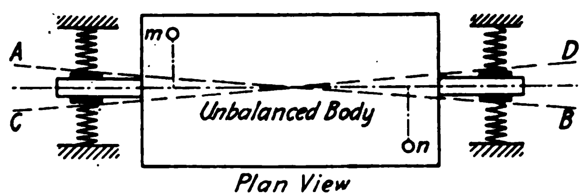

FIG. 2 FLOATING BEARING PRINCIPLE

as has just been seen, cannot secure dynamic balance, but can only distort the static balance.

11 It appears likewise that an attempt has been made to balance round disks, wheels, pulleys, etc., by pivoting them on one point and by marking the "high side." It is extremely difficult to ascertain just what is the underlying idea of such apparatus, as rotation of bodies on a fixed axis and rotation about a fixed point are two entirely separate chapters of dynamics, the latter much more difficult than the former, and the deductions of one apply in no way to the other. The same remark applies to the method consisting of rotating a body suspended on a flexible shaft (wire rope) and observing the high spots in this manner. All such attempts to ascertain the dynamic unbalance are perfectly irrational.

12 What it is absolutely necessary to know, in a statically balanced body, is:

a the exact location of the axial plane of the disturbing centrifugal couple *b* the exact numerical value of the disturbing couple

c the sign of the couple, i.e., the direction of the vector representing the couple.

13 Indeed, with the axial plane of the disturbing couple known, attention can be limited to that plane; and what is done on one side of the shaft will be repeated on the other side so as to preserve the static balance. Furthermore, with the numerical value of the couple *m.r.c* (Fig. 3) known, all that has to be done to secure dynamic balance is to introduce an opposing couple, *a.l.e*, of the same magnitude. Of course, this counteracting couple can be introduced in a variety of ways, small holes drilled on a large radius and far apart axially being equivalent to larger holes located on smaller radii and nearer to each other axially.

14 It will be seen from the above that a machine to deserve the name of dynamic balancing machine absolutely must indicate

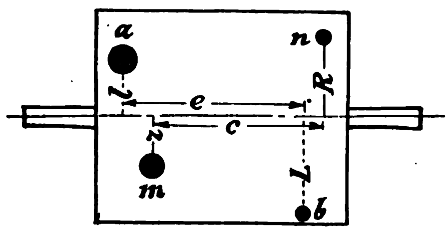

FIG. 3 COUPLE COUNTERACTING CENTRIFUGAL COUPLE

the plane of unbalance, as well as the numerical value and sign of the unbalancing couple. The author claims to have produced such a machine, of which the following is a brief description.

15 A rigid horizontal beam, such as a lathe bed, Fig. 4, is hinged at one end and supported by a spring at the other. The body to be tested, already in perfect static balance, is rotatably supported on the beam.

16 If dynamically unbalanced, the body will, when rotated, cause the beam to vibrate in a vertical plane, with a period of oscillation equal to the period of rotation of the body. In other words, if the speed of the unbalanced body is, say, 315 rev. per min., the beam will vibrate at the rate of 315 complete oscillations per minute, quite regardless of the characteristics of the spring (except possibly at the very beginning of motion).

17 Now imagine a second body, exactly similar in every respect to the first, also in perfect static balance but dynamically unbalanced

to precisely the same extent as the first body, temporarily associated with the same beam, say suspended under it.

18 If these two bodies are oppositely located as to balance and run precisely at the same speed (synchronously), then the unbalancing or disturbing couples will cancel out, and the beam will have no tendency to vibrate, no matter how badly unbalanced, individually, are the two bodies. This is the fundamental principle of the machine, —to determine unbalance by determining the unbalance necessary to neutralize its effect.

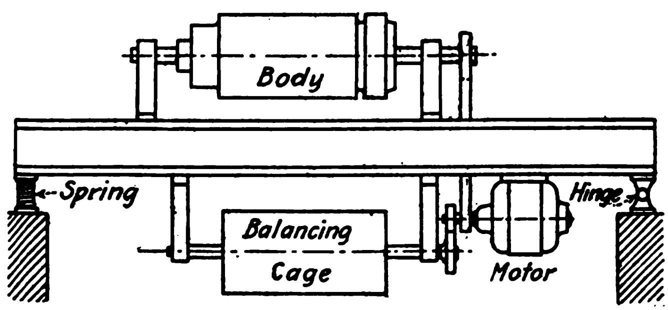

FIG. 4 PRINCIPLE OF DYNAMIC BALANCING MACHINE

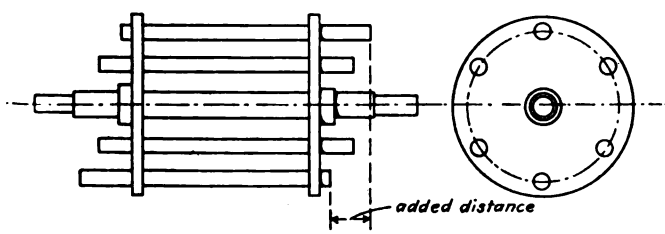

FIG. 5 CONSTRUCTION OF SQUIRREL CAGE

19 In the actual machine instead of the second body being an exact image of the original unbalanced body, it is a so-called squirrel cage, and this is rotated in unison with the article to be tested. The cage, Fig. 5, consists of two or more circular disks, carrying an even number of rods (usually six or eight) arranged slidably in the disks. The rods are accurately made and their common weight is known; therefore, any displacement of one of the rods with respect to the one exactly opposite will not affect the static balance, originally perfect, of the cage, but will introduce a certain centrifugal couple, according to the relative displacement or *added distance*.

20 For instance, suppose that the unbalanced body is a special fan, Fig. 6, and that the unbalance is due to two excess weights, grossly exaggerated in the figure. This will result in a centrifugal couple, and to counteract it the cage will have to be put into a state of unbalance, as shown by the relative displacement of the rods and as measured by the added distance.

21 Thus the cage has means for indicating the exact amount of unbalance which has been put into it in order to reproduce with the opposite sign the exact unbalance of the article being tested. For instance, the displacement or added distance of $\frac{15}{16}$ in. may represent

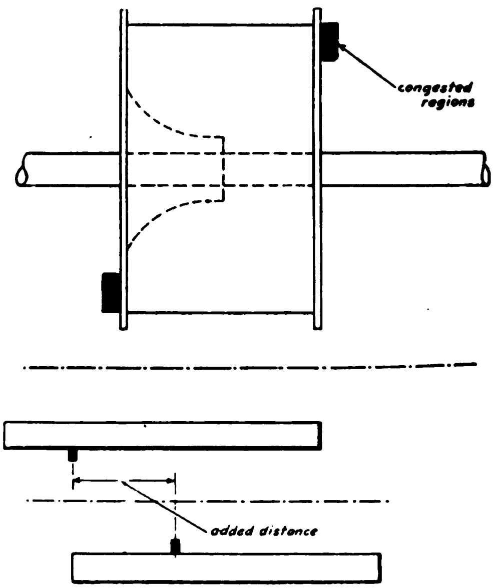

FIG. 6 DYNAMIC UNBALANCE IN SPECIAL FAN

(for a certain speed) a couple of, say, 120 oz-in. The plane of unbalance is easily established by the location of the two rods, the moving of which into a new position stops the vibrations; and the value of the couple is immediately given by the added distance.

22 It should be clearly understood that the centrifugal couple due to the body acts upon the beam in a simple harmonic manner, that is, according to the law of sines; but so does the effect of the balancing cage. In other words, when the axial plane of unbalance is vertical, the effect of unbalance on the hinged beam is the greatest, as also is the effect of the correcting element, the cage.

3 When the plane of unbalance is horizontal, and to this is ortional the free period of the small oscillation of the beam, that e correcting element is likewise horizontal, since the cage and the rotate in unison, and neither is in any manner felt by the beam h does not respond visibly to any but vertical efforts or the cal components of other couples.

So far as the spring (Fig. 4) is concerned, its object is to ify the amplitude of the vibrations, although an unbalanced will always cause the whole bed to vibrate with a frequency

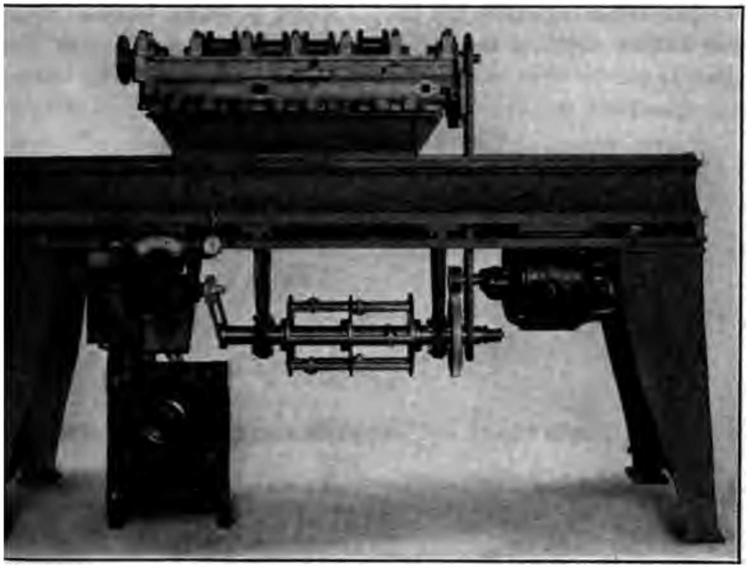

FIG. 7 DYNAMIC BALANCING MACHINE. BALANCING AUTOMOBILE ENGINE AXLE

sponding to the speed of the body (on this principle is based the known vibrating tachometer).

5 However, there is an additional advantage in the use of spring, as it is always possible to select the characteristics of this at, under its load, its free period of oscillation will correspond tly to the rate at which it is desired to run the test. Such a hronism has a large magnifying effect, so that even a slight lance results in a considerable amplitude of oscillation. The ral period of oscillation of the spring is calculated from

$$T = 2\pi\sqrt{MU_d}$$

where

T = period in sec. of one complete (double) oscillation
M = mass by which spring is actually loaded
U_d = unit deflection, ft. per lb. load
π = 3.1416.

26 The details of construction of the cage would not be of any material interest in the present discussion, but the main feature is that the rods can be adjusted axially while the cage is in rotation, the speed of the cage never being higher than 400 rev. per min. or so.

27 With regard to the proper speed at which to balance a body, the author submits the following considerations. Unless the body itself is elastic or is mounted on a flimsy shaft, the body balanced at any speed will run true at any other speed. If the shaft is not strong

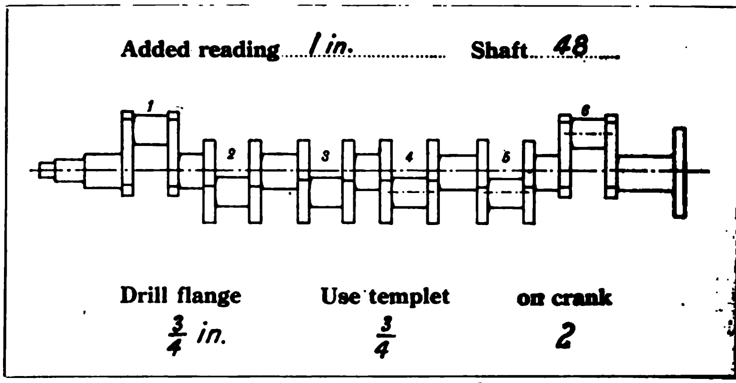

Fig. 8 Directions for Balancing Shaft

enough, no balancing machine can make it stronger. If the windings of a rotor seek to find their places under a certain speed and temperature, they should be allowed to do so, after which balancing can be done. In the "high spot" method, in which azimuth (commonly known as lag or lead) depends upon the speed itself, it is of course of considerable importance to watch the speed; but in the rational method here proposed the best speed is determined only by the characteristics of the spring, Fig. 4.

28 Fig. 7 illustrates a dynamic balancing machine for testing automobile crankshafts previously placed in perfect static balance. The information furnished by the machine is simply the relative displacement (added distance) of one of the pairs of rods of the squirrel cage. By referring to a set of specially prepared tables, similar to the one shown in Fig. 8, the operator can pick out readily

the necessary directions as to how to remove a certain amount of metal from one of the cranks, and how to drill the flange in order to secure perfect dynamic balance.

29 Fig. 9 shows a machine arranged for testing armatures weighing up to 2000 lb. previously placed in perfect static balance. The directions to be followed in balancing are given in the very easy form shown in Fig. 10. There can be no ambiguity in carrying out the instructions.

CONCLUSION

30 It is not necessary to dwell on the extreme importance of testing a body for unbalance in its natural position and upon its own

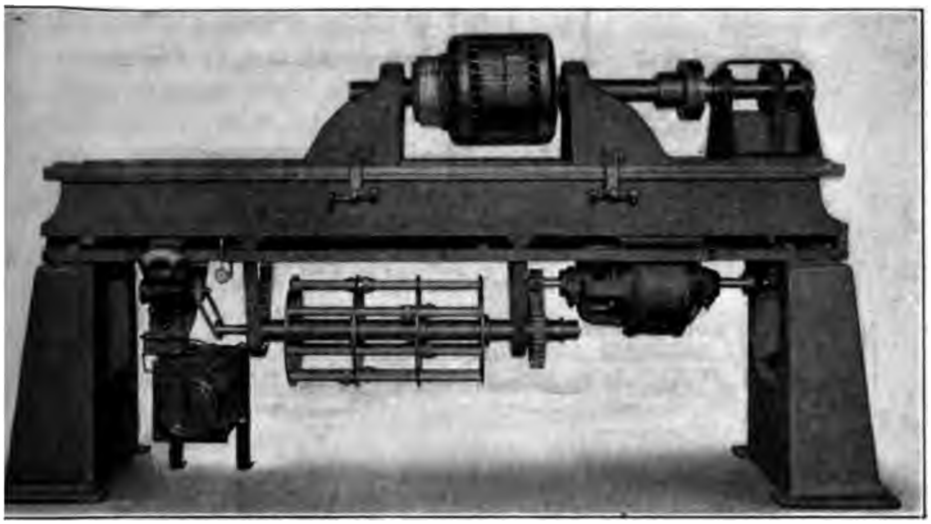

FIG. 9 DYNAMIC BALANCING MACHINE. TESTING AN ARMATURE

bearings. This machine, in its very conception, is adapted for testing under such conditions. The shaft in Fig. 7 is being tested in its own crank case.

31 The importance of static balance cannot be overestimated. Fortunately such a condition can be readily secured.

32 Theoretically, one pair of rods in the squirrel cage, Fig. 5, of the machine would suffice, since the relative position of the rods to the body being tested can be altered readily through the transmission (chain drive shown in Fig. 4). For convenience, however, it is best to have three or four pairs of rods, and even then it is some-

times necessary to change the angular position of the cage so that the balancing can be done by one pair of rods, and not two, as often happens at the beginning of a test.

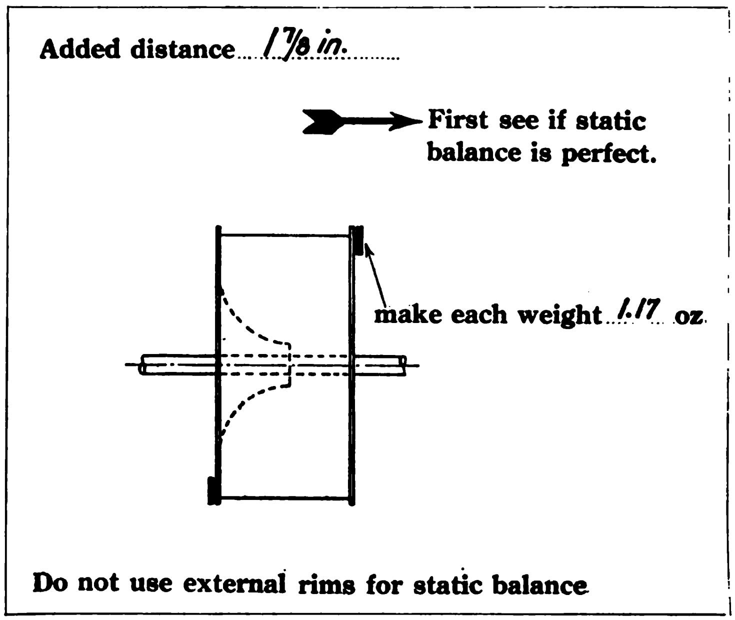

FIG. 10 DIRECTIONS FOR BALANCING BLOWER

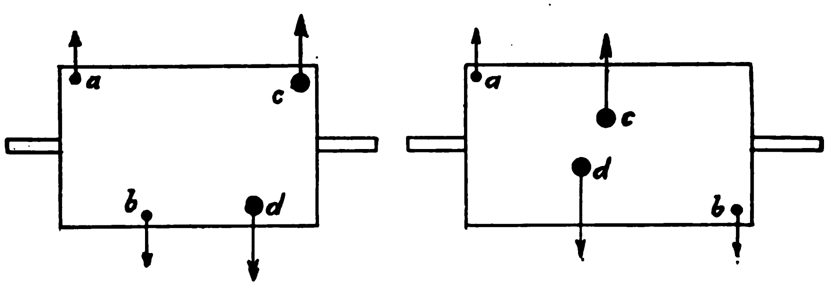

FIG. 11 SPECIAL CASES OF CORRECTING COUPLE

33 Fig. 11 illustrates a somewhat special case. Here the correcting couple *c.d*, numerically equal to the disturbing couple *a.b*, has been introduced for some local reason in the manner shown.

That is, the action of the correcting couple seems to be in a region rather remote from that of the disturbing couple. This simply means that the two actions will neutralize through (and within) the body. Even in such an extreme case, however, the supports or foundations of such an apparatus will be found to experience no vibratory effect.

DISCUSSION [1]

E. J. Loring (written). I would define a body as dynamically balanced when its axis of unconstrained rotation is concentric with its journals.

The method proposed assumes perfect static balance, but dynamic-balance conditions are far more sensitive because a slight static unbalance is greatly magnified at high speed. I do not agree that static balance is easily obtained. On the contrary unbalance always persists to some degree and must be provided for in the dynamic-balancing operation. A body dynamically unbalanced does not usually show high spots diametrically opposite at the two ends, as it would if a simple couple were acting.

When provision is made for attaching balancing weights in two well-separated planes between the bearings of a two-bearing shaft, it appears possible to determine the position and amount of added weight required in the plane nearer to each bearing in not more than three "shots," and with this information for each balancing plane the proper weight and position for each plane to clear both bearings simultaneously may be found by a simple calculation.

A slow-speed test is by no means conclusive; a body in dynamic balance at 400 r.p.m. may be dangerously far from balance at 3000 r.p.m. When corrected for 400 r.p.m. it is safer than when only statically balanced, but its balance is never fully proved at less than its full speed.

I would be interested to know how the correct plane of displaced rods is picked out from the behavior of the beam and the rotating body. Is it by a definite rule or by selection and trial?

F. Hodgkinson (written). It seems to me that the author's device is open to the objection that the member to be dynamically balanced must be put in perfect static balance before being balanced by the means described. In the case of heavy revolving bodies this is quite difficult, and is not to be satisfactorily attained by the usual method of rolling the body on ways or the like. Therefore, it seems to

[1] Complete discussion is published in The Journal for August 1916.

me that a machine for dynamic balancing should also render possible the elimination of errors in static balance.

It would seem evident that the spring support at one end of his frame should have a natural period of oscillation equal to the period of revolution of the body being balanced. This of course would mean a different spring for every different weight of body applied, which would require some calculation in advance of doing the work, where there is a great variety of things to be balanced.

The author speaks disparagingly of the older method of securing dynamic balance, that is, running the body at a reasonably high speed and marking the shaft and determining the "high spots." He says they can only serve to indicate the body is out of balance. As a matter of fact, a very well-organized system of adding weights may be employed which renders both the static and dynamic balancing of the body a simple operation. One of the hardships of the matter, however, is that the adding of weights or drilling, as the case may be, must be done more or less piecemeal, and the time occupied by speeding up and shutting down occasions a rather severe loss.

In stating that the static balance is not easily obtained, I referred particularly to bodies such as revolving fields of great weight where to mount these on ways, giving them a rolling balance, is unsatisfactory because, in spite of the surfaces being hardened and made as level as possible, the journals will sink into the ways to a slight degree, so extreme sensitiveness cannot be obtained because of the field having to run up hill.

It has been frequently found, in the case of very large fields, that an approximate static balance may be obtained by smearing the bearing surface and journals well with a heavy cylinder oil, dropping the field into the bearings and quickly removing the chains. The friction would be so low that the field would respond to an error in static balance better than the previously described method of rolling on ways. Of course the field must not be allowed to stand more than a fraction of a minute, or until the oil film has been squeezed out, for then of course the friction becomes very material.

So far as steam turbines are concerned, it has not been the practice of the company with which I am associated to resort to dynamic balancing except in a few extreme cases. Generally the speed of the drum is low, so that ordinarily static balancing is sufficient. Further, the disks or rings are short in their axial length as compared with their diameters, and a careful static balancing of these is found to be all-sufficient. For balancing these disks and rings, a special static balancing machine was devised many years ago and has been in continual use. (See U. S. Patent No. 710,148, September 30, 1902.) By the use of this machine a disk weighing 4000 or 5000 lb. may be given a static balance within an error of half an ounce.

F. Hymans (written). In addition to the detrimental effects of an unbalanced rotating body enumerated in this paper, loss of power in causing and maintaining vibrations of the structure supporting the body should be mentioned. This loss has been demonstrated by Sommerfeld (Z. d. V. d. Ing., 1902) in experiments with a small electric motor which carried on its shaft an eccentric weight. Placing the motor on a table, it was found that at 310 r.p.m. the table began to execute horizontal vibrations. In an endeavor to increase the motor speed the voltage impressed on the armature was raised. It was found, however, that the voltage could be considerably increased without causing any increase in the speed of the motor and the increased power consumption was absorbed in maintaining the vibrations of the table.

By perfect balance of a body rotating freely around a horizontal axis is understood the condition when there is no bearing pressure save that due to the body's weight. If it were possible therefore to measure directly the bearing pressure and its direction relative to the body, we should have the simplest form of a balancing machine and be in a position to determine the corrections necessary to bring about perfect balance. Unfortunately, however, direct measurement of forces is rarely possible.

It is not impossible to determine the corrective means for static as well as dynamic balance simultaneously on one and the same machine. However, the problem is considerably simplified if a static balance is first secured, as in a body so balanced the resultant of the centrifugal forces to which its particles are subject is a couple only, instead of a couple and a force, as in a body entirely unbalanced.

Considering more in particular the balancing machines of the vibrating type, we can broadly divide them in two classes, namely: *First*, machines in which the vibrations excited by the centrifugal couple are directly employed to determine its plane of action, as is the case with nearly all balancing machines existing heretofore. (See also description of the Lavaczeck machine in The Journal, March, 1916.) *Second*, machines in which the vibrations are merely employed to indicate lack of balance, of which class Mr. Akimoff's machine is to date the sole example.

Referring to the first-mentioned class, the position of the plane of unbalance is always determined by observing the position of the body at the instant of maximum amplitude of the oscillations. When the body is rotated at a sufficiently low speed, so that the period of rotation is very large as compared with the period of free vibration of the system, the oscillations of the machine will be in phase with the exciting forces, and it is a simple matter to determine accurately the plane and even the magnitude of the centrifugal couple. A machine so arranged would, however, not be sensitive enough for the purpose.

On the other hand, when the body is rotated at a speed at which the period of free vibration of the system is no longer negligible as compared with the period of rotation, there is a phase difference, due to damping forces, between exciting forces and oscillations. As the amount of this phase difference is uncertain, the plane of unbalance can only be determined approximately. In addition the oscillations can no longer be employed to calculate the magnitude of the centrifugal couple.

In the Akimoff machine it is quite immaterial if there is phase difference between oscillations and exciting forces. As described, an additional couple is introduced, which can be so adjusted in phase and magnitude that the vibrations of the balancing machine vanish. The machine affords therefore at once the means to determine exactly both the plane and amount of unbalance. Disturbing influences such as "weakness" of the shaft of the rotating body can be practically wholly eliminated by selecting a spring of a flexibility which is large as compared with that of the shaft. In other words, by the proper selection of the spring the speed of rotation of the balancing machine can be made sufficiently low that the transverse vibrations which the centrifugal couple tends to set up in the shaft of the rotating body will be practically of no effect on the beam of the balancing machine.

T. A. Bryson (written). The author explains that it is extremely difficult to ascertain the underlying idea of apparatus in which the axis of the body is pivoted at one point, and that the dynamics of the case is much more difficult of solution than that of a body having a fixed axis. Is not the action of the device shown similar to the former case, since all axial motions of the body are transmitted to the hinged frame?

It would seem that much might be gained by providing for close adjustment of the axial plane of the balancing weights while the cage is in rotation, thus performing both operations without stopping to change the angular position of the cage.

Is the chain and gear drive free from vibrations which would affect the results?

The Author. In explanation of the several points brought up by those discussing my paper, I will review two of these discussions.

Mr. Loring, unfortunately, gives no explanation or reasons to justify the several assertions which comprise his discussion. I will consider each paragraph separately.

1 The definition of dynamic balance is objectionable because it involves a conception of higher order; if one knows what an unconstrained axis is, no definition of what constitutes dynamic balance is needed. Furthermore, an axis, being a straight line, cannot very well be *concentric* with anything. This, of course, may be only a *lapsus linguæ;* but meant as a correction it is out of place.

2 Regardless of what the high spots show (in general they mean nothing at all), it was explained in the paper that the unbalance can only be due to a couple, provided that the body is in static balance; remove the couple and the body is in balance. This is being done every day in our shops, and by the users of these machines.

3 So far as the possibility of placing a body in balance in two or three "shots" is concerned, I will merely ask this question: Who has ever said, or on what machine was it ever proved, that the body was really placed in balance? Is it because the operator himself said so? Is it because a machine, wrongly constructed upon a radically wrong idea, showed so? Is it not clear that it is absolutely impossible to secure any such result in three shots, except if the machine is so constructed that it shows balance where none exists?

4 Here are two examples showing that slow speed may be very conclusive indeed: 21-in. blowers are being balanced on our machine at 450 r.p.m., and operate at 800 to 1300 r.p.m. very satisfactorily; by older methods no such satisfaction could have been secured at all. Another rotor was balanced by us at 350 r.p.m. and ran very well at all speeds, including its highest speed of 2500 r.p.m. There is nothing unusual about this. Balance is balance; if the rotor is so constructed that it changes it shape with its speed, it should be thrown away, because it cannot be balanced at all. But of course supposed illusory balance may show one result for a low speed and another, quite different, for a higher speed.

To Mr. Hodgkinson "it seems evident that the spring support on one end should have a natural period or oscillation equal to the period of revolution of the body being balanced. This, of course, would mean a different spring for each different weight of body applied...." As a matter of fact, however, this is not the case. By writing down the easy differential equation controlling this small oscillation, it will be seen that the free period depends not only on the weight but upon three other things, so that it is perfectly possible to imagine a great variety of different weights which would not necessitate any change whatsoever in the spring support. Full advantage is being taken of this fact in the arrangement of the machine, which is such that the operator has no calculations to make. This is a vital point that has been carefully considered in the design of my machine.

In conclusion, the trouble with an understanding of what constitutes dynamic balance is that it is based on the presence (or absence) of certain quantities called "products of inertia" with which the practical man is in general quite unfamiliar. I did not wish to introduce the products of inertia in my paper and for this reason had to explain things by something easier, the centrifugal couple, although this meant a much longer explanation. But I have seen some big practical men think the matter over and thoroughly digest the meaning of the expression *centrifugal couple*, after which, according to their own testimony, the "thing came as clear as daylight" to them, which, however, is only too natural, because it is based upon common sense pure and simple; only it takes a little dynamics to put matter into an easy shape in which common sense can grasp the real meaning of this would-be elusive conception *dynamic balance*.

The principle of my machine has been worked out in a purely deductive, analytical way, and not by building up or by a cut-and-try process. To begin with, I had this problem in mind: how to tilt the momental ellipsoid so that its axis, the nearest to the axis of rotation, will be brought into absolute coincidence with the latter. This can easily be done by building the cage around the body, which is correct, although in the immense majority of cases impractical. From this it was but one step to splitting up the body and the cage.

Inasmuch as we deal with couples, that is, moments, there was nothing else to do except to write down the equation of the oscillation of the beam and work out the parts in detail. This did not require a single assumption or postulate, but was derived directly from elements of rigid dynamics.

No. 1540

THE MEASUREMENT OF VISCOSITY AND A NEW FORM OF VISCOSIMETER

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In determining the lubricating properties of an oil, the viscosity test is considered of great value, since by means of this test a good oil can readily be distinguished from a poor one. It is therefore very important that the engineer be able to measure the viscosity of an oil and also the variation of viscosity with temperature.

2 The present paper, dealing with the measurement of viscosity, gives in part the results of a somewhat extended research on the lubricating properties of oils. It is to be followed by a paper giving the relation between the lubricating properties of oils and various easily measurable physical properties.

VISCOSITY

3 Matter in all states exhibits a gradual yielding to tangential forces which tend to change its form. This property is termed viscosity and may be defined quantitatively as the tangential force per unit area divided by the shear per unit time.

4 To gain a clear physical concept of this definition, consider a plane surface, Fig. 1, of area S, parallel to and at a distance d from another large plane surface, and the intervening space filled with a liquid whose coefficient of viscosity is η. If a given force, F, acting on this plane, is applied to S, the surface will be dragged along and

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will finally attain a steady velocity, which denote by V. In accordance with the above definition, the relation between these various factors would then be

$$\eta = F/S \div V/d \quad \text{or} \quad \eta = F \cdot d \div S \cdot V \ldots\ldots\ldots\ldots\ldots [1]$$

the value of η depending on the units in which these various factors are measured. If absolute values are desired, the factors on the right hand side of the equation are to be measured in C.G.S. units; but if relative values will suffice, as is the case in nearly all engineering work, they may be measured in any units.

5 It must be borne in mind that equation [1] is not true when the velocity, V, is changing, for then only a part of the force, F, is used to overcome viscosity, a part being used in giving accelerated

Fig. 1 Conception of Viscosity

motion. Under such conditions, another term must be added to this equation. The nature of this term is not difficult to see, for let equation [1] be written $F = \eta \cdot S \cdot V/d$. When there is accelerated motion, the force, F, will be divided into two parts, the part used to overcome viscosity and that used to give accelerated motion. Call these parts f_1 and f_2 respectively, then at any instant $F = f_1 + f_2$, where $f_1 = \eta \cdot S \cdot V/d$ and $f_2 = M \cdot dV/dt$, where M is the mass that is being accelerated and dV/dt is the average change of velocity per second of this mass. The complete instantaneous equation thus becomes

$$F = \eta \cdot S \cdot V/d + M \cdot dV/dt \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

and the viscosity cannot be found from this equation unless the last term can be evaluated, which is usually difficult and often impossible.

DEVELOPMENT OF A WORKING FORMULA

6 In most physical measurements, comparative values are more easily obtained than absolute values. As a result measurements are made in terms of some standard, the absolute value of which has been determined by a more or less laborious process. In making measurements of viscosity, the standard usually chosen is water at a temperature of 20 deg. cent.

7 Girard and Poiseuille, by studying the flow of liquids through capillary tubes, were the first to measure the absolute value of viscosity with anything like accuracy. On the basis of his excellent experimental work on the viscosity of water, Poiseuille deduced the formula

$$V = K \cdot D^4 \cdot H/L$$

where V = volume of liquid transpired
L = length of capillary
D = diameter of capillary
H = pressure
K = constant for each liquid at a given temperature.

Later, this empirical formula and its corrections were proved by several investigators.[1]

8 By assuming there is no slip at the surface of the capillary, that the liquid flows steadily without eddies or turbulent motion, and that there is no kinetic energy of efflux, then the transpiration formula for a liquid flowing under its own head becomes

$$\eta = \frac{\pi}{8}(hgr^4/lv)\,\rho t$$

where η = coefficient of viscosity (often contracted to viscosity)
h = liquid head
g = acceleration of gravity
r = radius of capillary
l = length of capillary
v = volume of flow
t = time of flow
ρ = density of liquid.

Experiment shows that the first assumption is true if the liquid is one that wets the surface of the capillary. The work of Reynolds shows the second assumption is true if the velocity of the liquid through the capillary is kept less than $700 \cdot \eta/\rho \cdot r$ cm. per sec. The third assumption, of course, can never be true. The liquid

[1] Stokes, Trans. Camb. Phil. Soc., 1849, vol. 8, p. 287; Wiedmann, Poggendorf's Annalen, vol. 99, p. 177; Hagenback, Pogg. Ann., vol. 109, p. 385; Stefan, Wien. Bar., vol. 46, p. 495; Couette, Ann. Chim. Phys., vol. 21, p. 433; Neumann, Vorträge über Hydrodynamik; Wilberforce, Phil. Mag., vol. 31, p. 407; Jacobson, Arch. f. Anat. u. Physiol. 1860, p. 80; Knibbs, J. Roy. Soc. N.S.W., vol. 29, p. 77; Boussinesq, Compt. Rend., vol. 110, p. 1160; Brillouin, La Viscosité (Gauthiér Villars, 1907).

must have kinetic energy when it leaves the capillary. In accordance with equation [2], a correction term must be added which, according to Couette, Finkener, and Wilberforce, should be $-v\rho/8\pi lt$. The complete expression for the viscosity is therefore of the form

$$\eta = \rho\ (A \cdot t - B/t)$$

where A and B are constants for any piece of apparatus, and ρ and t are the density and time of flow, respectively.

9 After a thorough examination of the recorded data on the viscosity of water, Knibbs concluded that the correct formula should be

$$\eta = \pi hgr^4\rho t/8\,lv - 1.12\,v\rho/8\,\pi lt$$

It is to be noted that the correction term is larger than that given by Couette. Moreover, this correction term varies with the time of flow, approaching zero when the velocity of flow is very slow, in which case t becomes very great. The value of the term increases with the temperature of the liquid since the time of flow decreases, and so the percentage of error in a viscosity vs. temperature curve due to neglecting this correction factor will increase abnormally toward the higher temperatures. This is due to two causes: the correction to be applied increases with the temperature and the value of the quantity to be corrected decreases rapidly with the temperature. Attention will be called to this fact when some of the various comparative methods are discussed. It is further to be noted that this correction term varies inversely as the length of the capillary. A short capillary requires a large correction term.

10 This formula, as corrected by Knibbs, has been submitted to careful experimental test by Hosking,[1] and by Bingham and White.[2] These investigators have determined the constants of the formula experimentally and have obtained fair agreement with theory. The capillary tube method, as employed by these experimenters, though complicated and laborious, can be depended upon for giving absolute values of the viscosity, and the accuracy of any viscosimeter can be determined by a comparison with the results obtained by this method.

CLASSIFICATION OF VISCOSIMETERS

11 *Class* 1. *Short Capillary.* In meters of this type, the liquid to be tested is forced either by gravity or by pressure through the

[1] Phil. Mag., 1900 [5], 49, 274; 1904 [6], 7, 469.

[2] Zeits. Physikal Chem., 1912, 80, 684.

capillary and the viscosity is determined in terms of the time required for a given volume to pass through the meter, as compared with the time required for a standard liquid to discharge the same volume.

12 A cross-section of a meter of this type is shown in Fig. 2. The essential parts of the instrument are a cylindrical bowl, A, in which the oil is placed and a short capillary tube, B, through which it is discharged. The instrument must have temperature controlling and measuring devices, means for starting and stopping the flow and volume and time measuring apparatus.

13 To this class belong the Saybolt meter, adopted as a standard by the Standard Oil Co.; the Engler meter, adopted as a standard

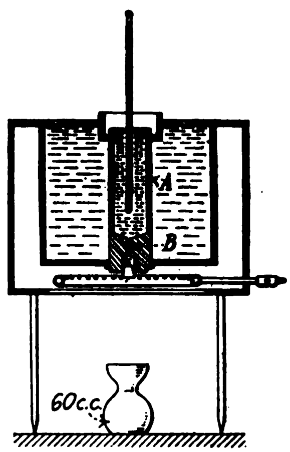

FIG. 2 SHORT CAPILLARY TYPE VISCOSIMETER

by the U. S. Government and Germany; the Redwood meter, adopted as a standard in England; the Scott meter; and the pipette, adopted as a standard by the Pennsylvania Railroad and much used by chemists. The majority of the viscosimeters on the market are of the short capillary type.

14 *Class* 2. *Orifice.* This type employs an orifice in place of the short capillary of the previous type. Fig. 3 is a section of such a meter, in the cylindrical bowl, A, of which the oil to be tested is kept at constant head above the orifice, B. The Carpenter meter belongs to this class.

15 *Class* 3. *Dropping a solid body through a tube filled with the liquid,* the solid body being usually a sphere or a plunger. Meters

employing this principle determine the viscosity in terms of the time required for the body to drop a certain distance through the liquid, as compared with the time required for the same body to drop through a standard liquid. To this class of meters belong the Perkins meter, which employs a plunger and vertical tube, and the

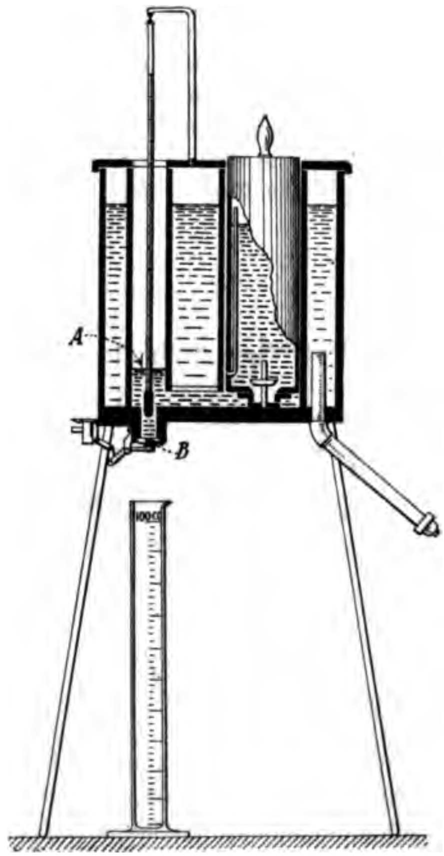

FIG. 3 ORIFICE TYPE VISCOSIMETER

Flowers meter, which employs a small steel sphere and a slanting tube.

16 *Class 4. Oscillating Disk or Cylinder.* These meters determine the viscosity in terms of the damping which the oscillating disk or cylinder experiences when placed in the liquid as compared with the damping when placed in the standard liquid. The Doolittle meter is an example of this class.

17 *Class* 5. *Rotating Disk or Cylinder.* This type determines the viscosity in terms of the speed of rotation of the disk or cylinder in the liquid under test as compared with the speed of rotation in the standard, the driving torque remaining constant. The Stormer meter is an example of this class.

DISCUSSION OF THE VARIOUS TYPES

18 The value of each of the above types, so far as accuracy is concerned, can be estimated by noting whether they operate in accordance with equation [1] or [2], but before speaking of the various types it should be noted that only equation [1] will give accurate comparative results.

19 Write this equation in the forms

$$\eta_x = F_x \cdot d_x/S_x \cdot V_x$$

where the subscript x refers to the substance and conditions met with in connection with the liquid to be tested, and

$$\eta_s = F_s \cdot d_s/S_s \cdot V_s$$

where the subscript s refers in a similar manner to the standard liquid. Solving these two equations for η gives

$$\eta_x = \eta_s \cdot S_s \cdot V_s \cdot F_x \cdot d_x/F_s \cdot d_s \cdot S_x \cdot V_x$$

an expression which simplifies to

$$\eta_x = \eta_s \, (V_s/V_x) \; (F_x/F_s)$$

if the same apparatus is used in both cases, whereas if we apply the same operation to equation [2] we get

$$\eta_x = \eta_s \, [F_x - M_x \cdot dV_x/dt_x)/(F_s - M_s \cdot dV_s/dt_s)] \, V_s/V_x$$

20 This expression cannot be simplified, for the expression within the brackets can never be made equal to unity. Moreover this expression is not constant, but varies with every liquid which is compared with the standard and with every change of temperature. It is therefore impossible to assign an accurate correction factor to an instrument which seeks comparative results if the instrument operates in accordance with equation [2], or, in other words, if there is any accelerated motion of the liquid during the testing operation.

21 In practice, the time, t, required for a definite volume of liquid to pass through the meter is measured instead of the velocity, V, and, if the force driving the liquid is gravity, the force F is easily

evaluated as $\rho \cdot h$, where ρ is the density and h the average head. Equations [1] and [2], in practice, thus become

$$\eta_x = \eta_s \cdot t_x \cdot t_s \cdot \rho_x \cdot \bar{h}/t_s \cdot \rho_s \cdot \bar{h} \quad \text{or} \quad \frac{\eta_x}{\eta_s} = \frac{\rho_x \cdot t_x}{\rho_s \cdot t_s} \ldots\ldots [1a]$$

$$\eta_x = \eta_s \cdot t_x/t_s[\rho_x \cdot \bar{h} - (M_x \cdot dV_x/dt_x)/(\rho_s \cdot \bar{h} - M_s \cdot dV_s/dt_s)] \ldots\ldots [2a]$$

THE ACCELERATION ERROR

22 It is obvious that equation [2a] applies to all meters coming under Classes 1 and 2 since the liquid starts from rest in the meter chamber and leaves with a certain velocity and must therefore experience acceleration. In all these meters on the market the acceleration term $(M \cdot dV/dt)$ is neglected since its evaluation is impossible and the formula then becomes identical with equation [1a]. An error which we shall call the acceleration error is thereby introduced.

23 The nature of this error can be predicted from an inspection of equation [2a]. The liquid having the smaller viscosity will pass through the meter with the greater velocity and the correction term $(M \cdot dV/dt)$ will be larger in proportion for this than for the more viscous liquid. If, therefore, water is chosen as a standard in determining the viscosity of lubricating oil or any liquid which is more viscous than itself, the value $[(F_x - M_x \cdot dV_x/dt)/(F_s - M_s \cdot dV_s/dt])$ will be larger than the value F_x/F_s and the value η_x as computed from the approximate formula [1a] will be too small. If, however, the liquid under test has a smaller viscosity than the standard the bracketed term will have a smaller value than F_x/F_s and the result as given by the approximate formula will be too large.

24 In the case of meters employing capillary flow, it is evident that the acceleration term will increase with the diameter of the capillary and decrease with its length and, as a result, we shall expect all meters using short capillaries to be subject to large error. Moreover, if we regard the orifice as a very short capillary, we should expect the error introduced in this type of meter to be still greater. The experimental results presented later prove these predictions to be correct.

SURFACE TENSION ERROR

25 Another error introduced in all the flow type viscosimeters, of both the capillary and orifice forms, is due to difference in the surface tension of the standard liquid and the liquid under

test. This error is prominent for those meters which discharge from the capillary or orifice into the air. As soon as the stream leaves the meter the film which forms on the free surface tends to contract, and thus decreases the cross-section of the stream and retards the discharge. This contraction is greater for a liquid of high than for one of low surface tension. If water is used as the standard, since its surface tension is greater than for oils the flow of water will be retarded more than the flow for oils. The time of flow, t_s, will be increased in greater proportion than t_x, thus giving the ratio t_x/t_s too small, and the error will cause the value η_x to be too small. For oils and most liquids the surface tension error and the acceleration error are additive.

26 This surface tension effect is, of course, negligible for a stream having high velocity, as the inertia of the moving mass prevents distortion of the stream lines. For the low velocities of efflux given by flow type viscosimeters, however, the effect is very noticeable, as can be seen by measuring the diameter of the efflux for two liquids of different surface tension. When conditions are most favorable for reducing the acceleration, namely velocity of efflux very small, they are most favorable for introducing the surface tension error.

CRITICAL VELOCITY ERROR

27 Another source of error in most capillary and orifice forms of viscosimeters is that the dimensions of the instrument are such that for the standard — water — the flow exceeds the critical velocity and the resulting turbulent flow makes the value of t abnormally large. This introduces an error, which also adds itself to the acceleration and surface tension errors. This fact was discovered during the research work about to be recorded, but has in the meantime been noted and investigated by Upton, who found the error so introduced by the Engler viscosimeter to be great.

28 The three errors — acceleration, surface tension and critical velocity — are prominent in all the meters named above as standards and we should expect the viscosity as given by these meters to be in general much too low. The experimental results will show that it is.

29 Those meters based on the principle of dropping a solid body through a tube filled with the liquid to be tested should be more accurate than the capillary or orifice forms as they are free from the surface tension error and usually from the critical velocity

error. All are, however, subject to the acceleration error. All the liquid displaced by the falling body suffers acceleration. These meters are all of the flow type and may be regarded as of the short capillary form, the reduced section at the point where the falling body is passing being the capillary. The liquid flows across this section as truly as it does through the orifice or capillary.

30 The relation between the damping which an oscillating disk or cylinder experiences and the viscosity of the liquid in which it is immersed is complicated, and indeed the true relation is not known. Meters based on this principle, such as the Doolittle meter, do not give accurate results. They are nearly free from the surface tension and critical velocity errors, however, and though somewhat slow and difficult to operate, they give better results than the majority of flow types.

31 The constant speed of rotation which a disk or cylinder immersed in a liquid will attain under the action of a constant torque is a true measure of the viscosity of the liquid. The Stormer viscosimeter attempts to operate in accordance with this principle, but the friction of the moving parts is necessarily such that a constant torque cannot be obtained. With proper refinement, this meter could be made to give good results. The fact that a meter based on this principle can theoretically give accurate results has led to the development of a new viscosimeter now to be described.

THE NEW VISCOSIMETER

32 This viscosimeter operates in accordance with the principle that a solid body having a surface of revolution experiences, when suspended in a rotating liquid, a torque which is proportional to the viscosity of the liquid. The instrument is shown diagrammatically in Fig. 4. The specimen, S, is contained within a cylindrical chamber which is caused to rotate uniformly by a motor, M, through a worm drive, R. A cylinder, C, is suspended within the specimen by a thin steel wire, W, so that the axis of the rotating liquid coincides with the axis of the cylinder. The specimen chamber is provided with a cap, V, so shaped that the excess liquid can overflow when the cap is seated and thus give constant conditions within the chamber. The specimen chamber is surrounded by an oil jacket, J, in which a thermometer, T, is suspended. The jacket oil may be brought to any desired temperature by means of a heating coil, or a side coil not shown in the diagram. The cover, P, of the jacket chamber

is provided with a scale which is marked in degrees or may be calibrated to read off directly, through the deflection of the pointer, *P*, the viscosity in terms of a standard liquid. The specimen chamber and the suspended cylinder are both made of copper to insure constant temperature throughout the specimen, and the outside of the specimen chamber is provided with blades which keep the jacket

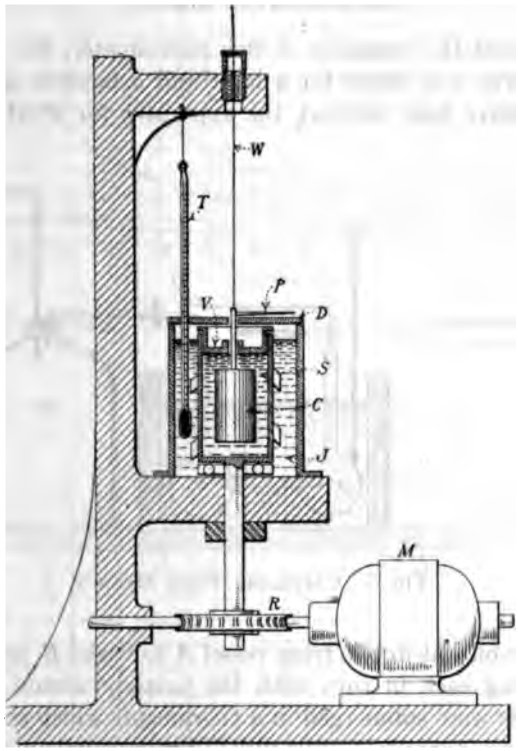

FIG. 4 THE NEW VISCOSIMETER

oil thoroughly mixed as the chamber revolves and thereby exposes the latter to a uniform temperature. This is an important factor toward insuring constant temperature throughout the specimen.

33 The experimental work has shown that the temperature of the specimen is uniform to within a small fraction of a degree. Moreover, the temperature of the specimen follows the temperature of the jacket oil so closely that the temperature-viscosity curve can be taken while the temperature is slowlv raised or lowered. This

proves to be a great saving of time. It also saves labor, for one does not need to stand by the instrument continually. The deflection of the pointer is at any instant a measure of the viscosity, so all that is required is to take simultaneous readings of temperature and deflection at intervals during the heating or cooling process.

EXPERIMENTAL RESULTS

34 To test the accuracy of this viscosimeter, the temperature-viscosity curve was found for a light and a medium gas engine oil by the capillary tube method, the apparatus for which is shown in

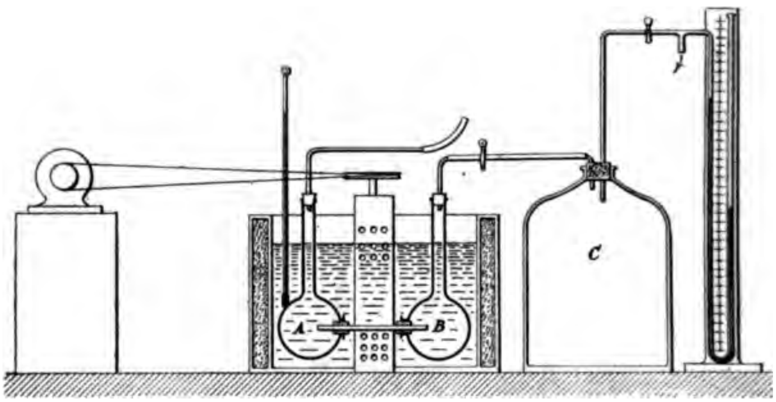

FIG. 5 CAPILLARY TUBE METHOD

Fig. 5. The oil was drawn from vessel A to vessel B, and vice versa, by connecting each in turn with the partial vacuum chamber, C. These vessels were submerged in a thermostat which could be maintained at any desired temperature. The results, in terms of the viscosity of water at 20 deg. cent., are given by the points enclosed in circles for curves 1 and 1*a*, Figs. 6 and 7, the latter referring to the lighter oil. The viscosity of these same oils as given by the new meter is represented on curves 1 and 1*a* by the points enclosed in squares. The agreement between the two methods is almost perfect.

35 The viscosity of these oils as given by a meter of the short capillary type, Fig. 2, is given in curves 2 and 2*a*. Curves 3 and 3*a* give the viscosity of these oils as obtained with a meter of the orifice form, Fig. 3. As predicted, the results given by the short capillary

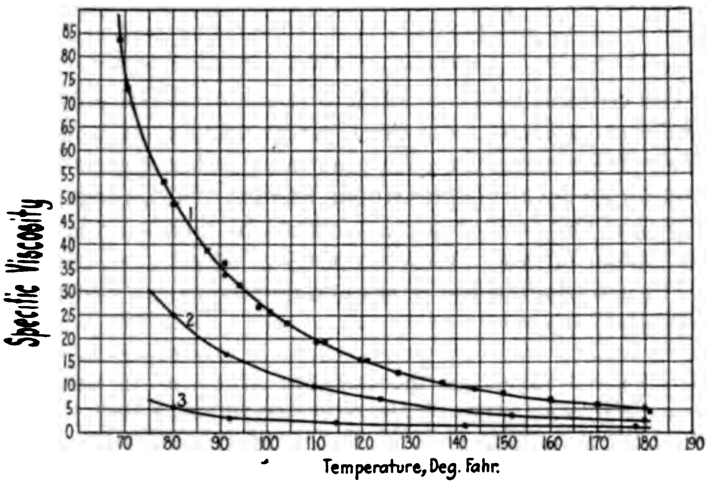

Fig. 6 Viscosity Curves for Medium Gas Engine Oil

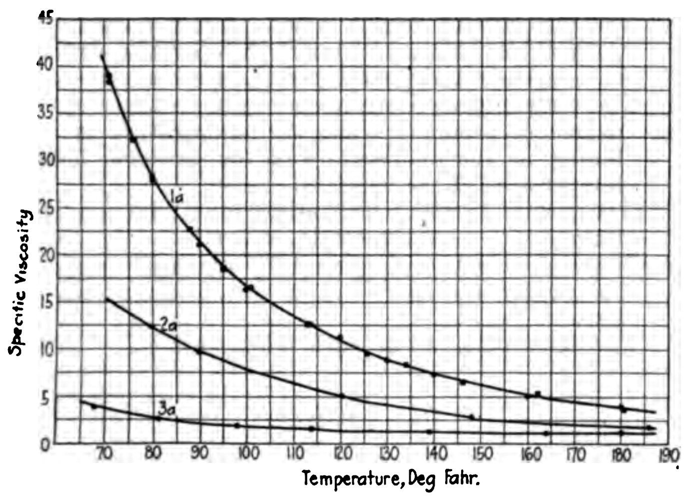

Fig. 7 Viscosity Curves for Light Gas Engine Oil

type are much too low (on the oils tested about 100 per cent too low) and the results given by the orifice type are still lower.

CONCLUSIONS

36 The errors inherent in all flow types of viscosimeters have been predicted and the predictions verified by experiment. The magnitudes of the errors are such that these meters cannot be depended on for giving even approximate results. The various meters of this type on the market do not give results which are in agreement, and no one of these meters can justly be claimed as a standard.

37 The new viscosimeter of the non-flow type described has the following advantages:

1 It gives values for the viscosity which are in agreement with those given by the standard capillary tube method.
2 During a series of tests at various temperatures, the oil is not handled.
3 The sensitiveness of the instrument can be made anything desired by changing the speed of rotation of the specimen or by using suspension wires of various diameters.
4 The density or change in density is not a factor in computing the results, in fact the instrument may be graduated to read off the viscosity directly.
5 The viscosity of liquids which contain particles in suspension can be measured, and the operation of the meter is independent of the color of the specimen.
6 The temperature-viscosity curve can be taken with a fair degree of accuracy while the temperature is rising or falling, as the temperature of the specimen follows the temperature of the jacket so closely.
7 The personal error which arises in determining time intervals with a stop watch is removed.
8 The instrument is simple, rigid and self-contained. It has no separate parts to get lost or glass parts to get broken.
9 The instrument can be made self-recording.

DISCUSSION

Winslow H. Herschel[1] (written). The authors practically admit that the long capillary viscosimeter is the most accurate instrument for determining viscosity. No objection can be raised against this method on grounds of accuracy, neither is it necessarily

[1] Chevy Chase, Md.

"complicated and laborious." Couette, who experimented with both the capillary and concentric-cylinder methods, regarded the capillary method as the more exact, and ascribed the error in the latter method to eccentricity. As far as can be determined from Fig. 4, I should judge that this error would be greater in the authors' instrument than in that of MacMichael,[1] in which the shear in the liquid is mainly between the bottom of a suspended disk and the bottom of the revolving cup. I believe there is more need of improving old forms of instruments and methods of calculating viscosity than there is in devising new instruments. What is required is an improved, long capillary viscosimeter which is self-contained and portable; and the abandoning of such instruments, formulæ and methods as lead to the expression of results in any other terms than of absolute viscosity. The revolving-cup type of instrument may have a field of usefulness in commercial work, where quickness of determination is of more importance than extreme accuracy, but it is handicapped by its greater cost as compared with a capillary-tube instrument.

EUGENE C. BINGHAM[2] (written). The great misfortune of technical viscosity is that the results are not expressed in terms of absolute viscosity. It is an advantage of the proposed instrument that it can be calibrated to read in terms of absolute viscosity, as is true of the MacMichael instrument. It is the more unfortunate that the authors have expressed their results in other than absolute units.

It is also an advantage of this type of instrument that an accurate determination of the density does not have to be made before the absolute viscosity can be determined.

The revolving-cylinder and -disk methods have been found to be less simple mathematically and less accurate experimentally than the capillary-tube methods. Before the use of any one of these instruments becomes established, it should be shown that it will reproduce the absolute viscosity of liquids of widely different, but accurately known, viscosities. The error due to kinetic-energy correction, eddy currents, lack of perfect centering, etc., should be carefully studied.

The tests which the authors have made are not convincing. They refer to the loss of pressure in giving kinetic energy to the liquid as the *error*, which is quite confusing. In instruments of the Engler

[1] Journal of Industrial and Engineering Chemistry, Nov. 1915.

[2] U. S. Bureau of Standards, Washington, D. C.

and Saybolt types the times of flow are simply compared to the times of flow of water. Since these are arbitrary numbers there is no *error* due to acceleration. There *are* sources of error in these instruments amounting to two or more per cent, due to lack of perfect temperature control, etc., but that is quite different from the 100 per cent *error* mentioned. The authors use the term *error* as synonymous with what we may better term *kinetic-energy correction.* Their Figs. 6 and 7 indicate the different values of the kinetic-energy correction for different instruments, which would have been brought out clearly had they at some point in their paper expressed their results in terms of absolute viscosity. The fact that the curves for the capillary tube and the revolving cylinder are nearly identical would appear to mean that the kinetic-energy corrections in both cases are nearly the same. It certainly does not prove that the kinetic-energy correction is negligible in either case.

Instruments of this type have often been suggested for use with suspensions. Such use should be made with caution, since, contrary to the statement in the paper, plastic substances do not "exhibit a gradual yielding to tangential forces" unless the tangential force exceeds a certain limit. Suspensions may be plastic, in which case the results with instruments of this type may not be interpretable on the ordinary assumptions.[1]

The authors (Figs. 6 and 7) have made the common mistake of regarding *times of flow* as proportional to viscosity. It cannot be emphasized too strongly that this is not the case, although apparent authority for such a statement may be found again and again in the literature of the subject. It is rarely that the serious consequences of this erroneous assumption are so clearly apparent.

E. H. Peabody (written). The determination of the viscosity of oils used for fuel purposes is becoming a matter of increasing importance, owing to the rapidly growing use of the so-called mechanical atomizer. This type of oil burner sprays the oil by means of centrifugal force which is imparted to it by forcing the liquid through small passages tangential to a central chamber, from which it issues from an orifice the center of which usually coincides with the axis of rotation. Thus, the viscosity of the liquid has a very direct and important bearing on the satisfactory spraying of oil by this method, and usually the viscosity of the oil has to be reduced by heating.

[1] Bulletin 278, Bureau of Standards, Washington, D. C.

A knowledge of the temperature-viscosity properties of fuel oils is almost a necessity to the users of plants in which the mechanical atomizer is installed.

There will be little gain, at least in so far as the users of fuel oils are concerned, if the net result of the work of the authors is to impose upon the unsuspecting public still another and additional *scale of viscosities.*

Almost any one of the instruments named by the authors, particularly those which have been adopted as standards by various interests (as the Saybolt, Engler, Redwood, etc.), would be entirely satisfactory if everybody used it. If, however, different standards are used, each having its own individual meaning, great confusion will result; and this, unfortunately, is the case with regard to viscosity. Thus, viscosity measured on the Saybolt scale is a very different thing from that measured on the Engler or Redwood, and it becomes necessary to specify whether the results are given in one or the other scale. Further, there appears to be no very definite way of making comparisons between the various scales, due, doubtless, to the very errors which the authors of this paper so clearly point out.

There is a crying need today of some international standard for the measurement of viscosity, and it is to be hoped that either this Society or some other will initiate some movement which will result in this very desirable agreement of all authorities.

In this connection, a very interesting suggestion will be found in the discussion of viscosity in a work entitled An Examination of Hydrocarbon Oils, translated from the German of Prof. D. Holde by Dr. Edward Mueller, and published by John Wiley & Sons, 1915. In this book a formula devised by Ubbelohde is described, by which viscosity in degrees Engler may be reduced to *absolute viscosity.* The suggestion is made that the viscosity of liquids be reported in terms of the viscosity of water; in other words, that the absolute viscosity of water be taken as unity, and the absolute viscosity of other liquids (having been determined by the Engler instrument and corrected by Ubbelohde's formula, or by some other method) be reported in terms of these units as *specific viscosity.* This seems to indicate a very satisfactory solution of the matter, and is worthy of consideration as an international standard. Why would it not be quite as satisfactory to report absolute viscosity of liquids in terms of the absolute viscosity of water, exactly as we take water as a basis in *specific gravity, specific heat,* etc.?

M. D. HERSEY (written). Mr. Hayes misunderstands Mr. Bingham's phraseology in regard to the plasticity of suspensions, which refers to samples of suspensions under test, not to the torsion wire itself. Mr. Bingham's conception of the threshold stress below which plastic substances do not exhibit ordinary viscous flow, is an important one, more fully set forth in Bureau of Standards Scientific Paper No. 278.

The authors' treatment of the errors of short capillaries overlaps considerably the treatment previously published by Flowers in a paper of the same title.

In the course of experiments on the change in viscosity under pressure, made at Harvard University during the past year, the writer has employed a torsion viscosimeter of which the outer cylinder consists of an ordinary test tube. Thus 30 cc. is a sufficiently large sample; and the container need not be cleaned, for, in changing samples, one test tube is removed and another inserted. The inner cylinder is slim enough so that end effects are small, and, therefore, the instrument can be designed mathematically so as to have any desired characteristics. A brief account of this work will be found in the September 19, 1916, issue of the Journal of the Washington Academy, under the title The Theory of the Torsion and Rolling-Ball Viscosimeters, and Their Use in Measuring the Effect of Pressure on Viscosity.

The paper of Hayes and Lewis is one of the most sensible and useful papers relating to lubrication which have come to the writer's attention; and especially so because of the vivid manner in which Figs. 6 and 7 draw attention to the fact, which ought to be more familiar than it is, that the viscosity scales of short capillary and orifice instruments are far from proportional to efflux times.

R. F. MACMICHAEL[1] (written). The original revolving-cup torsion viscosimeter seems to have been designed by Couette about 1890. The essential feature of this instrument, as distinguished from others of the same general type, is the use of stationary protecting caps over each end of the suspended cylinder to eliminate "end effects," thus rendering the calculations quite simple.

This is a high-grade scientific design, and, when properly constructed and operated, should be capable of giving exceedingly accurate results, expressed directly in C. G. S. units. However,

[1] Fort Pitt Hotel, Pittsburgh, Pa.

owing to the high cost and the number of parts required, the instrument is hardly practical for general commercial use.

Several years ago the MacMichael viscosimeter was designed without previous knowledge of similar efforts along this line for practical use in a clay-working establishment. This instrument is quite simple in construction, is durable, and is easily and quickly manipulated. One of these machines has been in constant use for several years, in the hands of unskilled laborers, and has given excellent service. Under these conditions readings ordinarily check within 1 per cent.

No attempt is made to calculate viscosities directly from measurements of the machine, as this would be extremely difficult, if not impossible. The method used is to calibrate the instrument with a standard testing sample of known viscosity, and assume that other readings are directly proportional in value. Through a wide range this does not seem to introduce serious errors.

For a standard testing fluid, a high-grade neutral mineral oil of high viscosity, such as some of the medicinal oils now on the market, is suggested, this oil to be supplied in metal containers by the U. S. Bureau of Standards, or some other recognized authority. The viscosity of these testing samples may be determined by means of the long capillary tube, which is sufficiently accurate for all practical purposes.

A cane-sugar solution of 7 parts by weight of sugar to 4 parts of cold water also gives good results. This should be tested after mixing, not depending on the weighed proportions to produce accurate results. Owing to its stickiness and tendency to form incrustations, this fluid is not so agreeable to use as an oil, but otherwise its properties are excellent.

In any type of instrument, whether capillary or torsional, a critical condition occurs at which turbulence takes place, and incorrect results are obtained. Reducing the speed, reducing the dimensions of parts, or increasing the viscosity of the fluid tends to check this action.

Also, in a freely suspended torsional instrument, at certain speeds and viscosities a pendulum-like oscillation of the suspended member occurs; this may increase rapidly and with violence, ruining the torsion wire unless checked. In the MacMichael instrument this is prevented by a suitably arranged oil-filled dashpot, which also acts as a damper, to improve the general action of the suspended member.

The "spiral action" referred to in the paper takes place also in the Hayes-Lewis machine, but as it occurs in a closed vessel it is not so easily seen. However, since this effect occurs equally with both the standard testing fluid and the sample being tested, and as the action itself is very moderate indeed, the errors introduced cannot be very serious. These spiral "end effects" are eliminated only in the Couette machine, as noted above.

The Hayes-Lewis machine appears to the writer to have neither the high-grade scientific features of the Couette instrument, nor the practical work-a-day features of some of the simpler designs.

Torsion viscosimeters probably have an important field in colloidal chemistry, and in the measurement of suspensions, such as clay slips. Likewise, they are well adapted for the testing of oil under unusual conditions, such as at very high or very low temperatures. When properly built they are rapid, accurate, and easily manipulated. However, they are of necessity quite expensive, so that unless a very radical change occurs in public opinion, or a demand arises for a standard testing machine reading in absolute units, it is very doubtful whether they will ever supplant the short-tube commercial instruments now in general use.

The Authors. In reply to Mr. Herschel, the authors do not "practically admit" that the long capillary method for determining viscosity is the most accurate. They simply claim this is the best absolute method for determining the viscosity of some standard liquid, but there are other methods, comparative methods, which determine the viscosity of an unknown liquid in terms of the viscosity of a standard liquid with less error than is introduced by the long-capillary-tube method in determining the viscosity of the standard liquid.

The long-capillary-tube method is certainly complicated and laborious. I have found from practice that several days are required to secure the data for a temperature-viscosity curve for a single oil. It must be remembered that the relation between the head and the tube dimensions must be such that the flow will never exceed the critical velocity, and this requires that the time of flow for the more viscous oils and for all oils at low temperatures will be abnormally great. Moreover, these capillary tubes are difficult to clean and hard to duplicate.

In regard to the MacMichael instrument, the lines of flow above and below the disk are spirals, and not circles. This is due to the centrifugal action of the rotating liquid, and as a result an error is introduced, due to the fact that the liquid undergoes acceleration. This error is not great, but, such as it is, it is largely avoided in the Hayes-Lewis meter, where a cylinder is used instead of a disk. In this instrument the ends of the cylinder do not approach the top and bottom of the specimen chamber closely, and as a result most of the viscous forces are confined between the cylindrical surfaces of the cylinder and the chamber. The stream lines within this region are true circles and the liquid undergoes no acceleration, and as a result the small acceleration effect at the ends is negligible. This is one of the reasons for our using a cylinder instead of a disk.

For most engineering purposes the specific viscosity (absolute viscosity divided by absolute viscosity of water at 20 deg. cent.) will serve fully as well as the viscosity expressed in absolute units. The specific viscosity is readily transformed to absolute units whenever such units are required, and the calibration of viscosimeters is somewhat simplified by calling the viscosity of water unity.

Replying to Mr. Bingham, no claim has been made that the Hayes-Lewis viscosimeter is more simple than the long capillary tube as an absolute instrument. It certainly is less simple mathematically, but this fact does not prevent the instrument from giving excellent comparative results.

In regard to the reproduction of the absolute viscosity of liquids of widely different but accurately known viscosity, we have attempted to do this. We have worked with water, light and medium oils, and glycerine. In each case the absolute value was obtained by the long-capillary-tube method in accordance with Knibb's formula:

$$\eta = \pi hgr^4 \rho t/8lv - 1.12v\rho/8\pi lt$$

where the meaning of the various letters is in accordance with Par. 8 of the paper. The coefficient of water at 20 deg. cent. so obtained was within 0.5 per cent of the value obtained by Hosking, and we have assumed the values obtained for the oils and the glycerine by this same method were of the same degree of accuracy. The values so obtained were in each case divided by the value obtained for water. In the case of the oils these values are plotted as dots in curves 1 and 1*a*, Figs. 6 and 7. Then the deflection given by our meter when water at 20 deg. cent. was placed in the specimen chamber was taken by means of a mirror attached to the pointer, and telescope and scale. This angle was divided into the deflection angle given by the oils and glycerine. These values are plotted as squares on the same curves. The curves show almost perfect agreement between the two methods for the oils at various temperatures. The agreement in the case of glycerine was not quite so good, the greatest variation being 2 per cent. We were several days in securing these results and attributed this variation to the absorption of moisture by the glycerine. All told, it seems these results are convincing.

It is to be noticed that all kinetic-energy corrections were made in determining the absolute viscosities in accordance with Knibb's formula, and as the results given by our meter are in good agreement, the work does prove that the kinetic-energy correction is negligible despite Mr. Bingham's statement to the contrary.

Lack of perfect centering and small variation in height adjustment of the cylinder cause little or no error, for the reason that the viscous forces are inversely proportional to the thickness of the oil between the cylinder and container. The gain in viscous forces on one side of the cylinder, due to imperfect centering, is largely offset by the loss on the opposite side, and this is also true for vertical displacement, providing the displacement or lack of perfect centering is not great.

The authors have not claimed an error of 100 per cent or any other figure for the Engler or the Saybolt instruments when they are merely used to obtain Engler degrees or Saybolt seconds, respectively. Their paper, as the name suggests, deals with the measurement of viscosity. And when, as is usually done when viscosity is determined with these instruments the formula $\eta_s = \rho_s t_s t_w$ is used for computing the viscosity, the 100 per cent error mentioned will be given. This error is due to the kinetic-energy correction which is not corrected for and which is quite confusing. It is true that Ubbelohde's formula is useful in making this correction for the Engler instrument, and the recent work done by the U. S. Bureau of Standards giving the relation between the Engler and Saybolt instruments makes possible the correcting of the Saybolt instrument readings. Even if Ubbelohde's formula were correct (and it is only claimed to be approximate), still corrections that must be made in this roundabout way are confusing. There is great need for an instrument that gives results which require no correcting when absolute values of viscosities are required, and which gives at all times readings proportional to the absolute values.

Suspensions made of tempered steel may be plastic if the tangential forces brought into play exceed the elastic limit, but there is no reason why such forces should be brought into play. In fact, our instrument is so constructed that the suspension cannot be twisted beyond this limit.

A close reading of the paper will convince anyone that the authors have not made the mistake of regarding time of flow as proportional to viscosity. Our curves show clearly the serious consequences of this erroneous assumption, and we have regarded this as the chief value of our paper.

In reply to Mr. Peabody, we did not intend to impose on the unsuspecting public another meaningless viscosity scale. In fact, when trying to interpret the results given by various meters we have found the same confusion that he has apparently found. We have also felt the need of some standard unit to which all readings should be reduced, and we decided that the best unit would be that of water at 20 deg. cent. (the same as Mr. Peabody has suggested). The ordinates of our curves are given as the values of the absolute viscosity of the oils divided by the absolute values of the viscosity of water at 20 deg. cent.

No. 1541

ON THE TRANSMISSION OF HEAT IN BOILERS

By E. R. Hedrick,[1] Columbia, Mo.
Non-Member

and

E. A. Fessenden, Columbia, Mo.
Member of the Society

The purpose of this paper is to show the inadequacy of certain current theories of heat transmission in boilers[2], and to outline a theory which agrees with known experiments to within reasonable limits of possible errors. The search for a new formula to express the drop in temperature along the boiler tube resulted from difficulties experienced by the authors in attempting to check the theoretical formulæ of Sir John Perry and others with the results of actual experiments quoted by Perry himself and by others.

2 Thus, Perry[3] gives a formula for the efficiency E of a tube of length l as

$$E = 1 - e^{-cl/d} \qquad [1]$$

where e is the Naperian base 2.71828

c is a constant

d is the diameter of tube.

3 For any particular tube, the term c/d is then a constant, which may be called k; and if the distances along the tube are denoted by x, as elsewhere in the present paper, this equation may be written in the form

$$E = 1 - e^{-kx}, \quad \text{or} \quad \log_{10}(1 - E) = - Kx$$

where $K = k \log_{10} e$

[1] Dept. of Mathematics, Univ. of Mo.

[2] See for example Perry, The Steam Engine, 1909; and Bulletin 18 of the U.S. Bureau of Mines, by Kreisinger and Ray, entitled The Transmission of Heat into Steam Boilers.

[3] Perry, l.c., p. 591.

Presented at the Spring Meeting, New Orleans, La., April, 1916, of The American Society of Mechanical Engineers.

4 In another place[1] Perry gives the actual values of E, according to his own computations for the famous French experiments. These values, and the corresponding values of $1 - E$ and $\log_{10}(1 - E)$ which we computed in order to check with Perry's formula [1], are given in Table 1.

TABLE 1 PERRY'S VALUES FOR E, FRENCH EXPERIMENTS

COAL..........	x in feet	0	3	6	9	12
	E	0.000	0.290	0.439	0.525	0.580
	$1-E$	1.000	0.710	0.561	0.475	0.420
	$\log_{10}(1-E)$	0.000	9.8513	9.7490	9.6767	9.6232
	K	....	0.0496	0.0341	0.0241	0.0178
COKE..........	E	0.000	0.334	0.501	0.595	0.658
	$1-E$	1.000	0.666	0.499	0.405	0.342
	$\log(1-E)$	0.000	9.8235	9.6981	9.6075	9.5340
	K	....	0.0588	0.0418	0.0305	0.0212

5 In the last row of each set of values are given the computed values of K; these are found by dividing the differences between successive values of $\log_{10}(1 - E)$ by the corresponding differences in x.

6 A different set of values K can also be obtained by direct substitution in the equation

$$-K = \frac{\log_{10}(1 - E)}{x}.$$

These values are given below for the same points as in Table 1:

x in feet	0	3	6	9	12
K (coal tests)	....	0.0496	0.0418	0.0359	0.0314
K (coke tests)	....	0.0588	0.0503	0.0436	0.0387

7 The authors feel that the method of differences used in Table 1 is preferable to the second method, since the second method counts from the fire-box end of the tube for each computation, and gives excessive weight to that particular portion of the tube. The first method takes each section of the tube separately and the value of K found applies to that particular section of the tube. The second method gives a value of K for length of the tube from the fire-box end to the end of the section considered.

[1] Perry, The Steam Engine, 1909, p. 432.

8 If K were constant, as it should be if Perry's formula holds, the two methods would of course give identical results. It will be seen, however, that the values of K are not constant within any reasonable limits, for either set of experiments, or for either method of computing. Instead, the values of K diminish steadily as x increases, and the last value is only about one-third of the first value.

Fig. 1 shows clearly the same discrepancy. In it, the values of $\log_{10}(1 - E)$ are plotted against the values of x. If Perry's formula

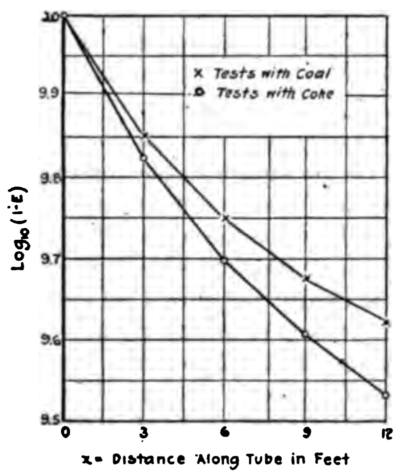

FIG. 1 FRENCH EXPERIMENTS, PERRY'S FORMULA

[1] held, the resulting figure should be a straight line, which it clearly is not. Unfortunately, the data given in these experiments are not sufficient to enable the authors to check against the formulæ given later in this paper.

9 The similar data given by Kreisinger and Ray[1] show quite as great discrepancy. A considerable number of other sets of data of the same nature are given below, and these are compared in each case both with the formulæ of Perry and with those of this paper.

10 Perry's work is based on that of Osborne Reynolds. He makes the fundamental assumption that the amount of heat δH transmitted across a small length δx of the tube is proportional to the difference in temperature τ between the gas and the water, and to the area of the tube wall:

$$\delta H = b\tau\, \delta x \qquad [2]$$

[1] Bulletin 18, U. S. Bureau of Mines, p. 53, Fig. 16.

where b is a constant.[1] This amounts to saying that the conductivity $\delta H/\delta x$ is a constant. From this he derives[2] the formula

$$\theta = \theta_1 \, e^{-cx/d}$$

where θ and θ_1 are equivalent to our τ and τ_0 and where c/d is a constant which we shall call c. Then,

$$\log \frac{\tau}{\tau_0} = -cx, \quad \text{or} \quad \tau = \tau_0 e^{-cx} \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

where c is a constant, τ_0 is the value of τ at the fire-box end of the tube, and x is the distance along the tube, measured from the fire-box end.

11 As noted above, these formulæ prove inadequate to express the results of experiments known to the authors. A very reasonable ground for the existence of such a discrepancy is to be found in the assumptions made by Perry on the constancy of the heat conductivity, or its reciprocal, the heat resistance.

12 Indeed, it is clearly recognized by everyone[3] that the principal sources of heat resistance are the peculiar state and the peculiar behavior of the gas and of the water next to the tube, rather than the intrinsic resistance of the tube wall itself. The total resistance is of the order of one thousand times the intrinsic resistance of the tube wall alone.

13 The nature of this extra resistance is indeed occult, but it is clear at least that the water immediately adjoining a hot tube would tend to assume an abnormal state, although a film of steam cannot be said to exist about the tube. In any event the water, if it is actually water, that lies against the tube wall constitutes a film of peculiar behavior. The gas inside the tube tends to form a more or less dead film next to the tube wall; the behavior and amount of this action depends upon the speed of the gas as well as upon the physical condition of the tube wall. These facts are well known and are clearly recognized, but their effect is apparently neglected.

14 All the phenomena just mentioned evidently depend upon the temperature. Even the speed of the gas depends upon the temperature, since the gas contracts as it cools. The dependence of the

[1] The constant b depends upon and changes with the diameter of the tube, the gas velocity, and other variable phenomena connected with the experiment, but is supposed to be the same throughout any one experiment.

[2] Perry, The Steam Engine, 1909, pp. 587–591.

[3] Perry, The Steam Engine, 1909, p. 593.

other phenomena is too evident to deserve explanation. It follows that the resistance itself depends upon the temperature, if we mean by resistance that total resistance mentioned above rather than the intrinsic resistance of the metal alone. This explains why a discrepancy should be present in the Perry theory, and it furnishes a basis for the assumptions to be made.

15 It is assumed that the total resistance changes according to a law which will be found later. The fundamental assumptions may be stated in several different forms. That started with originally is the assumption that as we proceed down the tube the amount of the heat loss from a small quantity of gas δQ occupying a length δx of the tube, in passing a given point, falls off according to the usual exponential formula for any damping out process; that is, that

$$\textit{loss of heat in } \delta Q = ce^{-mx} \qquad [4]$$

where m and c are constants.

16 This assumption seems to us to be a reasonable one, but its real justification occurs later, in that the consequences of this assumption seem to agree with experiments. Any assumption must depend for its validity on such an agreement. In itself it is at least as reasonable as that of Perry. Since we have worked out the resulting forms, it appears to us to be both reasonable and simple to take as our fundamental assumption another statement which is precisely equivalent to the preceding one.

17 Let us consider the difference $\phi - \phi_w$ between the entropy ϕ of the gas at gas temperature and the entropy ϕ_w which the same gas would have at the temperature of the water. *We shall assume that the decrease $\delta\phi$ in the entropy of the gas in a small section of the tube of length δx is proportional to the entropy difference $\phi - \phi_w$.* It will appear later that this assumption is indeed equivalent to that mentioned above.

18 Since ϕ_w is a constant, it follows that $\delta\phi_w$ is zero. Hence $\delta\phi$ is equal to $\delta(\phi - \phi_w)$, and the fundamental assumption may be written in the form

$$\delta(\phi - \phi_w) = -m(\phi - \phi_w)\,\delta x \qquad [5]$$

where m is a positive constant. We shall use also the well known relations

$$\delta\phi = \delta H/\theta \qquad [6]$$

$$H = c_p\,\theta \qquad [7]$$

$$\delta\phi = c_p\,(\delta\theta/\theta) \qquad [8]$$

where θ is the absolute temperature
H the heat content of unit weight
c_p the specific heat at constant pressure.

19 We shall assume that c_p is a constant, though a slight variation is known to exist; and we shall assume that the pressure is the same throughout the tube, though some variation undoubtedly occurs. Variations in c_p and in p are probably of slight effect in comparison with other sources of error, and the known facts would not seem to justify the use of any other definite statement for either.

20 From [8] we find

$$\phi - \phi_w = c_p \log_e \frac{\theta}{\theta_w} \qquad [9]$$

and therefore [5] becomes, in the limit as δx approaches zero,

$$\frac{d \log_e (\theta/\theta_w)}{dx} = - m \log_e (\theta/\theta_w) \qquad [10]$$

21 If we denote $\log_e (\theta/\theta_w)$, which is proportional to $\phi - \phi_w$, by s, we have

$$\frac{ds}{dx} = - ms \qquad [11]$$

whence by integration

$$\log_e s = - mx + \kappa \quad \text{or} \quad s = ke^{-mx} \qquad [12]$$

where κ and k are constants, and where $\kappa = \log_e k$. If $x = 0$, s has the value $s_0 = \log_e (\theta_0/\theta_w)$, where θ_0 is the absolute temperature of the gas as it enters the tube. Hence, inserting these values, $k = s_0$ or $\kappa = \log_e s_0$; and [12] may be written in the form

$$\log_e (s/s_0) = - mx \quad \text{or} \quad s = s_0 e^{-mx} \qquad [13]$$

In terms of θ these formulæ are

$$\log_e (\theta/\theta_w) = \log_e (\theta_0/\theta_w) e^{-mx} \quad \text{or} \quad \theta/\theta_w = (\theta_0/\theta_w)^{e^{-mx}} \qquad [14]$$

22 For comparison with numerical data it is desirable to have these formulæ expressed in common logarithms. Remembering that

$$\log_{10} N = \log_e N \times \log_{10} e$$

we may write the formula [14] in the form

$$\log_{10} (\theta/\theta_w) = \log_{10} (\theta_0/\theta_w) e^{-mx} \qquad [15]$$

Let us set

$$R = \log_{10} (\theta/\theta_w) \quad \text{and} \quad R_0 = \log_{10} (\theta_0/\theta_w)$$

so that R is proportional to s used above, or to $\phi - \phi_w$. Using this notation, and taking common logarithms again, we have

$$\log_{10} R = K - Mx \qquad [16]$$

where

$$K = \log_{10} R_0 = \log_{10} [\log_{10} (\theta_0/\theta_w)], \quad \text{and} \quad M = m \log_{10} e$$

In the numerical computations of this paper the quantities θ, θ/θ_w, R, and $\log_{10} R$ are tabulated in that order. Denoting the last of these by Θ:

$$\Theta = \log_{10} R = \log_{10} [\log_{10} (\theta/\theta_w)] \quad \text{[17]}$$

and we have

$$\Theta = K - Mx \quad \text{[18]}$$

where K and M are constants defined above.

23 In order to check any set of numerical data with this formula, all that is necessary is to calculate Θ by [17], and then to plot the values of Θ against the corresponding values of x; the resulting figure should be a straight line.

TABLE 2 COMPARISON OF PERRY'S, KENT'S AND AUTHORS' VALUES

τ	$\log \tau$	$\theta = \tau + 845$	$\log \theta$	$R = \log (\theta/\theta_w)$	$\Theta = \log R$
1207	3.98171	2052	3.31218	0.38522	9.5857
1020	3.00860	1865	3.27068	0.34382	9.5363
889	2.94890	1734	3.23905	0.31219	9.4944
786	2.89542	1631	3.21245	0.28559	9.4558
700	2.84510	1545	3.18893	0.26207	9.4185
624	2.79518	1469	3.16702	0.24016	9.3806
558	2.74663	1403	3.14706	0.22020	9.3428
499	2.69810	1344	3.12840	0.20154	9.3043
449	2.65225	1294	3.11193	0.18507	9.2674
405	2.60746	1250	3.09691	0.17005	9.2305
364	2.56110	1209	3.08243	0.15557	9.1919
328	2.51587	1173	3.06930	0.14244	9.1536
295	2.46982	1140	3.05690	0.13004	9.1140
266	2.42488	1111	3.04571	0.11885	9.0750
245	2.38917	1090	3.03743	0.11957	9.0437
224	2.35025	1069	3.02898	0.10212	9.0091

$\theta_w = 845$ $\quad \log \theta_w = 2.92686$

24 Through the kindness of William Kent we have one set of direct temperature measurements in a locomotive boiler tube at every foot of its length. These measurements, and the corresponding values of the quantities R and Θ are tabulated in Table 2. The points in the graphical figure for pairs of values of x and Θ are shown by small circles in Fig. 2. In the same table the values of τ and $\log_{10} \tau$ for comparison with Perry's formula [3] are given, and the corresponding points for pairs of values of $\log_{10} \tau$ and x are marked

by small crosses in Fig. 2. Were formula [3] correct, these points would lie on a straight line. The usefulness of such a method of plotting is evident, since discrepancies from a straight line are so readily seen. It will be noticed, of course, that the points marked by small circles lie in a straight line much more nearly than those marked by small crosses; this means that the formulæ [14]–[16] agree with the observed values better than does the formula [3].

25 Several similar sets of data were kindly furnished us by Chas. D. Young, Engineer of Tests for the Pennsylvania Railroad.

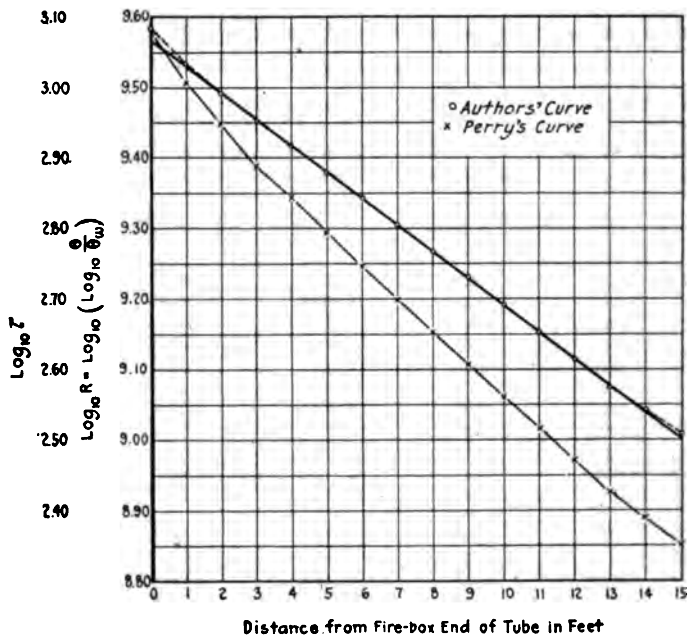

FIG. 2 KENT'S DATA CHECK AGAINST PERRY'S AND AUTHORS' FORMULÆ

These are given in Tables 3, 4, 5, 6, and the figures, both for comparison with our formulæ [14]–[16] and with Perry's formula [3] are plotted in Figs. 3, 4, 5, 6, with the same markings as in Fig. 2.

26 These figures speak for themselves. Particularly in view of the great difficulty experienced in making such measurements, the agreement with formulæ [14]–[16] is remarkably good, and any variations from a straight line are apparently at random rather than consistent. On the other hand, the points marked by small crosses, for comparison with Perry's formula, seem to show a consistent tendency to be low in the middle and high on both ends.

27 Perhaps the chief source of error in taking such measurements is the radiation of heat from the thermal couple itself to the relatively cold surface of the surrounding tube wall. The error due to this cause is not inconsiderable; it may run as high as 100 deg. fahr. and it is difficult to estimate because it depends on a number of uncertain factors.

28 It should be remarked also that the very first and the very last of the measurements may be relatively high, since the radiation from a thermal couple at the end of a tube is effective only through half the spherical angle about the point, whereas a couple located well within the tube radiates to the cold iron through practically the entire spherical angle. Evidences that this effect does exist are present in all these diagrams. It is certainly reasonable to give rather less weight to these two extreme points, but the next to the last on either end should show only random errors.

29 Another confirmation of the preceding theory is furnished by a series of tests which Professor Fessenden has been making. While this series as a whole is not yet completed and cannot now be published in all detail, a large number of individual tests are complete with respect to the data necessary for comparison with the theory of this paper.

30 In these tests, the quantities measured are the temperatures at the ends of the tube, the weight of gases passing per minute, the analysis of the gases, and the actual heat loss (in B.t.u.) for each foot length of the tube. The heat measurements are free from the radiation error mentioned above, and they should be more accurate. The apparatus used and the full details of all the results will be published separately later.

31 To convert the preceding equations into forms which can be compared with these measurements, let us return to equation [9], and let us substitute in it from [7] where H means the heat content reckoned from $H = 0$ at $\theta = 0$. We have then

$$\phi - \phi_w = c_p \log_e (H/H_w) \quad \ldots\ldots [19]$$

and the further reduction is precisely similar to that above, with the ratio H/H_w in place of the ratio θ/θ_w. Hence we have, as in [14]

$$\log (H/H_w) = \log_e (H_0/H_w)\, e^{-mx}, \quad \text{or} \quad H/H_w = (H_0/H_w)^{e^{-mx}} \quad [20]$$

and also, as in [16]

$$\log_{10} [\log_{10} (H/H_w)] = K - Mx \quad \ldots\ldots [21]$$

where

$$K = \log_{10} [\log_{10} (H_0/H_w)], \text{ and } M = m \log_{10} e \quad \ldots\ldots [22]$$

TABLE 3 COMPARISON OF PERRY'S, YOUNG'S AND AUTHORS' VALUES

Young's Values from Penna. R.R., Loco. No. 377

τ	$\log \tau$	$\theta=\tau+844$	$\log \theta$	$R=\log(\theta/\theta_w)$	$\Theta=\log R$
1002	3.0009	1846	3.26623	0.33989	9.5313
822	2.9149	1666	3.22167	0.29533	9.4703
762	2.8820	1606	3.20575	0.27941	9.4462
667	2.8241	1501	3.17638	0.24994	9.3978
527	2.7218	1371	3.13704	0.21070	9.3237
482	2.6830	1326	3.12254	0.19620	9.2927
417	2.6201	1261	3.10072	0.17438	9.2416
392	2.5933	1236	3.09202	0.16568	9.2193
347	2.5403	1191	3.07591	0.14957	9.1749
337	2.5276	1181	3.07225	0.14591	9.1641
297	2.4728	1141	3.05729	0.13095	9.1172
242	2.3838	1086	3.03583	0.10949	9.0394
227	2.3560	1071	3.02979	0.10345	9.0149
177	2.2480	1021	3.00903	0.08269	8.9175
167	2.2227	1011	3.00475	0.07841	8.8944
162	2.2095	1006	3.00260	0.07626	8.8823

$\theta_w=844$ $\log \theta_w=2.92634$

TABLE 4 COMPARISON OF PERRY'S, YOUNG'S AND AUTHORS' VALUES

Young's Values from Penna. R.R., Loco. No. 384

τ	$\log \tau$	$\theta=\tau+844$	$\log \theta$	$R=\log(\theta/\theta_w)$	$\Theta=\log R$
1477	3.16938	2321	3.36568	0.43934	9.6428
1242	3.09412	2086	3.31931	0.39297	9.5943
1022	3.00945	1866	3.27091	0.34457	9.5373
977	2.98989	1821	3.26031	0.33397	9.5237
877	2.94300	1721	3.23578	0.30944	9.4906
802	2.90417	1646	3.21643	0.29009	9.4625
747	2.87332	1591	3.20167	0.27533	9.4398
647	2.81090	1491	3.17348	0.24714	9.3929
567	2.7536	1411	3.14953	0.22319	9.3487
547	2.7380	1391	3.14333	0.21699	9.3364
487	2.6875	1331	3.12418	0.19784	9.2963
422	2.6253	1266	3.10243	0.17609	9.2457
387	2.5877	1231	3.09026	0.16392	9.2146
362	2.5587	1206	3.08135	0.15501	9.1903
337	2.5276	1181	3.07225	0.14591	9.1641
322	2.5079	1166	3.06670	0.14036	9.1472

$\theta_w=844$ $\log \theta_w=2.92634$

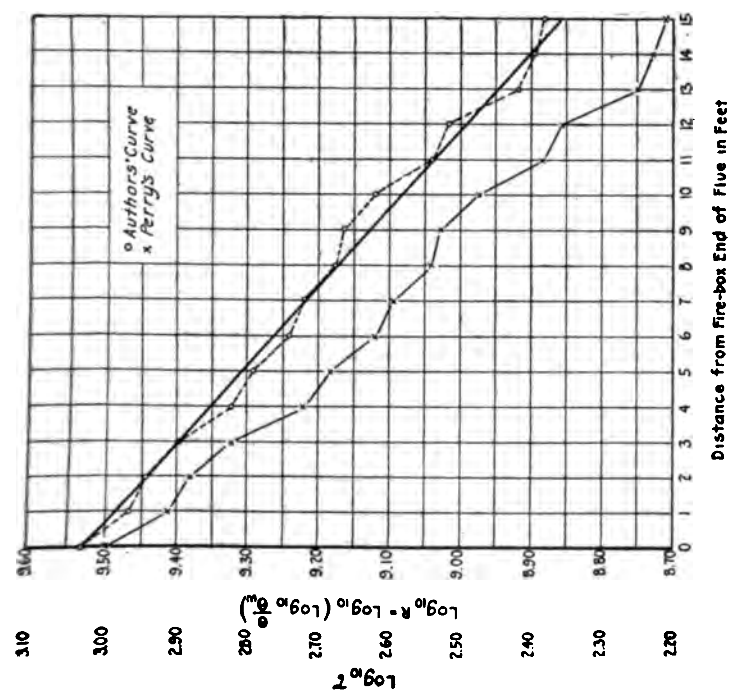

FIG. 3 PENNA. LOCOMOTIVE, TEST NO. 377

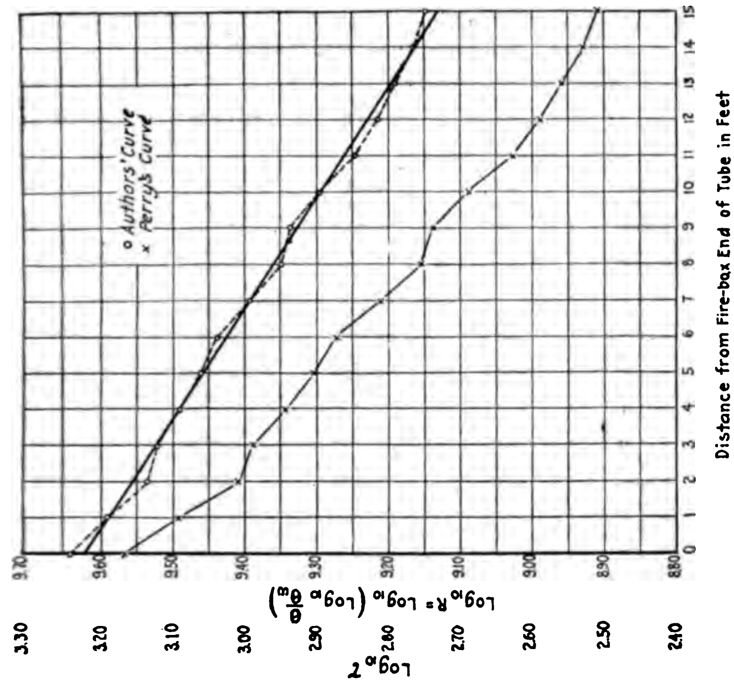

FIG. 4 PENNA. LOCOMOTIVE, TEST NO. 384

TABLE 5 COMPARISON OF PERRY'S, YOUNG'S AND AUTHORS' VALUES

Young's Values from Penna. R.R., Loco. No. 383

τ	$\log \tau$	$\theta=\tau+845$	$\log \theta$	$R=\log(\theta/\theta_w)$	$\Theta=\log R$
1316	3.1193	2161	3.33465	0.40679	9.6094
1096	3.0398	1941	3.28803	0.36117	9.5577
901	2.9547	1745	3.24204	0.31518	9.4986
781	2.8927	1626	3.21112	0.28426	9.4538
696	2.8426	1541	3.18780	0.26094	9.4165
596	2.7752	1441	3.15866	0.23180	9.3651
521	2.7168	1366	3.13545	0.20859	9.3193
466	2.6684	1311	3.11760	0.19074	9.2804
436	2.6395	1281	3.10755	0 17069	9.2322
396	2 5977	1241	3.09377	0.16691	9.2225
356	2.5514	1201	3.07954	0.15268	9.1838
291	2.4639	1166	3.06670	0.13984	9.1456
276	2.4409	1121	3.04961	0.12275	9.0890
256	2.4082	1101	3.04179	0.11493	9.0605
246	2.3909	1091	3.03782	0.11096	9.0452
221	2.3444	1066	3.02776	0.10090	9.0039

$\theta_w=845$ $\log \theta_w=2.92686$

TABLE 6 COMPARISON OF PERRY'S, YOUNG'S AND AUTHORS' VALUES

Young's Values from Penna. R.R., Loco. No. 378

τ	$\log \tau$	$\theta=\tau+846$	$\log \theta$	$R=\log(\theta/\theta_w)$	$\Theta=\log R$
1015	3.0064	1861	3.26975	0.34238	9.5345
975	2.9890	1821	3.26031	0.33294	9.5224
875	2.9520	1721	3.23578	0.30841	9.4891
795	2.9004	1641	3.21511	0.38774	9.4590
700	2.8451	1546	3.18921	0.26184	9.4180
595	2.7745	1441	2.15866	0.23129	9.3642
500	2.6990	1346	3.12905	0.20168	9.3047
495	2.6946	1341	3.12743	0.20006	9.3011
450	2.6532	1296	3.11261	0.18524	9.2676
395	2.5966	1241	3.09377	0.16640	9.2211
355	2.5502	1201	3.07954	0.15217	9.1823
335	2.5250	1181	3.07225	0.14488	9.1616
295	2.4698	1141	3.05729	0.12992	9.1137
255	2.4065	1101	3.04179	0.11442	9.0585
235	2.3711	1081	3.03383	0.10646	9.0271
215	2.3324	1061	3.02572	0.09835	8.9928

$\theta_w=846$ $\log \theta_w=2.92737$

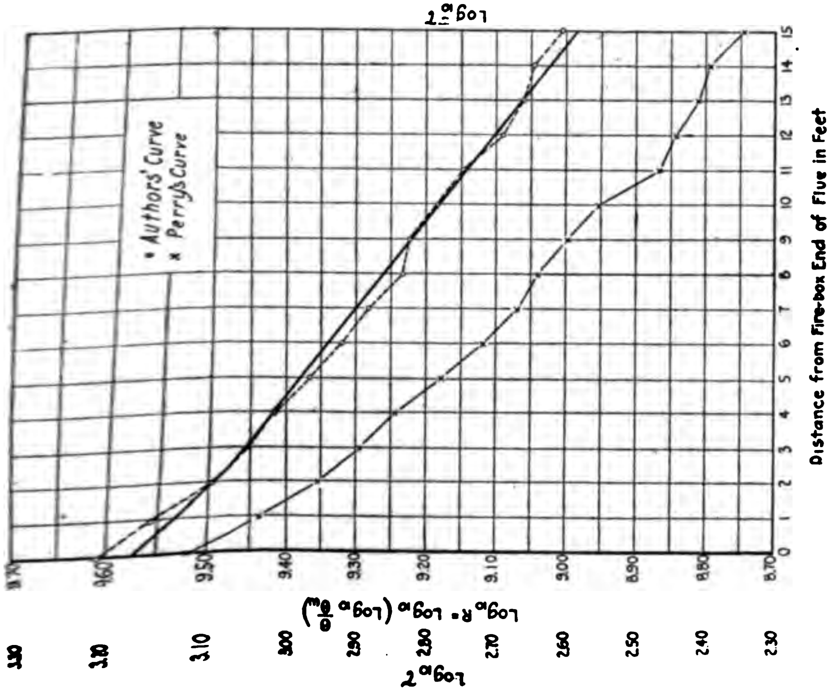

FIG. 5 PENNA. LOCOMOTIVE, TEST NO. 383

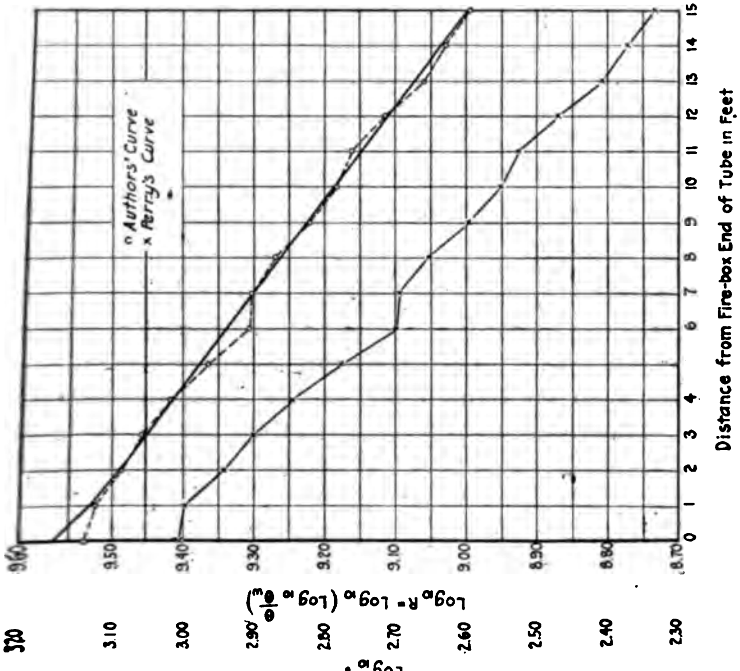

FIG. 6 PENNA. LOCOMOTIVE, TEST NO. 378

32 Formulæ [20] may be checked with the experiments jus mentioned in precisely the same manner as formula [16] was checke above with direct temperature measurements. In Tables 7, 8, 9, 10 11, five particular runs are given, and these same runs are plotte in Figs. 7, 8, 9, 10, 11. We have plotted in all about forty suc runs. The agreement is in all cases at least as good as that shown i these figures, these particular runs having been chosen purposel entirely at random.

33 The corresponding points for comparison with Perry's formula [3] are marked, as before, by small crosses. It will b noticed that there exists again a consistent tendency to be low in th middle. Thus, even if the first point (which may be high on accou of radiation from the furnace) and the last point be omitted, t points lie consistently below a line joining the new extreme poin Or again, if a line is laid through the first few points, so as to pred the values of the later ones according to [3], the actual points above it, and an analogous effect occurs if a line is passed throu several of the last points. An important feature is that these effe occur constantly in all the figures.

34 The coefficient M in equation [21] varies with the weight gas passing per minute through the tube. Figs. 7, 8, 9, 10, 11 sh the variation of M plainly, since M is the negative of the slope the figures are drawn. The relation between M and the weight gas per minute is shown graphically in Fig. 12, in which these quantities are plotted against each other from pairs of values fo in a series of nineteen experiments using a standard 2-in. boiler t with different furnace temperatures and gas velocities. Other periments not yet entirely completed in which smaller tubes are u seem to indicate that M also varies with the size of the tube. T data are not yet sufficiently complete to warrant a definite conclus as to the exact nature of this dependence, except that M is larg for small tubes than for large tubes.

35 If M were constant for all gas velocities, it would follow tha the boiler efficiency would be constant for all rates of driving. The decrease in the value of M is in accord with the observed practice, since it shows that the efficiency decreases as the boiler is forced.

36 It is easy to express the efficiency in general as a function of x as well as of M. For if the efficiency E of a tube of length x be defined to be the heat absorbed in that length of tube divided by the total heat content of the gases reckoned above water temperature we have

$$-E = \frac{H - H_w}{H_0 - H_w} = \frac{H_w}{H_0 - H_w}[(H_0/H_w)^{e^{-mz}} - 1] \ldots\ldots\ldots [23]$$

where

$$m = M \log_e 10 = 2.3026 \times M$$

TABLE 7

F No. 12, 1 Tube, Diam. 1.816 In., Furnace 1975 Deg. Fahr., Exit 456 Deg. Fahr., 0.677 Lb. Gas per Min., B.t.u. per Lb. at 212 Deg. Fahr.=164.41, H_0=457.31, H_w=109.71, H_0-H_w=347.60

$H-H_w$	$\log \frac{H-H_w}{H_0-H_w}$	H	$\log H$	$R=\log (H/H_w)$	$\Theta=\log R$
347.60	10.0000	457.31	2.66021	0.61998	9.79238
231.65	9.8238	341.36	2.53321	0.49298	9.69283
188.34	9.7338	298.05	2.47429	0.43406	9.63755
154.79	9.6486	264.50	2.42243	0.38220	9.58229
128.37	9.5674	238.08	2.37672	0.33649	9.53697
104.90	9.4797	214.61	2.33165	0.29142	9.46452
84.26	9.3845	193.97	2.28773	0.24750	9.39358
69.65	9.3018	179.36	2.25373	0.21350	9 32940
56.28	9.2091	166.99	2.22269	0.18246	9.26117
49.21	9.1510	158.92	2.20118	0.16095	9.20699
43.00	9.0924	152.71	2.18387	0.14364	9.15728

TABLE 8 COMPARISON OF PERRY'S AND FESSENDEN'S VALUES

F No. 15, 1 Tube, Diam. 1.816 In., Furnace 1505 Deg. Fahr., Exit 454 Deg. Fahr., 1.0063 Lb. Gas per Min., B.t.u. per Lb. at 212 Deg. Fahr.=163.79, H_0=530.06, H_w=162.84, H_0-H_w=367.22

$H-H_w$	$\log \frac{H-H_w}{H_0-H_w}$	H	$\log H$	$R=\log (H/H_w)$	$\Theta=\log R$
367.22	10.0000	530.06	2.72433	0.50731	9.70527
262.63	9.8544	425.47	2.62887	0.41185	9.61474
219.88	9.7773	382.72	2.58288	0.36586	9.56331
185.83	9.7042	348.67	2.54241	0.32539	9.51240
158.28	9.6345	321.12	2.50677	0.28965	9.46187
134.19	9.5629	297.03	2.47280	0.25578	9.40787
113.88	9.4915	276.72	2.39233	0.22502	9.35222
97.77	9.4252	260.61	2.41590	0.19897	9.29879
83.95	9.3591	246.79	2.39233	0.17531	9.24381
73.36	9.3006	236.20	2.37328	0.15626	9.19385
64.69	9.2460	227.53	2.35704	0.14002	9.14619

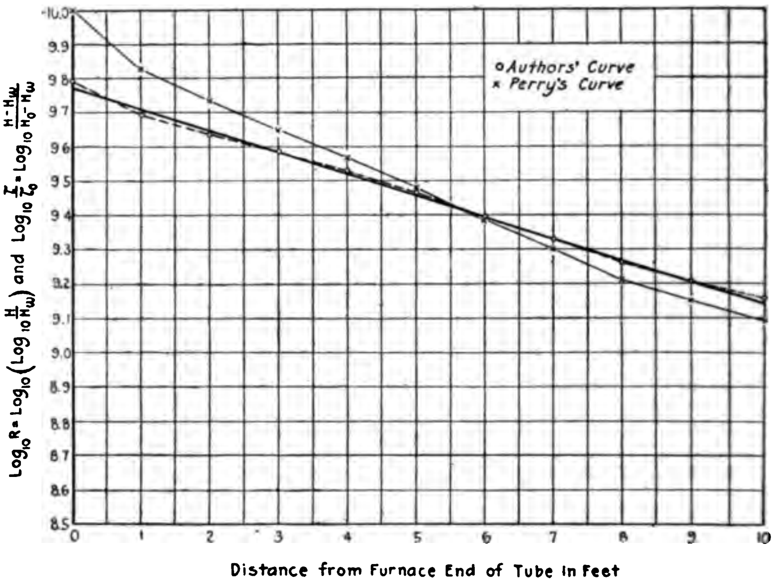

FIG. 7 FESSENDEN TEST No. 12

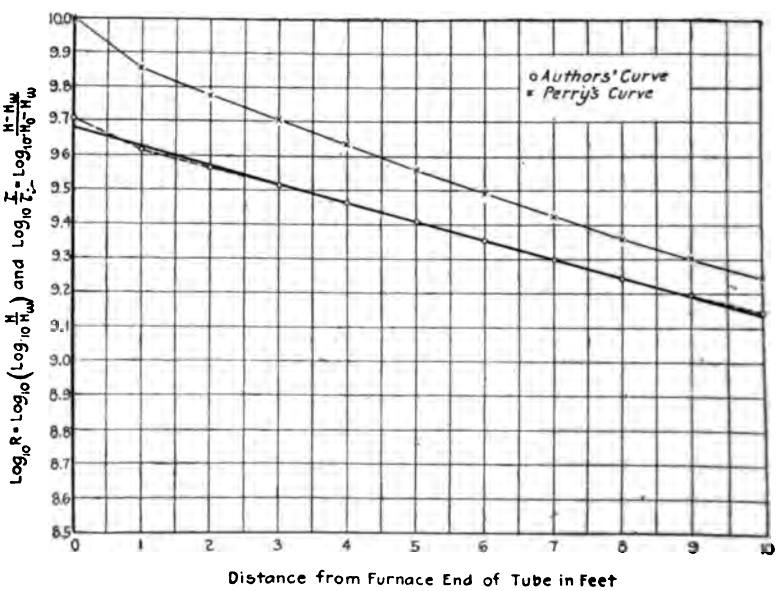

FIG. 8 FESSENDEN TEST No. 15

TABLE 9 COMPARISON OF PERRY'S AND FESSENDEN'S VALUES

F No. 114, 2 Tubes, Diam. 0.816 In., Furnace 1998 Deg. Fahr., Exit 264 Deg. Fahr., 0.523 Lb. Gas per Min., B.t.u. per Lb. at 212 Deg. Fahr.$=160.57$, $H_0=350.99$, $H_w=83.99$, $H_0-H_w=267$

$H-H_w$	$\log\dfrac{H-H_w}{H_0-H_w}$	H	$\log H$	$R=\log(H/H_w)$	$\Theta=\log R$
267.00	10.0000	350.99	2.54529	0.62104	9.79312
161.70	9.7822	345.69	2.39039	0.46614	9.66852
112.89	9.6261	196.88	2.29420	0.36995	9.56814
75.44	9.4510	159.43	2.20257	0.27832	9.44454
52.70	9.2953	136.69	2.13574	0.21149	9.32529
36.11	9.1310	120.10	2.07954	0.15529	9.19114
25.35	8.9773	109.34	2.03878	0.11453	9.05892
17.37	8.8136	101.36	2.00587	0.08162	8.91180
11.78	8.6444	95.77	1.98123	0.05698	8.75572
9.36	8.5453	93.35	1.97011	0.04586	8.66143
6.99	8.4183	90.98	1.95895	0.03470	8.54033

TABLE 10 COMPARISON OF PERRY'S AND FESSENDEN'S VALUES

F No. 112, 2 Tubes, Diam. 0.816 In., Furnace 1992 Deg. Fahr., Exit 291 Deg. Fahr., 0.733 Lb. Gas per Min., B.t.u. per Lb. at 212 Deg. Fahr.$=161.45$, $H_0=502.59$, $H_w=118.89$, $H_0-H_w=383.70$

$H-H_w$	$\log\dfrac{H-H_w}{H_0-H_w}$	H	$\log H$	$R=\log(H/H_w)$	$\Theta=\log R$
383.70	10.0000	502.59	2.70121	0.62606	9.79662
255.86	9.8340	374.75	2.57374	0.49859	9.69774
181.25	9.6743	300.14	2.47732	0.40217	9.60441
134.99	9.5463	353.88	2.40463	0.32948	9.51783
97.67	9.4057	216.56	2.33558	0.26043	9.41569
70.11	9.2617	189.00	2.27646	0.20131	9.30387
50.88	9.1225	169.77	2.22986	0.15471	9.18952
36.30	8.9759	155.19	2.19086	0.11571	9.06337
26.37	8.8376	145.28	2.16221	0.08706	8.93982
26.44	8.7267	139.33	2.14404	0.06889	8.83816
14.78	8.5854	133.67	2.12603	0.05088	8.70655

37 The rate of drop in temperature along the tube is expressible readily on carrying out the indicated differentiation in [10]; this gives

$$d\theta/dx = -m\theta\log_e(\theta/\theta_w)\ldots\ldots\ldots\ldots\ldots\ldots[24]$$

or, by [14]

$$d\theta/dx = -2.3026\, m\, R_0\, \theta\, e^{-mx}\ldots\ldots\ldots\ldots[25]$$

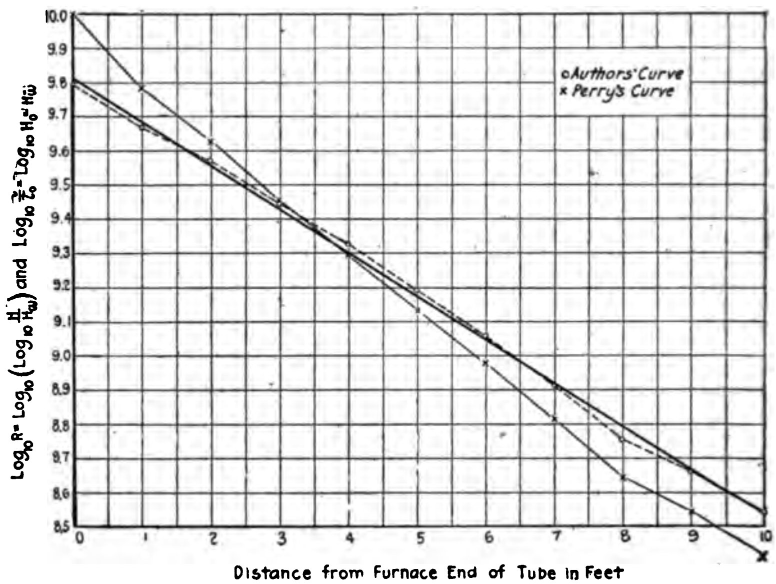

FIG. 9 FESSENDEN TEST NO. 114

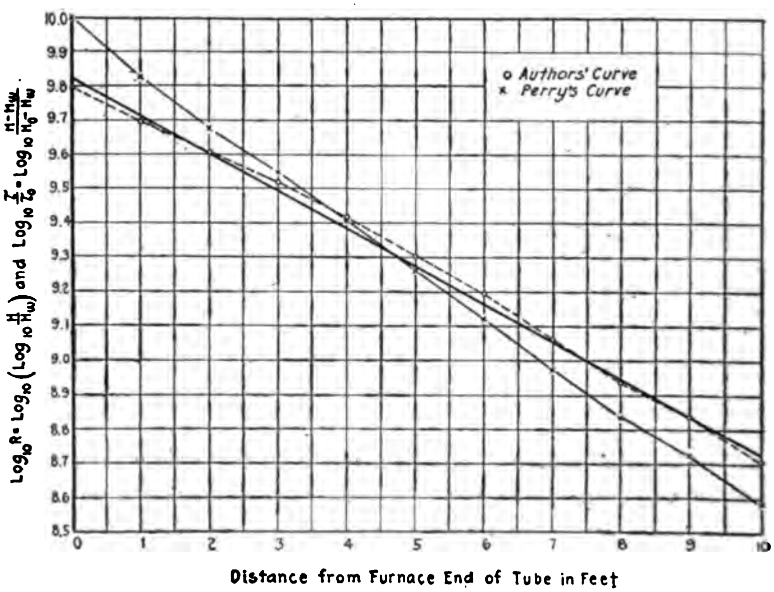

FIG. 10 FESSENDEN TEST NO. 112

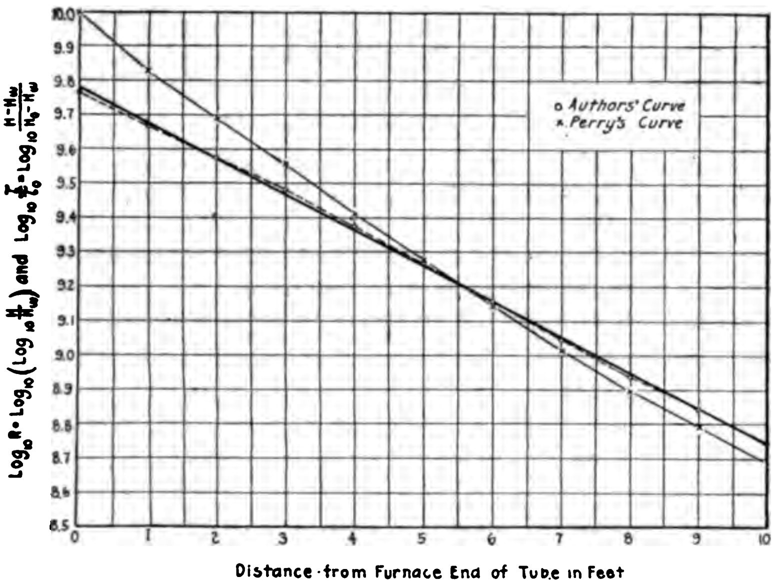

FIG. 11 FESSENDEN TEST No. 117

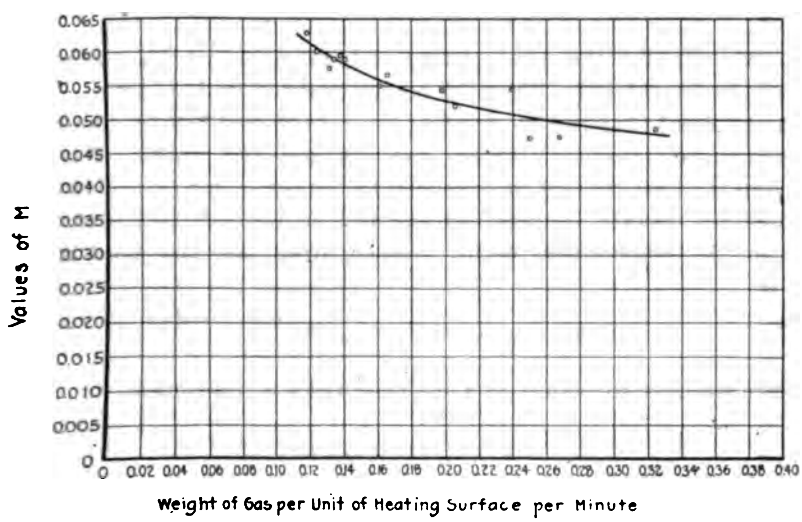

FIG. 12 VARIATION OF M

Since the volume of a given weight of gas is proportional to its absolute temperature θ, the weight of the amount of gas δQ in a section of length δx is inversely proportional to θ, and we have

$$\textit{loss of heat in } \delta Q = +\, ce^{-mx}$$

which agrees with [4], and shows the equivalence of the fundamental assumptions mentioned at the beginning.

TABLE 11 COMPARISON OF PERRY'S AND FESSENDEN'S VALUES

F No. 117, 2 Tubes, Diam., 0.816 In., Furnace 1787 Deg. Fahr., Exit 295 Deg. Fahr.
0.844 Lb. Gas per Min., B.t.u. per Lb. at 212 Deg. Fahr. = 162.04
$H_0 = 521.46$, $H_w = 136.79$, $H_0 - H_w = 384.67$

$H-H_w$	$\log \frac{H-H_w}{H_0-H_w}$	H	$\log H$	$R = \log (H/H_w)$	$\Theta = \log R$
384.67	10.0000	521.46	2.71722	0.58118	9.76431
259.07	9.8283	395.86	2.59754	0.46150	9.66417
187.66	9.6883	324.45	2.51115	0.37511	9.57416
137.63	9.5536	274.42	2.43842	0.30238	9.48055
98.74	9.4096	235.53	2.37205	0.23601	9.37293
72.40	9.2746	209.19	2.32045	0.18450	9.26600
53.59	9.1440	190.38	2.27962	0.24358	9.15709
39.73	9.0141	176.52	2.24679	0.11075	9.04434
30.25	8.8954	167.04	2.22282	0.08678	8.93842
23.96	8.7945	160.75	2.20615	0.07011	8.84578
18.75	8.6875	155.54	2.19183	0 05579	8.74656

38 Finally, the conductivity and its reciprocal, the resistance, may be formulated. Thus the conductivity is usually defined to be the quotient obtained by dividing the rate of heat loss by the difference in temperature between the gas and the water. We have taken the heat content proportional to the temperature, and we may therefore operate through [16] as well as through [20] since only ratios of temperatures occur. Hence, from [24] the conductivity γ is

$$\gamma = c_p m \frac{\theta}{\theta - \theta_w} \log_e \theta/\theta_w = c_p m \theta \, \frac{\log \theta - \log \theta_w}{\theta - \theta_w} \quad \ldots\ldots\ldots [26]$$

Since we have for values of θ less than twice θ_w the series

$$\log \theta - \log \theta_w = \frac{1}{\theta_w}(\theta - \theta_w) - \frac{1}{2\,\theta_w^2}(\theta - \theta_w)^2 + \ldots$$

we have also

$$\gamma = c_p m \frac{\theta}{\theta_w}\left[1 - \frac{1}{2\,\theta_w}(\theta - \theta_w) + \ldots\right]$$

whence it follows that γ approaches $c_p m$ as a limit θ approaches θ_w. The constant m which occurs throughout the preceding work may therefore be thought of as proportional to the limiting value of the conductivity as the temperature of the gases approaches that of the water. Thus the fact that the value of M, which is proportional to m, is smaller for a larger pipe, may be interpreted to mean that the gases in the larger tube have less contact with the tube wall per unit of volume in the larger tube; this would lead us to expect a smaller limiting conductivity in the larger tube and therefore also a smaller conductivity in general. This is precisely the fact mentioned above.

ADDENDUM

39 Through the kindness of Dr. D. S. Jacobus, the authors were furnished with an advance manuscript copy of the forthcoming bulletin of the Babcock and Wilcox Company, in which a set of tests conducted by that company in 1915 is described in detail. The officers of the company generously placed the complete data of one of these tests (No. 3) at our disposal for the purpose of testing still further the fundamental assumptions and the theory of this paper. Since these tests were not known to us when the foregoing parts of this paper were presented, and since these tests were carried out with great care and accuracy, the check afforded on our theory seems very important, and we wish to acknowledge our indebtedness to the Babcock and Wilcox Company for this weighty addition to the evidence in support of the theory.

40 The experiments were made on a set of twenty sections arranged along a central fire-tube, through which passed the hot gases from a gas furnace. The sections simulated small portions of a boiler along a fire-tube. Thus far, the experiments resemble very closely those of Professor Fessenden.

41 A noteworthy difference between these experiments, which somewhat affects the present discussion, is that the amount of heat transferred to the water was measured in the Babcock and Wilcox experiments by the increase in temperature of water allowed to run through each section without boiling. This resulted in somewhat different water temperatures in the various sections. In the Fessenden experiments, the water in each section boiled at atmospheric pressure, so that the temperature of the water in all sections was sensibly the same.

42 It is evident that a difference in temperature in the water at different points along the pipe would materially affect the theory and would disturb any attempted check at least slightly. The effect would be to show a greater drop in heat content of the gases near the cool end of the pipe where the water is slightly cooler in the Babcock and Wilcox tests than would occur according to the theory. Another disturbing element is that the state of the water very near the tube is quite different when boiling is allowed to occur as in a boiler;

when it is not, the total resistance is probably different, and this difference is probably greatest at the hot end of the tube.

43 Finally, the radiation correction is taken at constant value for all the sections, as done in the Babcock and Wilcox paper. • As a matter of fact, since the water is slightly hotter near the hot end

TABLE 12 BABCOCK AND WILCOX TEST 3

TWENTY SECTIONS, EXIT 388 DEG. FAHR., 1.9117 LB. GAS PER MIN., B.T.U. PER MIN. AT AVE. WATER TEMP. (150.69 DEG. FAHR.) IS H_w, $H_w = 281.50$. $H_{\text{furnace}} = H_{x=0} = 1435.41$. RADIATION PER SECTION 0.5 B.T.U. PER MIN

x	$\tau = T - t$	B.t.u. Loss/hr	Corrected Total H per min.	$R = \log_{10} \dfrac{H}{H_w}$	$\log_{10} R$	Differences
0		-	1435.41	0.70749	9.84971	
1	1877	9836.82	1271.13	0.65471	9.81605	0.03366
2	1667	7091.57	1152.44	0.61214	9.78685	0.02920
3	1481	5885.29	1053.85	0.57330	9.75838	0.02849
4	1318	5548.82	960.98	0.53319	9.72688	0.03150
5	1174	4832.14	879.83	0.49492	9.69454	0.03234
6	1047	4029.82	812.17	0.46016	9.66291	0.03163
7	934	3517.51	753.04	0.42734	9.63077	0.03224
8	835	3196.24	699.27	0.39516	9.59678	0.03399
9	747	2654.40	654.53	0.36645	9.56402	0.03276
10	669	2359.90	614.70	0.33918	9.53043	0.03359 •
11	601	2946.38	580.09	0.31408	9.49704	0.03339
12	540	1773.83	550.03	0.29090	9.46374	0.03330
13	485	1610.26	522.69	0.26876	9.42937	0.03347
14	437	1412.21	498.65	0.24832	9.39501	0.03436
15	395	1246.88	477.37	0.22937	9.36053	0.03548
16	356	1084.78	458.79	0.21213	9.32660	0 03393
17	322	979.99	441.97	0.19591	9.29205	0.03455
18	292	870.36	426.96	0.18091	9.25746	0.03459
19	264	782.32	423.42	0.16691	9.22249	0.03497
20	237	772.97	400.04	0.15262	9.18361	0.03888

This table shows the check with the formulæ of this paper by the reasonable constancy of the column of differences. The mean difference is 0.03331. The last section is comparatively unreliable for all purposes, and the corresponding difference is high. See also graphical figure, Fig. 13.

of the tube, the radiation correction probably should be slightly greater for the first sections. This would cause the data to show values rather less than those indicated by the theory near the hot end of the tube. This would be true even for small differences in the radiation, because the effect is cumulative, since the temperatures and heat content values are computed from the cooler end of the tube back toward the furnace, by successive additions.

44 All the disturbances are small compared with the totals, and the check should be reasonably close in spite of them. In resumé, the data should give points slightly low on the hot end, on account of radiation corrections, and also slightly low on the cold end, on account of the relatively cool water surrounding the pipe there.

TABLE 13 BABCOCK AND WILCOX TEST 3

COMPUTATIONS FOR CHECKS WITH FORMULÆ OF PERRY AND KENT. ORIGINAL DATA AS GIVEN IN TABLE 12

Section	$\tau = T - t$	$\log_{10} \tau$	Diff. log τ	Mean of $\tau = T - t$ over Section	Log Mean τ	L Heat Loss per Hour (Corrected)	Log L
1	1877	3.27346					
2	1667	3.22194	0.05152	1772	3.24846	7140	3.85370
3	1481	3.17056	0.05138	1574	3.19700	6181	3.79106
4	1318	3.11992	0.05064	1400	3.14613	5360	3.72916
5	1174	3.06967	0.05025	1246	3.09552	4655	3.66792
6	1047	3.01995	0.04962	1111	3.04571	4051	3.60756
7	934	2.97035	0.04960	991	2.99607	3530	3.54777
8	835	2.92169	0.04866	885	2.94694	3082	3.48883
9	747	2.87332	0.04837	791	2.89819	2696	3.43072
10	669	2.82543	0.04789	708	2.85003	2361	3.37310
11	601	2.77887	0.04656	635	2.80277	2072	3.31634
12	540	2.73239	0.04648	571	2.75664	1822	3.26055
13	485	2.68574	0.04665	513	2.71012	1604	3.20520
14	437	2.64048	0.04526	461	2.66370	1415	3.15076
15	395	2.59660	0.04388	416	2.61909	1250	3.09691
16	356	2.55145	0.04515	376	2.57519	1106	3.04376
17	322	2.50786	0.04359	339	2.53020	981	2.99167
18	292	2.46538	0.04248	309	2.48714	871	2.94002
19	264	2.42160	0.04378	278	2.44404	775	2.88930
20	237	2.37475	0.04685	251	2.39967		

T = temperature of gas at end of section named. t = temperature of water.

If Perry's formula is correct, the column "Diff. log τ" should be constant and the figure should be a straight line (see Fig. 13). The discrepancies are considerable and are consistent.

If Kent's hypothesis is correct, the plot of Log (mean τ) against Log L should give a straight line of slope 2 (see Fig. 14). The best line through the points located shows a slope approximately 1.2.

45 The values for the heat absorbed in each of the 20 sections were taken directly from the Babcock and Wilcox table for test No. 3, with their determination of the radiation correction added, and this was then reduced to B.t.u. per minute. The value of the total heat content H at any temperature above the heat content reckoned at zero for temperature absolute zero was computed from the gas analysis

they give, together with the usual formulæ for specific heats. We find for H at 32 deg. fahr. the value 117.55 B.t.u. per minute, on the basis of 1.9117 lb. of gas per minute. At the exit temperature (388 deg. fahr.) we find H_{388} = 400.04 B.t.u. per min. Adding to this the loss to the water in each section gives the value of H at the front end of each section; these are the values tabulated in the column headed H. The water temperature was not constant, but the variation was only

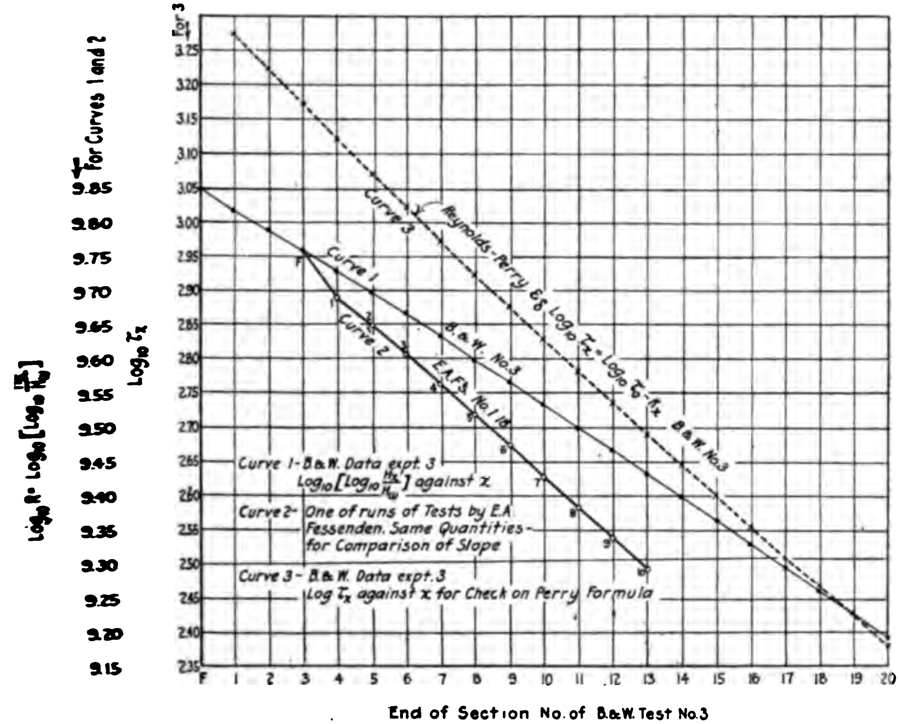

FIG. 13 BABCOCK & WILCOX EXPERIMENT No. 3 CHECK AGAINST PERRY'S AND AUTHORS' FORMULÆ

about 6 deg. An average value was taken as 150.69 deg. fahr. and H at this temperature was computed to be H_w = 281.50 B.t.u. per minute.

46 With these values, substitution is possible in the previous formulæ. The data are plotted in Fig. 13, on the same plan as before. The points computed for a check on our theory are again marked by small circles, and a broken line connecting them is drawn in full lines. It will be seen at a glance that the check, which again

consists in the fact that these points should lie on a straight line, is very good. There appears to be a very slight lowering of the points on both ends, but this is indeed small, and is apparently not more than would have been expected from the remarks made above.

47 We have shown on the same figure the attempt to check the data with the Perry formula mentioned above. (See p. 6, formula [3].) It will be seen that these points vary more distinctly

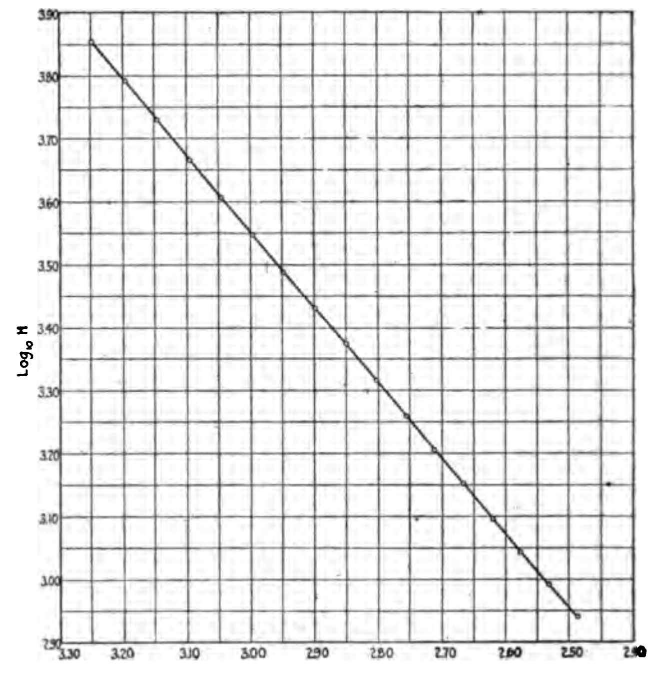

FIG. 14 BABCOCK & WILCOX EXPERIMENT NO. 3 CHECK AGAINST RANKINE'S AND KENT'S FORMULÆ

from a straight line, and in a consistent manner similar to the behavior of such points in other sets of data given in this paper.

48 The formulæ by several other writers agree substantially with those of Perry. Thus, the formulæ given in Bulletin No. 18 of the Bureau of Mines are substantially the same as Perry's formulæ in so far as the variables dealt with in this paper are concerned.

49 We have attempted to check with the formulæ given by Jordan, but no satisfactory method suggested itself. From what computation seemed possible from the data given and from Jordan's

formulæ, we are not inclined to believe that these formulæ will fit the data very accurately.

50 Finally, we have checked through the assumption by William Kent (Steam Boiler Economy) that the rate of transmission is proportional to the square of the temperature difference. We have plotted the temperature differences as shown by Babcock and Wilcox against the transmission rate shown for the twenty sections. If the Kent law were correct, it is obvious that the logarithms of the temperature differences plotted against the logarithms of the transmission rates should give a straight line whose slope is 2. Fig. 14 shows indeed a reasonably good straight line, but the slope, instead of being 2 is about 1.2. Thus, the formula, instead of being of the form

$$q = \frac{(T - t)^2}{a},$$

where a is between 160–200, as Kent suggests, should be rather approximately of the form

$$q = \frac{(T - t)^{1.2}}{a},$$

where a = (about) 400, if the Babcock and Wilcox tests are to be satisfied. It is to be remarked that the q used here is the rate of transmission per unit surface, but not per degree difference in temperature. The rates mentioned in the Babcock and Wilcox paper are rates per unit surface per degree difference in temperature. The formulæ of Rankine are substantially the same as those of Kent.

51 We feel therefore that these tests agree substantially with the theory presented in the present paper, and that the same data do not agree to a like extent with any other formulæ known to us.

DISCUSSION

Forrest E. Cardullo presented a written discussion in which he said that equation [4] of the paper, although appearing to be rational, could not represent any actual physical law, as the quantity of gas considered was continually diminishing according to some law which was a function of x; and that the assumption that decrease in entropy in a small section of a tube of length δx is proportional to the entropy difference $\phi - \phi_w$ also appeared to be rational until, upon investigation, there was found to be no reason for believing that the entropy, or rate of change of entropy, had anything to do with heat transfer, the fact of its leading to a correct conclusion being a mere coincidence.

Since the velocity of gases flowing through a constant cross-section will be greater at high temperatures, because of larger specific volume of gas, equations derived from assuming the rate of heat transfer proportional only to temperature difference will not represent the facts unless modified to take into account the well-known fact of increased conductivity with increased velocity. Thus the empirical equation happily chosen by the authors, $Q = KT \log T/t$, where Q is the rate of heat transfer in B.t.u. per sq. ft. per hr., T the absolute temperature of the gas at the given point, t that of the water, and K an experimentally determined constant depending, among other things, upon the weight of gases flowing through the cross-section in a given time, leads to astonishingly accurate results, while the usual assumption, that $Q = K(T - t)$, gives systematic, though not serious, inaccuracies.

The equation from Perry is manifestly untrue unless we assume that the air supplied to the fire is at the same temperature as the water in the boiler.

William Kent in a written discussion said the results of boiler tests given in the authors' paper and accredited to him were the averages of four tests made by C. D. Young and also given in the authors' paper. Professor Kent made a comparison of figures obtained by averaging Mr. Young's tests with the average of Professor Fessenden's five tests, the results of the Babcock and Wilcox test and the results of computations of two hypothetical cases, based on Rankine's formula, which differ only in the amount of gases ($f = 20$ and $f = 30$) per pound of fuel. Rankine's formula is based on the assumption that the heat transmission is proportional to the square of the temperature difference, and it appears to agree with the results of recent boiler tests far more closely than do the formulæ of Reynolds and Perry. (See Steam Boiler Economy, 2nd edition, p. 310.) Taking the difference in temperatures of two successive sections as a measure of the loss of heat by the gas and of the heat transmitted to the water, and the mean difference of the temperature between the gas and the water as the average of the temperatures at the beginning and at the end of a section, minus the constant temperature of the water, a table was constructed showing the relation of the heat loss to the mean difference between the temperatures f the gas and water. These figures are plotted on the diagram hown in Fig. 15.

It is interesting to note that the hypothetical curve for $f = 30$ grees almost exactly with the four locomotive-tube tests between 100 and 700 deg. temperature difference, but below the lower tem-erature the curve merges with the curve of the Babcock and Vilcox tube. The curve of the Fessenden tube shows a steeper nclination between $(T - t) = 297$ and 157 than it does at the

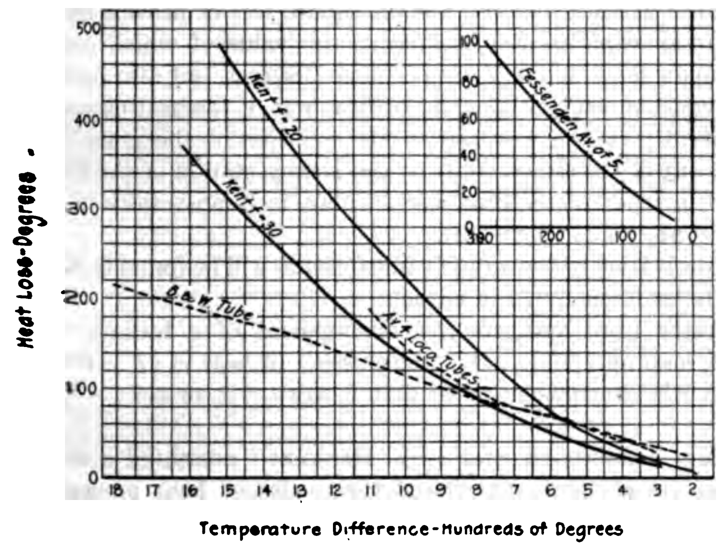

FIG. 15 RELATION OF HEAT LOSS TO MEAN DIFFERENCE BETWEEN TEMPERATURES OF GAS AND WATER

lower temperature range, from $(T - t) = 157$ to 33, while the curve of the Babcock and Wilcox tube shows a considerably flatter inclination at the higher than at the lower temperatures. All the curves, except that of the Babcock and Wilcox tube, show a distinct tendency to becoming convex to the axis of the abscissæ. The Babcock and Wilcox tube curve cannot be made to fit the equation of any curve unless two of its points, $(T - t) = 1350$ and 1246, are shifted downwards, when it becomes a straight line, indicating that the transmission of heat is proportional to the difference in temperature, which is contrary to all the other curves. The curves of the Fessenden tests coincide nearly with two straight lines of differing inclination. From $(T - t) = 297$ down to 157, the loss of heat is $0.37\,(T - t)$, while from $(T - t) = 157$ down to 33, the loss is $0.29\,(T - t)$.

The result of over thirty years' study has led Professor Kent to the conclusion that for boilers driven at a rate above 3 lb. evaporation per sq. ft. of heating surface per hr., at which the radiation loss is small, the formula for boiler efficiency is that of a straight line, $E = A - B(W/S - 3)$ in which A is the maximum efficiency at the 3-lb. rate of driving, W is the pounds of water evaporated from and at 212 deg. per hr., S is the square feet of heating surface, and B is an experimental coefficient, the value of which depends chiefly upon the amount of air per pound of fuel, but also upon the completeness of combustion and the cleanness of the heating surface. It apparently has no relation to the velocity of the gases. The ten best out of the sixteen tests at the Delray station of the Detroit Edison Co. reported by Professor Jacobus in Transactions in 1911 gave $E = 81 - 1.33\,(W/5 - 3)$.

The derivation of the straight-line formula will be found in Kent's Steam Boiler Economy, 2nd edition.

Professor Kent suggests the development of a formula based upon the assumption that the transmission of heat is $(T - t)^x \div a$, in which x is an exponent other than 2, say 1, 1.2, or 1.5.

Henry Kreisinger and J. F. Barkley [1] presented a written discussion in which they reviewed the modes of heat propagation discussed by them in various publications.[2]

To show that no formula can be based on the temperature measurements furnished by Professor Kent and C. D. Young, Messrs. Kreisinger and Barkley reviewed the reasons for inaccuracies.

The original equation of Reynolds,[3] $H = A\,(T - t) + B(T - t)qv$, not touched upon by the authors, applies to the convection part of the heat transmission in Fig. 16, stopping at the dry surface and not extending through the plate to the water; that is, t is the temperature of the dry surface. Perry distinctly states that t is the temperature of the film of gas next to the metal. Further in his discussion,[4] he considers the change in conductivity of the gas film and its effect upon t, obtaining an expression which includes the prob-

[1] Junior Electrical Engineer, U. S. Bureau of Mines.
[2] Bulletin 18, and Technical Paper 114, U. S. Bureau of Mines.
[3] Bulletin 18, Bureau of Mines, p. 114.
[4] Perry, The Steam Engine, 1909, p. 588.

able thickness and conductivity of the film as the velocity of the gases and their temperatures vary. He concludes that in existing boilers the resistance of the metal itself is insignificant, but as better circulation is provided on both sides of the metal the total resistance must approximate more and more that of the metal itself. A footnote adds, "The above investigation shows that the following simple way of putting the whole matter is legitimate within certain limits of velocity, etc."

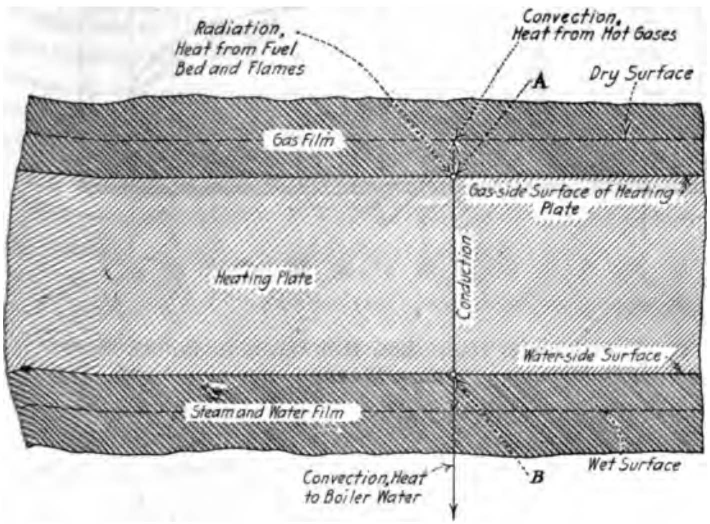

Fig. 16 Path of Heat Travel from Hot Gases Through Heating Plate Into Boiler Water

Perry then develops the simple form of equation used by the authors, $\theta = \theta_1 e^{-cx}$, in which θ is the temperature at any point along the path of the gases when the initial temperature θ_1 is given, these temperatures being measured above the boiler water. In Fig. 13 the points obtained with the simple Perry expression are about as much one way from a straight line as the points derived from the authors' formula are the other way.

Fig. 17 shows that the largest temperature drop is in the layer of gas about ⅛ in. thick next to the wall of the tube, and only a very small drop from the gas-side surface of the tube through the metal and through the water film to the boiler water. The problem is to get the heat from the gas into the metal of the heating plate.

Perry's simple formula gives an expression for true boiler efficiency which is independent of the initial temperature. How nearly this is true when a boiler receives heat only by convection through its flues is shown in Fig. 18, which gives the results of some experi-

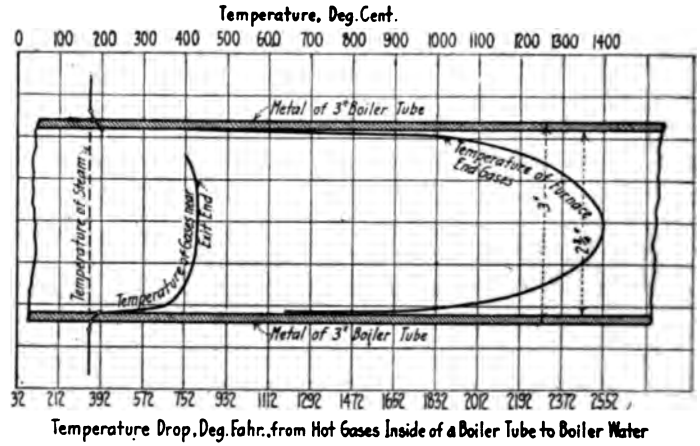

FIG. 17 TEMPERATURE DROP FROM HOT GASES IN BOILER TUBE TO BOILER WATER

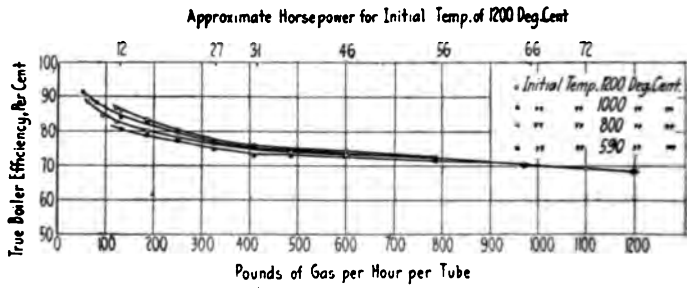

FIG. 18 TRUTH OF PERRY'S FORMULA WHEN BOILER RECEIVES HEAT ONLY BY CONVECTION

ments. When a small weight of gas flows through the tube and the velocity is low, the efficiencies for 1200 deg. cent. and 600 deg. initial temperature are 4 or 5 per cent apart. As more gases are pushed through the tube at a higher velocity, the scrubbing effect increases, and the efficiency curves for the different temperatures come together. There seems to be considerable range, both as to temperature and weight of gas, for which the formula can be applied with but little error.

JOHN E. BELL, in a written discussion, pointed out that the constant m in the authors' equation [24] has the dimension L^{-1} and is independent of θ, but depends on the physical characteristics of the gas, the velocity of flow, and the dimensions of the channel.

Introducing the entropy of the gas, particularly as a mean value, seems unwarranted, and certainly equation [8] is incorrect. The left-hand side of this equation is the variation in the mean entropy of gas passing any section of the tube for a slight variation in the location of this section along the tube. This mean entropy must be measured in the same way as the mean temperature, which is found by averaging over the whole section, taking into account both the variation in temperature and the variation in velocity of flow. The mean entropy would be figured in this way, and equation [8] would not hold under these conditions.

If R represents the rate of heat transfer under different conditions, W the weight of the gas flow, and S the surface per unit length of channel, then, otherwise using the notation of the authors' paper,

$$\frac{d\theta}{dx} = \frac{-RS}{c_pW}(\theta - \theta_w)$$

Comparing this equation with equation [24] it is seen that

$$R = \frac{mc_pW}{S} \cdot \frac{\theta}{\theta - \theta_w} \log_e \theta/\theta_w$$

which is equation [26] of the paper with a little different notation. If this equation be correct, the transfer rate is the product of two factors, one mc_pW/S, independent of the temperatures and dependent on the weight flow and the dimensions of the channel, the other $\frac{\theta}{\theta - \theta_w} \log_e \theta/\theta_w$, independent of the weight flow and channel dimensions and dependent only on the temperature of the gas.

Experimental data published by the Babcock and Wilcox Company and dealt with in the authors' addendum show that

$$R = a + bw$$

where a is a constant independent of the temperature and gas flow, b is a coefficient that varies with the temperature, and w is the weight of gas per sq. ft. of flue area per hour. This cannot be of the form above unless the constant a has a zero value. Reynolds, Perry and other writers who have followed them have assumed that such is the case and have been led into conclusions which are not checked by facts. The experiments of the Babcock and Wilcox Company give to a the value 2.1, and the experiments of Jordan assign even a greater value. Since the value of this constant is unquestionably independent of the temperature, it appears that the law proposed in the paper can be only approximately true. It does not seem that Professor Fessenden's experiments were worked up to a point where the comparison made above could be satisfactorily carried out.

LAWFORD H. FRY (written). The authors offer an interesting type of empiric formula to represent the transmission of heat along a boiler flue, and the examples given of its application in specific intances show that in many cases constants can be found which make the formula fit actual conditions very closely. It will be found, however, that the *constants* are constant only for each particular case, and in its present form the formula can only be applied to give a smooth curve for existing data, and no means are offered by which the constants can be determined for varying conditions. There is a suggestion that progress in this direction will be made when Professor Fessenden's complete experiments are presented to the Society. Until this is done the formula will have little practical value. At present, if the gas temperature is measured at the inlet to a boiler flue and at one other point along the flue, the authors' formula enables us to determine the fall of temperature along the flue under the conditions at which the measurements were made, but does not enable us to predict the heat transmission or temperature drop in the flue under other conditions as to rate of flow of gas or initial temperature.

The authors apply their formula to the fall of temperature as measured in the tubes of locomotive boilers at the Pennsylvania Railroad Locomotive Testing Plant. In doing this it should be borne in mind that the temperatures measured in the flue in this way are probably affected by radiation to the walls of the flue and are undoubtedly lower than the true gas temperatures.

If a general formula for heat transfer is to be established it must take into account the nature of the gas flowing in the flue, the rate of flow, the dimensions of the flue, and the variation (if any) of the temperature along the wall of the flue. In this connection it would seem desirable for the authors in the continuation of their paper to give a somewhat wider bibliography and to consider some of the work which has been done in the direction of establishing a general formula, for example that by Nusselt[1] and by Leprince-Ringuet.[2] It would also appear to be necessary to take into account the variation of the specific heat with the temperature. The authors assume that the specific heat at constant pressure is constant, saying that the known facts do not seem to justify any other definite statement. It is, to say the least, surprising to compare this with the very carefully compiled equations for specific heat given in the Babcock and Wilcox Company's pamphlet to which the authors refer. These equations are as follows, C being the instantaneous specific heat and t the temperature in deg. fahr:

For CO_2 $C = 0.1983 + 8.35 \times 10^{-5} t - 1.67 \times 10^{-8} t^2$

" O_2 $C = 0.2154 + 0.000019\, t$

" N_2 $C = 0.2343 + 0.000021\, t$

" H_2O $C = 0.4541 + 3.2 \times 10^{-6} t + 2.825 \times 10^{-8} t^2$

For temperatures of 400 and 2000 deg. fahr., which represent approximately the range of temperatures in a boiler flue, these formulæ give the following values for the specific heats of the gases in question:

	400 Deg. Fahr.	2000 Deg. Fahr.
Carbon dioxide	0.2290	0.2985
Oxygen	0.2230	0.2534
Nitrogen	0.2427	0.2763
Water vapor	0.4599	0.5735

If these values be applied to a smokebox gas having the following analysis (taken from B. & W. pamphlet, p. 52):

	Volume	Weight
	Per cent	Per cent
Carbon dioxide	11.1	14.9
Oxygen	6.9	6.8
Nitrogen (difference)	82.0	70.4
Water vapor	....	7.9
	100.0	100.0

[1] Zeits. des Ver. deut. Ing., July 9, 1910.
[2] La Revue de Mécanique, Nov. 1911.

the instantaneous specific heat is found to be

0.2568 at 400 deg. fahr.
0.3019 at 2000 deg. fahr.

In other words, the variation in the specific heat of the gases in passing from one end of a boiler flue to the other is far from negligible.

THE AUTHORS. We are pleased to note that most of those who comment upon our paper agree that our formulæ represent experimentally observed facts better than other proposed formulæ. We regret that Messrs. Kreisinger and Barkley fail to appreciate this close agreement.

The original Reynolds-Perry equation used by the Bureau of Mines, and quoted by Messrs. Kreisinger and Barkley,

$$H = A(T - t) + B(T - t)qv \qquad [27a]$$

we prefer to write in the form

$$H = (A + Bqv)\,(T - t) \qquad [27b]$$

In this form, q is the density and v the velocity of the gas. Then qv is proportional to the weight of gases passing, hence

$$H = (A + Bw)\,(T - t) \qquad [27]$$

In any single experiment the weight of gases passing must obviously be the same for all parts of the tube, hence, for any single experiment the equations above reduce precisely to a statement that the heat transmitted is proportional to the temperature difference, and lead directly to what we have called Perry's formula. Hence the statement that this formula was not touched upon in our paper is only superficially true. It should be obvious that a test of what we call Perry's formula should also be a test of this one.

Possibly formula [27] might be made to fit experiments if the quantities A and B were admitted to be variables and not constants.[1] Our own tests, as well as those of the Babcock and Wilcox Company, indicate clearly that B is itself a function of the temperature difference, which varies along the tube, and possibly also of the physical properties of the gases. These experiments also indicate that A probably varies with $(T - t)$.

Mr. Bell, quoting from the bulletin describing the Babcock and Wilcox experiments, uses the formula

$$R = a + bw \qquad [28]$$

[1] It is to be noted that no values for A and B are given in the Bureau of Mines bulletin where formula [27a] is strongly emphasised.

to express the heat, transfer rate. This is identical with equation [27] above, except that b is a variable depending upon the temperature difference instead of a constant. It should be noted that the Babcock and Wilcox bulletin (p. 66) calls attention to the fact that the straight lines shown on the chart referred to are in reality secants or tangents to curved lines which really represent the facts. The same bulletin also shows that a varies with the weight; and we have some evidence, though not entirely conclusive, that a is also a function of the temperature difference. It would thus appear that equation [28] should be

$$R = u + vw$$

where u and v are functions of the temperature difference at least, and probably also of other conditions. It then seems that equation [28] may in reality be simply the first terms of a series which might well be expressed by the logarithmic form we have used.

For Mr. Fry's information it may be stated that in all of our own experiments the specific heat of the gases was carefully taken into account.

It should be noted that in making his calculations upon the Fessenden experiments, Professor Kent has used the heat content of the gases above the water temperature instead of the temperature differences used in compiling other data, thus displacing the curve for these experiments far to the right of its proper place. We object seriously to the averaging of several tests; and wish to state that we did not know that the data so kindly furnished us by him were made up of averages from a number of Mr. Young's tests.

Professor Kent's contention that the boiler efficiency, and therefore the heat-transfer rate, has no relation to gas velocity is manifestly incorrect. This has been shown by many experiments; and furthermore, Professor Kent's own formulæ apparently take some account of gas velocity in that they employ the gas weight.

We do not understand Professor Cardullo's use of the term "rational" as opposed to "empirical." We contend that some assumption underlies every theory. We cannot conceive how any assumption can be called rational which does not check with experiment. The statement that the rate of heat transfer is proportional to the temperature difference is, in reality, an assumption; and as it is not borne out by experiment, why should it be styled "rational"?

No. 1542

MEETINGS SEPTEMBER - DECEMBER

MILWAUKEE, SEPTEMBER 13

Lecture: Growing and Gathering of Rubber Latex, L. J. D. Healey, chief chemist, Federal Rubber Company. Followed by an inspection trip through the plant of the Federal Rubber Company.

MINNESOTA, SEPTEMBER 14

Held at Minneapolis. Address by C. W. Obert, Mem.Am.Soc. M.E., Secretary Boiler Code Committee, on Society Affairs.

BIRMINGHAM, SEPTEMBER 16

Trip through the Avondale Works of the Continental Gin Company and the plant of the Coyne and Joubert Foundry Company.

CINÇINNATI, SEPTEMBER 21

Joint meeting with Engineers' Club of Cincinnati. Paper: Some Present and Future Carburetion Problems, C. F. Kettering, Mem.Am.Soc.M.E.

NEW ORLEANS, OCTOBER 2

Discussion of paper on The Design and Test of a Large Reclamation Plant, G. C. Noble. Paper published in this volume.

NEW YORK, OCTOBER 10

Address: Explosives, Dr. Charles L. Reese, chemical director of the E. I. Du Pont de Nemours Powder Company.

MINNESOTA, OCTOBER 12

Held in St. Paul. Subject: Lubricants, Charles Fortner, of the Standard Oil Company.

DETROIT, OCTOBER 13

Dinner. Officers chosen as follows: M. E. Cooley, president; E. C. Fisher, T. H. Hinchman, G. W. Bissell, and J. W. Parker.

PROVIDENCE, OCTOBER 13

The Providence Engineering Society. Address: The Relation of Engineering to Public Affairs, Ex-Lieut. Governor Zenas W. Bliss.

ST. LOUIS, OCTOBER 14

L. C. Nordmeyer, Mem.Am.Soc.M.E., recounted his personal experiences in China, and also gave a description of Chinese life.

BUFFALO, OCTOBER 18

Dinner, followed by an address by President D. S. Jacobus, on Relationships of the National Societies to the Local Societies.

ST. LOUIS, OCTOBER 18

Paper: Standardized Boiler Construction, E. R. Fish, Mem. Am.Soc.M.E.

PHILADELPHIA, OCTOBER 24

Subject: The Development of Our Fleet and Naval Stations, Prof. William L. Cathcart, Mem.Am.Soc.M.E. Published in THE JOURNAL, February, 1917.

BIRMINGHAM, OCTOBER 25

Subject: The Use of By-Product Coke-Oven Gas as a Fuel, and Some of the Problems Presented in Its Use at the Ensley Steel Works, W. P. Caine, Assoc.Am.Soc.M.E., followed by short talks by Mr. Gaboury, of the Woodward Iron Company, and H. P. Ryding.

PROVIDENCE, OCTOBER 25

Providence Engineering Society. Illustrated Address: Iron, from the Ore to the Finished Article, Prof. W. H. Kenerson, Mem. Am.Soc.M.E.

BUFFALO, NOVEMBER 1

Illustrated Lecture: Motion Study, Frank B. Gilbreth, Mem. Am.Soc.M.E.

LOS ANGELES, NOVEMBER 2

Address: Deep-Mine Hoisting, E. T. Sederholm, Mem.Am.Soc. M.E.

MINNESOTA, NOVEMBER 9

Held in Minneapolis. Address: Coke and Its By-Products, E.

F. Friedman, chief chemist of the McLaughlin, Gormley, King Company, Minneapolis, Minn.

NEW YORK, NOVEMBER 14

Illustrated Lecture: Submarines, Charles H. Bedell, electrical engineer of the Electric Boat Company, New London, Conn.

BIRMINGHAM, NOVEMBER 15

Illustrated address by Paul Wright, Mem.Am.Soc.M.E., on the manufacture of cast-iron water-pipe. Followed by a discussion of the paper on The Progress of Economic Power Generation and Distribution by Samuel Insull. Paper published in THE JOURNAL, November, 1916.

BUFFALO, NOVEMBER 15

Address: The Navy; Its Fleet and Naval Stations, Prof. Wm. L. Cathcart. Published in THE JOURNAL, February, 1917.

NEW HAVEN, NOVEMBER 15

Fall meeting. At the afternoon session there were papers on Methods of Testing Metals, Prof. W. K. Shepard; Applied Metallography, Prof. C. H. Mathewson. At the evening session a paper was read by S. J. Berard, instructor in machine design, Sheffield Scientific School, on Methods of Duplicating Drawings, and Frank B. Gilbreth, Mem.Am.Soc.M.E., spoke on Developments in Time-study Methods.

CHICAGO, NOVEMBER 17

Army and Navy Night. William A. Moffet, Commander of the Great Lakes Naval Training Station, gave an illustrated address on the training of cadets; Lucian B. Moody, Major Ordnance Department, Rock Island Arsenal, spoke on the army ordnance department, and Captain Christie, of the new United States Aviation Station at Chicago, outlined the methods of training aviators.

MILWAUKEE, NOVEMBER 17

Joint meeting with the Milwaukee Engineers' Society and the Milwaukee Section of the American Chemical Society. Address: Characteristics of Base Metal Thermocouples, Prof. Otto L. Kowalke.

PROVIDENCE, NOVEMBER 22

Providence Engineering Society. Illustrated Address: City

Planning as an Engineering Problem, Nelson B. Lewis, chief engineer of the Board of Estimate and Apportionment, Bureau of Public Improvements, New York.

PHILADELPHIA, NOVEMBER 28

Subject: Mechanical Development of Aviation, Neil MacCoull, of the Westinghouse Machine Company, Pittsburgh, Pa; illustrated. Followed by Dr. Charles E. Lucke, Mem.Am.Soc.M.E., who gave a description of the problem of aeronautical engine design.

BUFFALO, NOVEMBER 29

Paper: Graphite, C. H. Bierbaum, Mem.Am.Soc.M.E.

ST. LOUIS, DECEMBER 3

Dinner. Address: Some Economic Aspects in Relation to the High Cost of Living, Prof. Franklin Gephart.

ST. LOUIS, DECEMBER 6

Joint meeting with the Associated Engineering Societies of St. Louis, under the auspices of the Engineers' Club of St. Louis. Photoplays on the subjects of railroad safety and the evolution of transportation, supplemented by a paper on Railroad Trespassing, by A. A. Krause, commissioner of safety, Missouri, Kansas and Texas Railway System.

BALTIMORE, DECEMBER 13

Illustrated address: Recovery and Use of By-Products from Coal Tar, Captain F. H. Wagner, Mem.Am.Soc.M.E.

BUFFALO, DECEMBER 13

Paper: Four-Cylinder Automobile Engine, Alfred P. Brush, consulting engineer, of Detroit.

ST. LOUIS, DECEMBER 13

The Associated Engineering Societies of St. Louis and the Engineers' Club attended a lecture, dance, and luncheon in the club rooms of the United Railways Company. Non-technical Lecture: Some Experiences of a Civil Engineer, A. G. Allan, assistant engineer, valuation department, Missouri Pacific Railway. Illustrated.

PHILADELPHIA, DECEMBER 14

Joint meeting with The Franklin Institute. Illustrated Lecture:

The Cooling of Water for Power Plant Purposes, Prof. Carl C. Thomas, Mem.Am.Soc.M.E.

CINCINNATI, DECEMBER 21

Joint meeting with the Engineers' Club of Cincinnati. Paper: The Concrete Bridges and Viaducts of Cincinnati, Frank L. Raschig.

PROVIDENCE, DECEMBER 27

Illustrated Address: Work of the United States Reclamation Service in the Far West, Prof. F. H. Newell, Mem.Am.Soc.M.E.

THE ANNUAL MEETING

One of the outstanding features of the thirty-seventh Annual Meeting of the Society, held in the Engineering Societies Building, December 5 to 8, 1916, was the large attendance. The total registration was 1868, of which 953 were members; the total registration exceeded that of the previous year by 431. There were delegates from most of the local Sections, sixteen of the twenty being represented at this meeting.

As outlined in the program which follows, the meeting was opened on Tuesday evening with the presidential address by Dr. D. S. Jacobus on Education in Engineering, and the three following days were devoted to the professional sessions and other events of the convention. There were twelve professional sessions, at which 36 papers were presented. On Friday afternoon a public hearing on the Boiler Code began which continued on Saturday.

For the first time these events, as well as the President's reception on Tuesday evening, were held on the remodelled and spacious fifth floor of the Engineering Societies Building.

At 12.30 o'clock on Wednesday morning, services were held in honor of the memory of John E. Sweet, Past-President, Honorary Member, and founder of the Society. Sessions were adjourned at this hour for the purpose of holding these services. The exercises were in charge of a special committee of which John H. Barr, Manager, Am.Soc.M.E., was Chairman. Addresses were made by Past-President Worcester R. Warner, Past-President Capt. Robert W. Hunt, and Albert W. Smith, Mem.Am.Soc.M.E.

A members' smoker was given on Wednesday evening. This was a repetition of the event of the preceding year, and was again a marked success.

On Thursday evening a symposium on Aviation was held in the auditorium of the Engineering Societies Building, at which there were four prominent speakers. Dancing followed the lecture, and refreshments were served.

On Friday evening the alumni of the following colleges held reunions, and many of the college men joined in what has become an annual event: Brown University, Cornell University, Lehigh University, Massachusetts Institute of Technology, Pennsylvania State College, Polytechnic Institute of Brooklyn, Stevens Institute of Technology, University of Illinois, University of Michigan and Worcester Polytechnic Institute.

The convention was in charge of the Committee on Meetings, H. L. Gantt, Chairman; and the entertainment in charge of the New York Section Committee, H. R. Cobleigh, Chairman, with E. J. Prindle as Chairman of the Sub-Committee on Entertainment; the President's Reception was under the direction of the House Committee, William N. Dickinson, Chairman; the welfare of the ladies was provided for by the Ladies' Committee, Mrs. Herbert Gray Torrey, Chairman.

PROGRAM

Tuesday Evening, December 5

Opening session. President's address: Education in Engineering, Dr. D. S. Jacobus. 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 6

BUSINESS MEETING

Reports of the Council, Standing Committees and Special Committees, and new business.

MISCELLANEOUS SESSION

THE PROPORTIONING OF SURFACE CONDENSERS, Geo. A. Orrok.

THE TESTING OF HOUSE-HEATING BOILERS, L. P. Breckenridge and D. B. Prentice.

WATER FOR STEAM BOILERS — ITS SIGNIFICANCE AND TREATMENT, Arthur C. Scott and J. R. Bailey.

STEAM SAFETY VALVES, George H. Clark.

EFFICIENCY OF PROPULSIVE MACHINERY AND LATE DEVELOPMENTS IN NAVAL ENGINEERING, H. C. Dinger.

STANDARDIZATION OF POWER PLANT OPERATING COSTS, Walter N. Polakov.

REPORT OF EFFICIENCY TESTS OF A 30,000-KW. CROSS-COMPOUND STEAM TURBINE, H. G. Stott and W. S. Finlay, Jr.

THE DESIGN AND TEST OF A LARGE RECLAMATION PUMPING PLANT, G. C. Noble.

Wednesday Morning

INDUSTRIAL SAFETY SESSION

PROPOSED CODE OF SAFETY STANDARDS FOR CRANES.

JOHN E. SWEET MEMORIAL

Addresses by Past-President Worcester R. Warner, Past-President Capt. Robert W. Hunt, and Albert W. Smith.

Wednesday Afternoon

MISCELLANEOUS SESSION

THE UTILIZATION OF WASTE HEAT FOR STEAM-GENERATING PURPOSES, Arthur D. Pratt.

GRAPHIC METHODS OF ANALYSIS IN THE DESIGN AND OPERATION OF STEAM POWER PLANTS, R. J. S. Pigott.

POWER-PLANT EFFICIENCY, Victor J. Azbe.

THE IMPACT TUBE, Sanford A. Moss.

THE FLOW OF AIR AND STEAM THROUGH ORIFICES, Herbert B. Reynolds.

TEXTILE SESSION

HEAT TRANSMISSION THROUGH VARIOUS TYPES OF SASH, Arthur N. Sheldon.

SPONTANEOUS IGNITION STUDIED BY MEANS OF PHOTOGRAPHIC PLATES, Frederick J. Hoxie.

VIBRATION IN TEXTILE-MILL BUILDINGS, G. H. Perkins.

TOPICAL DISCUSSION ON INDUSTRIAL MANAGEMENT

Discussion by E. E. Barney, R. B. Wolf, Frank B. Gilbreth, Sanford E. Thompson, A. J. Baker, Richard A. Feiss, H. L. Gantt, W. S. Rogers, W. O. Platt, A. R. Burnett, David Myers.

MACHINE SHOP SESSION

STANDARDIZATION OF MACHINE TOOLS, Carl G. Barth.

A PROPOSED PLAN FOR THE ACTIVITIES OF THE MACHINE SHOP PRACTICE SUB-COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, H. K. Hathaway.

RECEPTION AND TEA

Given by the Ladies' Committee in the rooms of the Society.

Wednesday Evening

Smoker held on the fifth floor of the Engineering Societies Building.

Thursday Morning, December 7

FIRST VALUATION SESSION

ACCURATE APPRAISALS BY SHORT METHODS, J. G. Morse.

HOW DOES INDUSTRIAL VALUATION DIFFER FROM PUBLIC-UTILITY VALUATION? John H. Gray.

RELATION BETWEEN PERPETUAL-INVENTORY VALUE AND APPRAISAL VALUE, Charles Piez.

Thursday Afternoon

SECOND VALUATION SESSION

VALUATION OF INDUSTRIAL PROPERTIES VS. VALUATION OF INDUSTRIAL METHODS, Walter N. Polakov.

PRODUCTIVE CAPACITY A MEASURE OF VALUE OF AN INDUSTRIAL PROPERTY, H. L. Gantt.

GAS POWER SESSION

A GAS PRODUCER FOR BITUMINOUS FUEL, O. C. Berry.

THE COMMERCIAL SAMPLING AND ANALYSIS OF PRODUCER GAS, P. W. Swain.

THE RATIO OF THE SPECIFIC HEATS AND THE COEFFICIENT OF VISCOSITY OF NATURAL GAS FROM TYPICAL FIELDS, Robert F. Earhart.

AN INVESTIGATION OF THE INTERNAL-COMBUSTION ENGINE AS APPLIED TO TRACTION ENGINES, A. A. Potter and W. A. Buck.

ILLUSTRATED REVIEW OF THE DEVELOPMENT OF THE WERKSPOOR MARINE DIESEL ENGINE, Thomas O. Lisle.

Thursday Evening

SYMPOSIUM AND ANNUAL REUNION

Symposium on Aviation held in the Engineering Societies Building. Illustrated addresses, including moving pictures of recent flights of the original Langley machine. Followed by the annual reunion and dance.

Friday Morning, December 8

MISCELLANEOUS SESSION

ILLUSTRATED REVIEW OF THE DEVELOPMENT OF OUR FLEET AND NAVAL STATIONS, W. L. Cathcart.

HEAT TREATMENT OF WROUGHT IRON CHAIN CABLE, F. G. Coburn, W. W. Webster, and E. L. Patch.

STEAM BOILER SESSION

AN ANALYSIS OF MARINE SAFETY VALVES, WITH SUGGESTIONS FOR REPAIRS AND IMPROVEMENTS, E. F. Maas.

THE TALBOT BOILER, Paul A. Talbot.

THE DOWNFLOW TYPE OF STEAM BOILER, John C. Parker.

RAILROAD SESSION

CLASP BRAKES FOR HEAVY-PASSENGER-EQUIPMENT CARS, T. L. Burton.

MECHANICAL DESIGN OF ELECTRIC LOCOMOTIVES, A. F. Batchelder.

PULVERIZED FUEL FOR LOCOMOTIVES, J. E. Muhlfeld.

Friday Afternoon

Public Hearing by the Boiler Code Committee.

Friday Evening

College Reunions.

Saturday Morning, December 9

Public Hearing by the Boiler Code Committee, continued.

No. 1543

EDUCATION IN ENGINEERING

PRESIDENTIAL ADDRESS, 1916

By Dr. D. S. Jacobus, New York, N. Y.
President of the Society

So much has been said and written of late about education that it might seem too worn a topic to bring before our Society, but having been a teacher of engineering subjects for over twenty years, and having for over ten years since that time been employed in what some are pleased to call the practical side of the engineering profession as distinguished from the teaching side, I feel qualified to pass judgment on certain features and trust you will bear with me if what I say seems more or less elementary.

Let us start with the professor — and let me say here that once a professor always a professor, provided one has felt the joy that goes with being in close touch with his students and in knowing that there is a mutual understanding and trust. Some are apt to class the professor as an impractical individual whose principal joy in life is to dream and idle away the long summer vacations, while others give him what is more nearly his just due. Let us look back on our own professors, those who taught us, and see what led to our respect and what qualities we would now regard as the most important. I feel sure most of us would not depart very far from the judgment of those who responded respecting the list which was prepared by the Carnegie Foundation for the Advancement of Teaching as the essential factors desirable in young engineers. The order of importance based on the replies received is as follows:

1 Character, integrity, responsibility, resourcefulness, initiative.
2 Judgment, common sense, scientific attitude, perspective.
3 Efficiency, thoroughness, accuracy, industry.
4 Understanding of men, executive ability.
5 Knowledge of the fundamentals of engineering science.
6 Technique of practice and of business.

Presented at the Annual Meeting, December, 1916, of The American Society of Mechanical Engineers.

As we grow older we see that the personality of the men more than any other quality left its lasting stamp, and that it is this more than anything else that causes us to look back with kindness and affection to our college days. Character is the first on the Carnegie list, and this as well as many other qualities that are to an extent independent of the course of study precede those that relate to the technical side of the course. In the discussion of engineering education the details of the courses are apt to be given undue weight, which is a grave mistake. System can never take the place of personality, and a given course that would be a success when taught by the right men might be a failure if taught by others.

Let us analyze the factors in the Carnegie list. First of all it is apparent that most of them apply to any profession and that they are not limited to engineering. Again, it might appear that one should succeed in a profession if he possesses the first factors in the list, even though he may not possess those included in the last two items. It would be a dangerous conclusion for a young man to feel that his success is assured if he has character and the other qualities which come near the head of the list, and that it is not necessary for him to have thorough knowledge of the technique of his profession. Such an unfortunate would soon find that the ethical qualities alone will not earn his bread and butter, and that a lack of knowledge in his profession will be an immense handicap.

What are the particular elements that spell success for a young engineer in addition to the elements that are common to all professions? Dear Uncle John Fritz once said that he could tell whether a young lad would be apt to succeed by watching him unobserved when sweeping a sidewalk. Yes! and we will all agree that much depends on what is in the lad. Eckley B. Coxe talked with me earnestly in the good old days as to what he considered the failure of some colleges to secure the right timber in their graduates. He compared the ordinary system of examinations to riddling the men through a series of sieves and throwing out the unfortunate fellow who had peculiarities that made it impossible for him to pass through the mathematical or other sieve. Mr. Coxe was a firm believer that there is a "something" required in the making of an engineer which cannot be gaged by an ordinary examination, and I remember well that he told me he employed some highly paid men who did not know an integral sign from a dollar mark, and that he knew of others classed as the best of graduates he would rather pay high salaries to keep out of, than in, his organization. What is the "something"?

What is it that the old-time superintendent has in mind as he shakes his head sadly when asked how some young graduate is progressing. The lad may have character, judgment, and perhaps everything in the Carnegie list, or at least he may think he has, and yet our old-time friend, often from the land of the heather, knows there is something lacking. And we ask again, what is this "something"?

Would that I could define the "something." It includes such qualities as:

Taking a personal interest in one's work
Amenability to discipline
Perseverance under adverse circumstances
Cheerfulness and amity — the human side.

None of these appear separately in the Carnegie list.

Let us analyze these qualities. It will be seen that possibly a lad from a grammar school would be as likely to possess them as the college man. If this is so, then there is a lack in this side of the college training, and it is here that we should improve. Consider the first: "Taking an interest in one's work." In this the college man often fails. I have known of cases where on being censured for lack of accuracy the college man said that he could not take enough interest in the work as it was not up to his capabilities and could be done by any office boy; and often a bright office boy would pass him. The young man to succeed must take such an interest in his work that he will take pride in doing it in the very best way, be it what it may. He should be determined to hang on until the finish. The man who has an engagement that calls him away when he is asked to help in an emergency, seldom wins. The practical foreman will often keep the young college man he is trying out on monotonous work to determine his staying powers. Industry is given as an essential in the Carnegie list, but the determination and staying powers that count for success include more than simple industry. The man must be more than industrious; he must not only work hard and continuously, but be so interested in his work that he feels a personal responsibility in it. Further, the "industry" should be of the sort that is not governed by the clock, but which calls for many nights of labor, either in home study or in the field. Success is the fruit of hard labor achieved through the building up of many things into a grand and harmonious whole. The Good Book speaks of time-servers and those who work from the heart; there is no better criterion than this to distinguish between one who will fa one who should succeed.

The second is "amenability to discipline." There is a importance assigned to this than there should be in other as in teaching lines. The old-time preceptor with his hickory stout ruler was but carrying out the home discipline where th *obey* carried its full force. Perhaps the wave will change tc back the good features of the old *régime*, — let us hope so. V tainly can and are doing much in our colleges to teach resp authority and obedience to rules, but we should do more; let overlook this most important part of an engineer's equipmei emphasize the importance of his being governed by these pri in the field.

The third is "perseverance under adverse circumstances the fourth, "cheerfulness and amity." The most unfortunate of mind a young engineer can get into is to feel that ever against him and that his efforts are not appreciated. I hav told the young men that those who select them take a pride in them succeed. Believing a thing is so may often make it so, fore the young man should encourage cheerful beliefs. To su one should make friends of those about him. There is no surer making a man an enemy than to mistrust him. Great things a rule be done only through coöperation, and in the broad a the human side has much to do with success. "Study you as well as the job" is a good motto to follow, but with it sh coupled a perseverance and cheerfulness that will induce th to study you.

In applying for a position the young graduate will often his cause by inquiring too minutely respecting the prospect vancement. Again, after being employed he may have the id in view of his special training his work should be in line w veloping this training and that he should be given a certain ence over an untrained man. Such an attitude is bound to discontent and misunderstanding. I have advised graduate times to "size up" their chances before applying for positio not to talk prospects, and have further impressed on them th must enter the outside world on an even basis with their co-w and that any advantages they secure through education mus through the ability this education gives them to do better wo the others.

Having thus briefly reviewed the qualities necessary in a neer, let us consider the student. How can we bring out those qualities which we all agree are the most important. Character can best be taught by example, and the teacher must serve as an inspiration and guide. To teach effectively one must be a master of his subject, and this applies particularly to those qualities placed near the head of the Carnegie list.

We hear and read much respecting teaching methods. Methods are not all; the spirit which unconsciously develops is a most vital factor. I cannot fail to observe that there has been a great change in one respect since I was a boy. In my school days a failure to acquire a subject was laid to the student; now there is a tendency to lay it to the teacher. Much of this is due to the attitude of the parents. A fond mother once told me she thought it strange that no teacher could be found capable of teaching her boy mathematics, not realizing that she herself might be the greatest stumbling block through transmitting the idea that the fault lay anywhere else than with her son. A way never has been and never will be found to acquire knowledge without study, and much of the trouble experienced by some students arises through a lack of study or a failure to appreciate the fact that the fault is their own.

Where the teacher is often at fault is in his failure to train the lad to study in a proper way. To memorize is the lowest form of study, and much of our present system may be justly criticised in this respect. A certain amount of memorizing is essential to "train the mind," as those who taught us used to say, but there is much in the doctrine expressed by that lovable professor, John E. Sweet, in the words, "What is the use of teaching a man a mass of material he is bound to forget?" We should develop initiative and encourage originality, and this cannot be done through memorizing or through any fixed course. It is here that a teacher who is worthy of his calling excels one who takes the easiest way. There is an easiest way in teaching just as there is an easiest way in studying, and both spell lost time and lost opportunities. The easiest way in teaching is to follow a beaten course and rely on the recitations and routine examinations in grading the men. To preserve the standard it is appreciated that the weaker men should be weeded out, and by so regulating the course that the weaker men fall by the wayside it might appear that the brighter men will remain and that there will be a "survival of the fittest." Such a course is fraught with the gravest dangers, as some of the best men may be dropped or may leave through being discouraged, and the college, though it may do good to many, may also do untold harm. Again, those who remai and graduate may have highly trained memories with little initi tive and power of reasoning. The right way is for the professor know the individual characteristics of each of his students and to governed by this in deciding a case. Often if he can get at the ro of a special difficulty it can be cleared away and there will be further trouble. Still, there are cases where the student does have an aptitude or liking for engineering, and in such instances is a kindness to start him in some other line of work.

To accomplish the best results the course should be so laid that there will be the right amount of time for rest and recreati The student should be encouraged to spend a proper amount of ti in outdoor exercise. A healthy and clean body is indispensable a healthy mind and is even more to be desired than a trained mi Just as soon as a course becomes so crowded that there is no ti for anything but work and study, it will lose in efficiency, as th will then be no time for daily relaxation of the mind and for prope developing initiative. To obtain the best results the student m work enthusiastically and cheerfully, and if the course is such t he is constantly on the anxious seat, his perspective will beco shortened and he will work at a disadvantage. We should train student to work in the same way that he should work after graduation. It is often remarked that students working their through college stand surprisingly well, considering the amount time available for study. We all agree that the man who does has determination that makes him bound to succeed, but it is determination that leads to a proper method of study and the cle ing of his mind through thinking at times of entirely different su jects than his studies that makes his brain keen and quick. T students should be encouraged to occupy their spare time in so congenial way; there is nothing better than a good, healthy fad. B it is asked, how shall we know that the spare time is used to adva tage, and would it not be best to completely fill every hour of th day and thus, as some have said, keep the boy out of mischief? It here that the human side of the professor must be brought into play and to properly apply this side he must know his pupil and know him well. The classes must not be so large that it is impossible to obtain the intimate personal relation which is indispensable.

Recitations and examinations should be so conducted as t bring out more than can be memorized and to take into account th widely varying factors of temperament. It is far easier to lead an have the students follow than to call for initiative and guide them back if they depart from the right path, but the easiest way results mainly in training the memory, whereas the other includes the developing of qualities that are most apt to lead to success.

To teach engineering efficiently the professor should have practical experience. Sometimes this is acquired before the professor begins his career as a teacher; in most cases, however, the professor undertakes practical work in connection with his college duties. To secure the best results where the professor undertakes practical engineering work, the roster should be so arranged that certain of the professors dealing with the more practical engineering lines are given latitude in meeting the classes. One must not be misled into the idea that if a certain latitude is provided it will be an easy matter for the professor to undertake this work. He will find that to accomplish results he must often work night after night, as well as through the vacation periods, and unless he is willing to do this it is folly for him to endeavor to gain practical experience and a reputation in this way.

There is no enterprise that will reflect more credit on a college than undertaking properly conducted experimental investigations, especially if those suitable for the purpose are made the bases of papers presented to engineering societies or published in the technical press. The making of experiments and the issuance of reports carry with them grave dangers, in that there may be a tendency to do work for which the professor has not the proper capacity, and reports may be made that will throw discredit on his college. A professor should not be permitted to endorse commodities where the idea is simply to obtain his name, or worse than this, his name in connection with his college, for use in advertising matter; nor should he allow his name to be used in tests which may be of more value to the promoter than to the engineering profession. Above all, he must be careful in his reports and in his testimony as an expert not to bring discredit to his profession and his college by departing a hair's-breadth from the truth or by testifying in a way which will evade the truth. There is danger in any enterprise where responsibility is assumed, and here as in all other such enterprises success or failure is dependent on the organization and character of the men. A great advantage, aside from efficiency in teaching, of having professors in close touch with the outside world, is that such professors can place many of the graduates in positions for which they are especially fitted. This naturally can only be done where the fessor, in addition to having a large acquaintance in the practical field, also comes into intimate contact with the students so as to know their individual peculiarities and characteristics.

Let us now consider the broad aspect of education. "We live to learn" applies most truly to the engineering profession. We must continue to study and learn as long as we hope to be active and advance in our profession. Just as soon as one reaches a point where he does not appreciate the value of other people's views, just so soon will he come to a standstill. It has been truly said that a big man will take most kindly to suggestions and that only the little man will resent them. We learn from day to day and soon appreciate the fact that time spent at school is but the beginning of a lifelong course. How does the school course assist us, and what should it include to be of most use? How long a course should it be, and should we send our sons to college? In the replies respecting the Carnegie qualifications, knowledge of the fundamentals of engineering science and technique of practice and of business were placed at the end of the list. If the judgment of those who responded is correct, are we not taking too much valuable time for such training, and would it not be just as well to shorten the college course? It is an open question whether much could not be eliminated from the college course to advantage, provided special attention is given to teaching the fundamentals. Once thoroughly mastered, the student will always carry with him the fundamentals of physics and mechanics, whereas much that relates to special applications will soon be forgotten. Again, the college should graduate its men young enough to preserve the adaptability of mind which the younger man possesses to a far greater degree than an older. If the assumption is made that the fundamentals are thoroughly mastered and the student graduated a year in advance of what he would be should he study along a number of specialized lines, most of those who employ graduates would favor the younger man with a keen knowledge of the fundamentals to the older man with what might be a lesser knowledge of the fundamentals and more knowledge of specialized lines. The question of whether the school course should be shortened depends on how it is shortened. Shortening the time spent in college may not shorten the course of study as a whole.

As far as can be determined from statistics, the average age of graduation from college has not changed appreciably during the past century, being somewhat over twenty-two and a half years. It would therefore seem that, irrespective of what is taught, there are elements entering a college course which have led to men graduating at about this age. I believe it advisable to have a four-year college course, but I would make the entrance requirements and the course such that the students would be graduated at as early an age as has been and is the current practice.

I have taken the stand and firmly believe that our present entrance requirements do not accurately gage the ability of an applicant, and that it is a mistake to make the entrance requirements too high for an engineering college. High technical requirements may result in the preparatory schools specializing along narrow lines in an endeavor to have the student enter college at an early age, to the sacrifice of a broader education. Again, high technical requirements discourage those who have worked in machine shops and the like and later on desire to enter a college. I have watched and encouraged such men and appreciate how hard it is for them to bring their minds back to the different studies; still, once these men get the right hold, they make the best of students and later the best of engineers. Where I have expressed this view the question has been asked, "How can we select men for the colleges if we do not rely on examinations, and how can we secure as good men with lower requirements as with higher requirements?" My reply has been that, in addition to the examinations, each and every applicant should be met by a board, and all of the circumstances connected with his case considered. Further, the examinations should be given in a way that will bring out the reasoning powers of the students, which can be accomplished if each man is given an oral examination of some length. Here again we have the harder against the easier way, but I am firmly of the opinion that there is no easy way of doing good teaching.

There can be no doubt that for many classes of engineering work much in the average college course is not needed. In certain instances we might well give over some of the time spent in specializing to teaching the fundamentals, with enough application to secure a thorough drill and to give the students confidence in their ability to solve practical problems. Fundamentals can be thoroughly mastered only through application. Further, it has been my experience that in training the average mind the fundamentals must be approached from different viewpoints, and that there is a great advantage in several teachers presenting subjects requiring the fundamentals to be applied in various ways. A student may memorize all definitions and be able to solve certain problems in a routine way, but on viewing a principle from another angle from the one to whi(he is accustomed he may be completely at sea.

The aim in teaching should be to produce a man who will be tl most useful to his fellow-men and to his country. Just as we giv so we receive. The fruit of usefulness is achievement, and the buil(ing up of achievement, success. We should above all consider tl human side and aim to produce a broad man with spirit and dete mination. It is my firm belief that in many cases we could produ(graduates who would more nearly meet our ideals by cutting ot much from the present courses of study and by teaching moı thoroughly matter already in the courses. The course should b broad enough to include some of the so-called cultural studies, an it further should deal with the business side of the profession. Dı Alexander C. Humphreys was the first to introduce a regular cours of economics in an engineering college, the course including account ing, depreciation, shop cost, law of contracts, specifications, ap praisals and business methods in general. As an illustration o matter already in the course that it would be well to teach mor thoroughly, I can refer to the writing of concise and logical report in good English. To this might be added speaking on one's feet be fore a meeting.

To really know a thing one must have more than an abstrac knowledge. Much can be gained by having the students make o witness experiments which verify or illustrate the principles. Wha a student does with his hands in connection with his head he remem bers far better than knowledge which he has obtained from text books or lectures. The courses in departments having laboratorie should be so arranged that work in the laboratories is given in paı allel with the work in the class room. There must be close coöpera tion between the departments, with this end in view. It was a paı of my pleasant duty when teaching to review the fundamentals afte they had been taught by others. Good results came through the us of apparatus to illustrate the principles and in giving the student practice in the solution of many problems in which the principle had to be viewed from different standpoints. That the apparatu helped was evidenced many times by the students sketching out th principles as applied to the experiments before they applied them t a problem, which showed that they carried the principles through visual recollection of the experiments. Again, the principles weı applied over and over again in laboratory experiments. T thoroughly master principles the hand, sight, and mind must b brought into intimate harmony and brought into harmony many times, and unless this is done the student will not have an understanding that will make the principles dependable working tools.

Instructions for laboratory practice are often so complete that the students are led to such an extent that there is a failure to call for the proper amount of initiative. This may or may not be so with a given set of instructions, all depending on the way in which the teacher presents the subject. He may often economize much time by using properly prepared notes, and he may train the initiative by inspiring the students to follow up the details and bearing of the experiments. Special exercises where the students are required to determine certain quantities without the use of any instructions, either oral or written, or with few instructions, form a most useful adjunct, but here again all depends on the way in which the subject is presented. Too much should not be done in setting up laboratory apparatus for the students. The men should be encouraged to ask questions; where questions are not asked it is a certain sign that something is wrong with the method or the teacher.

Laboratory exercises form a useful basis for practice in writing reports. The best results are secured by returning all reports with carefully marked criticisms, and requiring the student to rewrite those that do not come up to the standard. It may often require two or three or even more rewritings to make a report readable, but if this method is followed up conscientiously it will result in vast improvement. Here again the method is not the easiest one. A lack of proper examination of reports soon results in a falling off in interest and in the effort of the student. There is nothing more discouraging to a student than to have his work carelessly examined, and in good teaching much time must be spent on this part of the work.

As to education in general, we all agree that our sons should be trained along well-balanced courses. The shop foreman who received only a grammar-school education will send his son to college, for he knows what it means to be handicapped through lack of training, and how hard it is to make up for that lack of training by studying at nights later in life after long working days; as study he must to keep in the race with the college-trained man. In many instances where I have worked long enough with the men at a plant to really know them, it has been touching to find how many, especially the older ones, appreciate the fact that they have lost their opportunities. Many a time such men have come to me and said that I should make it my duty to warn young men to get an education. Yet many of these men had all the qualifications in the Carnegie list except the two items which those discussing the list have classed as the least important. One does not have to turn to the successful man to find character; many a man who has missed his chance has noble qualities and is a prince at heart. We should not therefore belittle these two items, even though we agree on the importance of those that precede them, for their lack has led many a man to failure.

One of the greatest pleasures of the professor is to watch the progress of those he knew so well after they leave him. It is indeed interesting to see the boy in the man, for although his students may grow old, there are still the same traits and peculiarities. The professor has a grave responsibility in that he directs while the mind and character are in a plastic state and his imprint may mark the destiny of his charge. It has been truly said that training a boy is like growing a young tree: easy to direct if taken in time, but hard if neglected or if started in the wrong way. Again, boys develop like growing trees, some young, sturdy trees may outtop the others, some may be checked for lack of space for development, and transplanting to some congenial spot may help. It would be a poor planter who would discard some thrifty, young tree that might not come up to all his ideas of symmetry. The good planter would so bend and train it that its development would overcome its faults.

Having outlined the general principles of engineering education, the question may be asked, how should we proceed to obtain the best results? I believe that much good would be accomplished if those who employ graduates would coöperate in laying out the courses of study. The visiting boards of some colleges are a step in the right direction, but we need something closer and more concrete than is secured through most such boards. The engineering societies could do most fitting and useful work if they would go seriously into the problem. What is needed is a careful consensus of the judgment of many. A single man or a small group of men might have radical ideas which might do much harm if enforced. Even though an idea may be essentially good, care should be taken that the pendulum is not swung too far one way or the other, or the course will be unbalanced. All of us who have taught know of the frequent requests made by engineers, usually graduates of the colleges, that instruction be given along particular lines which have been of importance to them, and that should even a fraction of such requests be complied with the course would soon be overcrowded, or if other matters were dropped would soon consist entirely of specialized branches. The demand for instruction along particular lines has resulted in the courses of technical colleges being crowded at the expense of efficiency, and no additions should be made unless these seem desirable after a thorough study of the course as a whole to make sure the added matter will not displace something more valuable or result in a lesser degree of thoroughness in some more important branch.

We often hear the statement that something should be added to teach common sense. If by this is meant adding something to keep the student awake to the fact that there is much more for him to do than memorize and that he should keep his mind alert to grasp and coördinate facts and essentials, the suggestion is a good and basic one. The idea that there is a lack of common sense in graduates of colleges is sometimes over-carried. Often the one making the statement has a mind so fixed that he regards a graduate who does not look on a thing in the exact way he does as lacking in capacity. He wrongly regards the failure to follow fixed methods in reasoning as a lack of what he considers his own good common sense. The very failure to follow fixed methods is a safeguard which has been cast about the young mind to insure advancement and progress. Were the world to follow along the lines laid out by our older friend with his highly prided but set ideas, we might soon be in a rut that would prevent all advancement, whereas the young man, though he may fail, may also win through starting out enthusiastically in new and unknown fields. Let us therefore forgive the young man if he makes mistakes in matters that seem self-evident, and remember that the very freedom of his mind is the greatest safeguard for continued progress.

Summarizing, I would make the following suggestions:

1 That there be a closer coöperation between the engineering colleges and those employing graduates, and that the engineering societies be encouraged to work along this line.

2 That the technical requirements for entrance to colleges be lowered rather than raised, that preparatory schools be encouraged to give more attention to teaching good English and to giving a broad general education, and that applicants be also judged as to their initiative and general make-up in deciding whether they should be admitted.

3 That the courses be so arranged as to train the initiative and develop the human side; that the students be taught to work in a cheerful and efficient way; that there be proper time for daily relaxation of the mind and that the students be encouraged to use this time to the best advantage.

4 That the professors get down to hard work with their students and know all of them well enough to be thoroughly acquainted with their personal characteristics.

5 That the professors in practical engineering subjects have practical experience so that they can speak with authority.

6 That professors in charge of practical engineering departments be encouraged to undertake practical engineering work, and that their college work be so arranged that they can be relieved from meeting classes when the engineering work makes this necessary.

7 That special attention be given to teaching the fundamentals of engineering science, even though this may result in the elimination of certain specialized branches.

8 That greater practice be given in the writing of concise and logical reports in good English, and in speaking on one's feet.

9 That the students be encouraged to confer with the professors, and that regular hours be provided for this purpose, all to the end that the teachers may extend a helping hand where needed and that there be a mutual understanding and trust.

No. 1544

THE PROPORTIONING OF SURFACE CONDENSERS

By George A. Orrok, New York, N. Y.
Member of the Society

During the discussion of my paper on The Transmission of Heat in Surface Condensation,[1] I was asked to present formulæ covering the application of the results of my experiments to the design of surface condensing apparatus. Since the appearance of the paper there have been a number of papers on allied subjects whose authors have approached the subject from a somewhat different standpoint, and the conclusions presented have been of varying character and usefulness in the design of surface condensing apparatus. Only one author, Loeb,[2] has presented new experimental data which can be used for checking the heat-transfer constants, but considerable mathematical work has been done along the line of following out Jordan's deductions from Osborne-Reynolds's statement of the law of heat transfer.

2 I propose in this paper to discuss the state of the art of "heat transfer in surface condensation," and to establish design formulæ for use in proportioning condensing apparatus.

THE PROBABLE LAW OF TEMPERATURE RISE

3 The law of temperature rise of the water in a condenser tube is still unknown. My own tests led me to believe that this law departed somewhat from the logarithmic law as stated by Smith and Josse, but their results are sufficiently variant to allow for a slight modification. Loeb, however, has presented a set of tests with plotted curves, from which he deduces that the law is of the exponential form, with approximately 0.9 as the exponent. My plotting of

[1] Trans. Am.Soc.M.E., vol. 32, p. 1139.
[2] Jour. Am. Soc. Nav. Eng., May, 1915.

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his numbers places the exponent nearer 0.8, both of which are no far from my adopted figure of 0.875. These variations are not o prime importance for practical design, and indeed may be caused by the apparatus used in the making of the experiments; and I am stil of the opinion that we may use $N = K\theta^{0.875}$ as the basis of our work N being the total heat transfer per sq. ft. per hour in B.t.u., θ the temperature difference, and K a constant determined by experiment.

VISCOSITY NOT AN IMPORTANT FACTOR IN DESIGN

4 It has been stated in London *Engineering* (Jan., 1914) that the extreme variation in the results of heat-transfer experiments has been caused by a neglect of the variation due to the viscosity of the water, and quite an extended argument was given to show that some of these experiments were reconciled when the correction was applied. Wilson has carried this method to its logical conclusion in his paper A Basis for Rational Design of Heat-Transfer Apparatus,[1] lately presented to the Society. Our work in the design of surface condensers however, is mainly concerned with a very small variation in temperature, the upper limit of which is fixed by the vacuum carried at somewhere between 80 and 90 deg. fahr., while the lower limit in America is between 65 and 70 deg., and in Europe perhaps a few degrees lower — in all a variation of perhaps 20 deg. In the winter we can carry good vacuums with low heat transfer and our troubles are few, but the problem is the summer problem, and here the difference in viscosity is small. As a matter of fact, I believe Osborne-Reynolds straight-line law does not apply where there is a change of state on either side of the metallic surface. Observers making use of this hypothesis have been frequently misled by the bunching of the results in the condenser range, with the consequent spreading of the results outside this range. This may be seen in Fig. 8 of Wilson's paper.

CRITICAL VELOCITIES NEED NOT BE CONSIDERED

5 There has been considerable comment on the effect of stream flow and turbulent flow on the heat transmission, and as there is a marked change in the friction characteristics between the two states of flow, the conclusion has been drawn that there will be a similar effect on the heat-transmission characteristics. Osborne-Reynolds investigated this subject and gave formulæ for the critical velocities. He finds two critical velocities, the lower, which we may call V_c

[1] Trans. Am.Soc.M.E., vol. 37, p. 47.

below which all motion is stream-line and if disturbed artificially will return to stream-line flow; and an upper critical velocity, V_a, above which all motion is turbulent. Between the two is a field in which stream-line motion may be maintained if no artificial causes upset it, or turbulency may be set up and will maintain itself when so started. He gives formulæ for these two velocities, which, following Parker, are

$$V_c = \frac{0.0388\,P}{D} \quad \text{and} \quad V_a = \frac{0.2458\,P}{D}$$

where D is the internal diameter of the tube in feet and P is Poisseuille's ratio, *i.e.*, the ratio between the viscosity and density of the water. If the values of v = viscosity and δ = density be taken in C.G.S. units,

$$P = 56.2\frac{v}{\delta}\begin{cases} = 1.000 \text{ at } 0 \text{ deg. cent. (32 deg. fahr.)} \\ = 0.734 \text{ at } 10 \text{ deg. cent. (50 deg. fahr.)} \\ = 0.455 \text{ at } 30 \text{ deg. cent. (86 deg. fahr.)} \end{cases}$$

This agrees with Osborne-Reynolds's value, which is

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

where T is the temperature in centigrade degrees (not the absolute temperature, as stated in Wilson's paper). The table of correction factors in Wilson's paper may be used as a table of P by multiplying the correction factors by 0.625, thus correcting the values to 32 deg. fahr. The critical velocities for a 1-in. No. 18 B. W. G. condenser tube will then be as follows, T being the mean temperature of the water:

T deg. fahr.	40	50	60	70	80	90	100	115	130	150
V_c	0.50	0.422	0.362	0.318	0.278	0.25	0.224	0.192	0.166	0.14
V_a	2.84	2.400	2.060	1.810	1.580	1.42	1.270	1.090	0.945	0.80

6 In condenser practice and in most of the apparatus for heat-transfer experiments it is nearly certain that the flow is turbulent above V_c. Velocities from 6 to 10 ft. per sec. are the common range in modern surface condensers, so that in every practical case the critical velocity need not be considered.

CONDITIONS DETERMINING THE VALUE OF THE COEFFICIENT K

7 Surface condensing apparatus is never tested to the capacity of the surface to transfer heat, and Gibson and Bancel[1] have shown this in their characterizing the "active" and "inactive" zones in a

[1] Trans. Am.Soc.M.E., vol. 37, p. 975.

surface condenser. The condenser must be designed for the maximum load that may be put on it when the entering water is at it maximum temperature, and additional surface must be installed a a factor of safety against dirty, oxidized tubes and the presence o undue amounts of air. The depression of the hot-well temperature below the vacuum temperature, a well-marked phenomenon in many condensers, may be eliminated by good design, and many tests by careful investigators have been reported in which the depression was zero. Certain designs of the dry-tube type may give hot-well temperatures somewhat higher than the average temperature in the condenser. The hot-well depression is especially marked where drowned lower tubes are used.

8 Neilson and others have objected to conclusion (*b*) in my 1910 paper[1] that the heat transmission is approximately proportional to the square root of the cooling-water velocity. Within the condenser range the experiments may perhaps be better represented by $KV_w^{0.6}$ instead of $KV_w^{0.5}$. Weir (Trans. I.E.S.S.), in his paper read Oct. 22, 1912, gives a curve for the relation of U to V_w for design purposes. The curve corresponds very closely to $U = 250\ V_w^{0.6}$, the maximum variation being less than 2 per cent. It is possible that a closer agreement might be secured by using more figures in the exponent, but such a procedure is hardly necessary, as probably we have already exceeded the accuracy of our experimental apparatus. The effect of air in reducing heat transmission has been shown in my 1912 paper,[2] and a term introducing the second power of the pressure ratios recommended to allow for it in design. This subject is still unsettled and will remain so until Smith's experiments are repeated on a proper scale, with known amounts of air present in the condenser. We may say that the term $\left(\frac{P_s}{P_t}\right)^2$ represents the reduction of heat transmission by air, with a possible variation of 10 per cent. The difficulty is in obtaining a figure for the amount of air present and its temperature. The use of the air bell enables the operating man to keep his leakage down to the minimum, and where water-jet air pumps are used the addition of the testing dry vacuum pump and air bell will usually pay for themselves in a very short time.

9 The attempt to find a better tube material than Admiralty brass has failed, and the standard tube today is a heat-treated Admiralty-mixture tube whose material coefficient is 0.98. This coef-

[1] Trans. Am.Soc M.E., vol. 32, p. 1135.

[2] Trans. Am Soc.M.E., vol. 34, p. 713.

ficient can now be neglected. A number of experiments with dirty and oxidized tubes have been made, but since it is possible to clean a tube by mechanical means and secure the original heat transmission, the cleanliness coefficient may be also neglected in design and only used in checking up a condenser test where allowance must be made for a dirty tube condition.[1]

10 Assuming that I had practically air-free steam in the experimental apparatus and clean tubes, the value of K is $K = \frac{U\theta^{\frac{1}{8}}}{V_w^{0.6}} = 470$. The value 630 given in my paper was arrived at by assuming an air pressure of about 0.065 in. of mercury. If the air-term influence be taken to the second power, $K = 570$. In both cases 90 per cent cleanliness and 0.98 for material were assumed.

11 We know that the results of single-tube experiments may be approached in real condensers, and the closer the approach when the condenser is supplied with sufficient steam the better the performance of the condenser. We know that they never can be approached with light loads or with colder water, because only so much heat is transferred as is present. This definitely prevents the use of Jordan's "mass flow" formulæ or the electric resistance simile. These hypotheses can only be true where no change of state takes place, and this condition is entirely foreign to heat transfer in condenser practice.

12 We may now put down the following principles:

a The law of temperature rise in the tube is of the exponential form, with $0.875 = \frac{7}{8}$ (0.9 or 0.8, perhaps) as the exponent.

b We may neglect viscosity for design purposes in America where the water temperature is 65 to 70 deg. fahr. and the vacuum 28 in. and over.

c We may also neglect critical velocity, since all condenser velocities are far above it.

d For velocities in condenser practice we may take K as varying as the 0.6 power of the velocity.

e The reduction in heat transmissive power due to air is very approximately covered by the use of a term $\left(\frac{P_s}{P_t}\right)^2$ as a reducing factor of K.

f For design work we may reduce the value of K once for all to account for the cleanliness of the tube and the material itself.

g The value of K will then be reduced to somewhere between 300 and 400 for design purposes, say 325 for average good working.

[1] Report of Com. on Prime Movers, Natl. Elec. Light Assn., 1916.

CONDENSER DESIGN

THE AUTHOR'S METHOD

13 Condenser specifications usually call for the condensation of a certain amount of steam per hour, W (lb.), and the maintenance of a vacuum referred to standard barometer or absolute pressure, P_v, with condensing water at a certain temperature, t_0. The quantity of condensing water, Q, the hot-well temperature, t_s, the power used by the auxiliaries and the steam consumption of the auxiliaries may be also specified, but the problem is reasonably determinate when W, P_v, and t_0 are known. It is important that the place of measurement of the vacuum P_v be specified, as there is a well-defined drop through most condensers, the vacuum being greatest at the air-pump suction, less in the condenser, and least at the prime-mover nozzle. This drop may amount in well-designed condensers to 0.2 in. of mercury. The specified vacuum should be measured at the prime-mover nozzle, and the designer will then allow 0.1 to 0.2 in. for the drop in the condenser.

14 The absolute pressure in the condenser will then be 29.92 − (vacuum + 0.2 in.) = P_v. The steam tables may now be consulted and t_s the temperature corresponding to the vacuum, λ = the total heat and q = the heat of the liquid be taken therefrom. The final temperature of the condensing water approaches t_s, and for close work may be $t_1 = t_s - 5°$. In ordinary practice $t_1 = t_s - 7°$, or in some cases 8° or 10°. The ratio of condensing water to steam condensed, which is $R = \frac{\lambda - q}{t_1 - t_0}$, may next be found. Many designers consider it close enough to use 1000 for $\lambda - q$, on account of the moisture in the exhaust steam, and some few use 968. I prefer to use $\lambda - q$ not on account of its accuracy but as a slight factor of safety. Q, the quantity of condensing water = WR in lb. per hour.

15 At this point the size and thickness of the condenser tubes as well as the water velocity must be chosen. Most of the large installations in Eastern United States use 1-in. tubes on account of bad water conditions, although it has been stated that smaller tubes are more efficient (*vide* Weighton, Neilson, and Weir). There are no definite experiments to decide this point, although from Loeb's work on ¾-in. tubes we may deduce a value of $K = 600$ against $K = 470$ for my experiments on a 1-in. tube. Nevertheless, it is doubtful if anything is saved by the use of anything smaller than ¾-in. when the

pumping head is considered. $1\frac{1}{4}$-in. tubes have also been used, but the consensus of opinion seems to be that a 1-in. tube can be cleaned better, keeps in good condition longer, and can be made better than either smaller or larger tubes. Let us take 1-in. tubes, No. 18 B.W.G., as standard. The formulæ may readily be altered for other sizes of tubes when the constants are known.

16 With 1-in. tubes velocities from 7 to 10 ft. per sec. are now usual. We may take 8 ft. per sec. as the standard value for V_w.

17 The number of 1-in. tubes in one pass will be given by the formula

$$n = \frac{Q}{990\, V_w}.$$

18 The length of water travel (l), or the total tube length, may be found from

$$l = \frac{30.8\, Q}{325\, V_w^{0.6}\, n}\left[(t_s - t_0)^{0.125} - (t_s - t_1)^{0.125}\right]$$

and the total tube surface S from

$$S = 0.262\, nl.$$

19 With 1-in. tubes from 40 to 80 will occupy 1 sq. ft. of tube-plate area, depending on the spacing and general design. The average will be about 60, making the cross-sectional area of the condenser (sq. ft.) equal to ($n \times$ number of passes) $\div$ 60. If there are two passes the length of the condenser will be $L = \frac{l}{2}$ + depth of water boxes.

NIELSON'S METHOD OF DESIGN

20 Neilson, in his paper read before the Institution of Engineers and Shipbuilders of Scotland, has proposed the following method of design.

21 W, P_v (= absolute pressure of vacuum, lb. per sq. in.), and t_0 must be given. The diameter, d, and thickness of tubes, t, in inches, and the water velocity in ft. per second, V_w, must be assumed. He also assumes nine constants, to be determined by experiment, which are as follows:

$$\left.\begin{array}{ll} A = \text{air-richness factor} & = 1 \\ Y = \text{cleanliness factor} & = 5 \\ Z = \text{design factor} & = 1 \end{array}\right\} \text{for good practice}$$

$$\begin{array}{ll} C_1 = 10 & C_4 = 0.5 \\ C_2 = 18 & C_5 = 0.5 \\ C_3 = 70 & C_6 = 2.5 \end{array}$$

He states that these values may require some adjustment. The temperature of the discharge water, t_1, is assumed, and he then finds

$$Q = \frac{1000\,W}{t_1 - t_0}$$

22 The steam temperature corresponding to P_v is taken from the steam tables, and the mean temperature difference θ is found from

$$\theta = t_s - \frac{t_1 + t_0}{2} - C_6$$

23 The number of tubes in one pass is next found from

$$n = 0.000816 \frac{Q}{V_w\,(d - 2\,t)^2}$$

24 He now finds the weight of steam condensed per square foot of surface per degree difference of temperature, or

$$\frac{U}{1000} = \frac{C_1}{\dfrac{C_2}{\dfrac{P_v}{A} + \dfrac{1}{Z}} + Y + \dfrac{C_3 d^{C_4}}{C_5 + V_w}}.$$

25 The surface necessary is then found from

$$S = \frac{W}{\dfrac{U}{1000} \times \theta}$$

and the length of tube element

$$l = 3.82 \frac{S}{nd}$$

which may be divided in passes to get a convenient tube length.

26 He claims that high steam velocity through the tube bank is conducive to high heat transmission, and recommends the following formula for the area between the tubes:

$$\text{Area} = \frac{W V_s^{0.5}}{C_7}$$

where W is the weight of steam condensed per hour, lb., V_s the volume in cubic feet of 1 lb. of steam at vacuum pressure, and C_7 a constant varying from 35,000 to 55,000.

MORROW'S FORMULÆ FOR CONDENSER DESIGN

27 Morrow, in his book on Steam Turbine Design (Longmans & Co., London), has given a multiplicity of condenser formulæ. He, following Weighton's investigations, establishes three ratios as the

fundamental basis together with their "equation of compatibility," namely, $\frac{Q}{W} = R$, $\frac{W}{S}$ = condensation rate, and $\frac{S}{A}$ = surface-section ratio, where A is the water area of one pass. Then $\frac{Q}{W} \times \frac{W}{S} \times \frac{S}{A} =$ 223,000 V_w is his equation of compatibility, which must always be true.

28 W, the vacuum (and hence t_s), and t_0 are the given quantities as before: $\frac{Q}{W} = \frac{1050}{t_s - t_0 - t_a}$; t_a is the "drop," or difference between the steam temperature and the outlet water temperature, and may range from 2 to 3 deg. for conditions where $t_0 = 90°$ and $\frac{W}{S}$ is below 15, to 14 deg. for $t_0 = 50°$ and $\frac{W}{S} = 35$.

29 The water velocity in the tubes is next assumed, and the number of (1-in.) tubes in one pass is found from

$$n = \frac{Q}{990\, V_w}.$$

30 He gives no method of finding l, but says it is common practice to make l as long as possible, putting the limit at 36 ft. with ¾-in. tubes and 28 ft. with ⅝-in. tubes. Assuming a length l, he finds

$$S = \frac{lQ}{3780\, V_w}$$

31 $\frac{Q}{W}$, $\frac{W}{S}$ and $\frac{S}{A}$ are now found, and the equation of compatibility may be applied.

BAUER AND LASCHE'S RECOMMENDATIONS

32 Bauer and Lasche (Marine Steam Turbines, Swallow's translation, Henley, New York) start with W, t_0 and t_s, as before. They recommend a drop of 7 deg., so $t_1 = t_s - 7°$. R is calculated from $R = \frac{(\lambda - q)}{(t_s - 7°) - t_0}$. $Q = WR$, and V_w is assumed at 6 to 8 ft. per sec. U is taken at 450 for 6 ft. per sec. and 654 for 8 ft. per sec. $\theta = t_s - \frac{t_1 + t_0}{2}$, and the rest of the calculation is made on the basis of $W(\lambda - q) = SU\theta$, n or l being assumed.

BEST TUBE LENGTHS AND TEMPERATURE RISES

33 It will be noticed that Neilson, Morrow, and Bauer and Lasche use the arithmetical mean for the mean temperature differ- ence. Neilson says, "Equations for θ_m involving the use of hyp bolic logarithms are not to be recommended for condenser desig the reason being that the steam-air mixture in the condenser is n

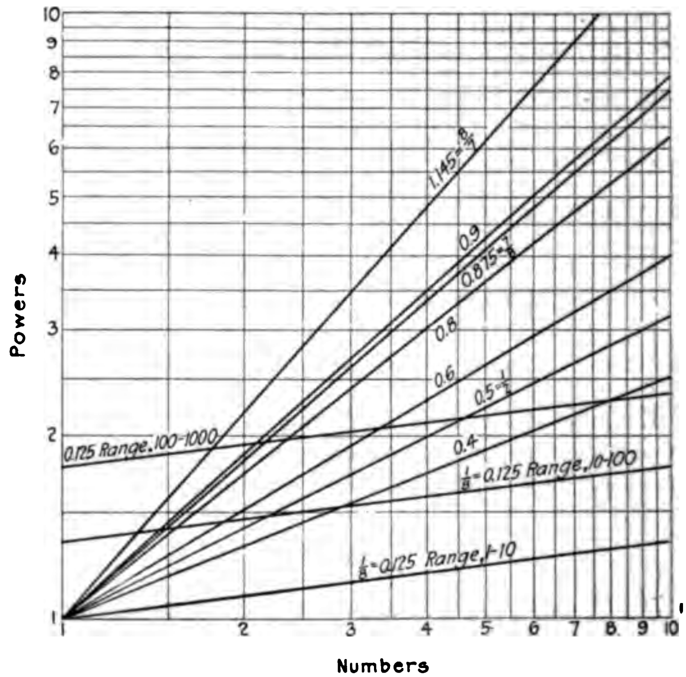


FIG. 1 LOGARITHMIC CHART FOR DETERMINATION OF FRACTIONAL AND DECIMAL POWERS OF NUMBERS

homogeneous. Weighton, upon whose experiments Morrow's wc is largely founded, did not consider θ at all in his paper. Ma other authorities follow this rule also, and I must acknowledge t position to be correct providing that equal increments in the t perature rise of the water in the condenser tube accompany e increments of length. The tests of Smith and Josse as well a own have definitely proved that this is not so. Thermometers p in the water boxes of any two- or three-pass condenser will also this fact very plainly.

34 Now for each curve of temperature rise except the arithmetic there is a definite relation of $\frac{l}{d}$ which is best for any given condition. Under the arithmetical law any values of d, n, and l which will give the necessary surface should be equally efficient, and 10 tubes 100 ft. long should work as well as 1000 tubes 1 ft. long from the heat-transfer standpoint.

35 The differences between the logarithmic law and the exponential with an exponent between 0.9 and 0.8 are quite small, but the length of the tube required by the formulæ is considerably shorter using the logarithmic formula. These length formulæ may be derived from the expressions for S given in the fourth column of the table on p. 1152, vol. 32, Trans. Am.Soc.M.E.

36 Taking the steam temperature t_s at 90 deg., the entrance water temperature t_0 at 70 deg., and the discharge at 80 deg., with a water velocity of 8 ft. per sec. and 1-in. tubes, I figure the best tube lengths and temperature rises as follows:

	Tube Length, Ft.	Rise in Deg. Fahr.
Arithmetical Law	23.2	0.4310 deg. per ft. of tube
Logarithmic Law	24.0	0.4918 deg. in first ft.
Exponential Law (exponent = 0.875)	26.8	0.5624 deg. in first ft.

It is to be hoped that some investigator will definitely settle this question on which so much in condenser design depends.

37 Fig. 1 is a logarithmic chart for the ready determination of such fractional and decimal powers of numbers as are called for by the formulæ given in the paper. For example, the 0.6 power of 4.5 is 2.5, found by following the ordinate erected at 4.5 on the horizontal scale to its intersection with the line marked 0.6, and then projecting horizontally to either vertical scale.

DISCUSSION

William Kent said he had no fault whatever to find with this paper, except that it needed an appendix, and he hoped that next year the author would give another paper supplementing this one.

The principal question he wanted answered was: Given, say, a 1000-h.p. engine, requiring 10 lb. of steam per horsepower-hour, making 10,000 lb. of steam per hour, with 29 in. of vacuum and circulating water at 70 deg. fahr. temperature, how many square feet of cooling surface should be put in the condenser, and how should this surface be arranged? The temperature of the cooling the vacuum, and the number of pounds of steam ... are the data we must start with, and the number ... surface and its arrangement are to be determined.

ILLIAM D. ENNIS (written). In their present ... r's several papers would all of them need to be consulted ... definite program for procedure in condenser design ... ing is offered as a summary and amplification of his method.

t W = maximum weight of steam to be condensed, lb. per hr.

t_s = temperature of steam

Q = weight of condensing water, lb. per hr.

t_0 = inlet temperature of condensing water

t_1 = outlet temperature of condensing water

S = aggregate external surface of condenser tubes, sq. ft.

U = coefficient of transmission

t_m = mean temperature difference

L = latent heat of vaporization of steam at temperature t_s

x = dryness of the steam at temperature t_s.

ien

$$WxL = Q\,(t_1 - t_0) = SUt_m \qquad [1]$$

ie following rules will fix values to be taken in the above: H' denotes the B.t.u. in 1 lb. of steam at throttle conditions, heat of liquid corresponding with the temperature t_s, and B eam rate of the engine, lb. per i.h.p.-hr., then

$$xL = H' - h_s - \frac{2545}{B} \qquad [2]$$

rule is on the safe side, since it assumes all heat not converted work to be still present in the exhaust.

or t_1 use a value 5 deg. to 10 deg. below that of t_s.

e steam temperature t_s is that corresponding with the absolute steam pressure p_s at the exhaust nozzle. If v denotes the vacuum at the condenser, in. of mercury, and v_n the loss of vacuum een condenser and exhaust nozzle (also in. of mercury), then, ing the influence of any air present,

$$p_s = 14.696 - 0.493\,(v - v_n) \qquad [3]$$

ood design, Orrok uses $v_n = 0.2$.

e value of U that is now recommended by the author is $\ldots/t_m^{\frac{1}{4}}$, where V = water velocity in tubes, ft. per sec., which water velocity varies from 7 to 10. In usual practice, this is equivalent to a value of U around 800, or about four times the value once employed. The expression given for U is sufficiently applicable for any standard material in a new tube. (Foul tubes may decrease U 50 per cent.) The effect of the steam-richness ratio is now thought to be represented by $(p_s/p_t)^2$ rather than by $(p_s/p_t)^5$ as in Vol. 32, p. 1162. With tight condensers and good vacuum pumps (p_s/p_t) may be expected, according to the author, to lie between 0.95 and 0.97. The constant in the expression for the value of U has been made to contemplate such values. Such values of U for commercial design are further considered below.

The mean temperature difference, assuming that the exponential law holds (see Trans.Am.Soc.M.E., Vol. 32, p. 1211), is

$$t_m = \left[\frac{\frac{1}{8}(t_1 - t_0)}{(t_s - t_0)^{\frac{1}{8}} - (t_s - t_1)^{\frac{1}{8}}}\right]^{\frac{8}{7}} \qquad [4]$$

The arithmetical mean is $t_s - \dfrac{t_1 + t_0}{2}$. The logarithmic mean is

$$\frac{t_1 - t_0}{\log_e \dfrac{t_s - t_0}{t_s - t_1}} \qquad [4a]$$

(See Trans.Am.Soc.M.E., Vol. 32, p. 1211.) This gives a value close to t_m.

The expanded equation for design is then

$$Q\,(t_1 - t_0) = \frac{325\,V^{0.6}S}{t_m^{\frac{1}{8}}} \cdot t_m = 325\,V^{0.6}S\,t_m^{\frac{7}{8}}$$

or

$$S = \frac{Q}{40.63\,V^{0.6}}\left\{(t_s - t_0)^{\frac{1}{8}} - (t_s - t_1)^{\frac{1}{8}}\right\} \qquad [5]$$

Let d = outside diameter of tubes, in.
t = thickness of tubes, in.
n = number of tubes in each pass of condenser
l = aggregate length of passes, ft.

Then area per tube for passage of water $= \dfrac{0.7854\,(d - 2\,t)^2}{144} =$ $0.00547\,(d - 2\,t)^2$ sq. ft. Water velocity = cu. ft. per sec. ÷ aggregate area per pass

$$= V = \frac{Q}{62 \times 3600 \times 0.00547\,n\,(d - 2\,t)^2} = \frac{Q}{1220\,n\,(d - 2\,t)^2}$$

or

$$n = \frac{Q}{1220\,V\,(d - 2\,t)^2} \qquad [6]$$

Also, $S = \frac{\pi d}{12} nl$, whence

$$l = \frac{3.83\,S}{dn} \qquad \text{[7]}$$

As an example, assume a unit of 2000 maximum i.h.p. at 226 lb. throttle pressure and 27.68 in. vacuum, using 12.725 lb. of dry steam per i.h.p.-hr. Take the inlet water at 70 deg., outlet water at 95 deg. and water velocity at 8 ft. per sec.

By Equation [3], $p_s = 14.696 - 0.493\,(27.68 + 0.2) = 1.0$, whence t_s (from the steam table) is 102 deg. Then by Equation [4] the mean temperature difference is

$$t_m = \left(\frac{\frac{1}{8} \times 25}{32^{\frac{1}{8}} - 7^{\frac{1}{8}}}\right)^{\frac{8}{7}} = 16.64 \text{ deg.}$$

(Note that the arithmetical mean is 19.5 deg. and — by Equation [4a] — the logarithmic mean is 16.5 deg.)

At 226 lb. throttle pressure, $H' = 1200$. Since $h_s = 70$, Equaon [2] gives

$$xL = 1200 - 70 - \frac{2545}{12.725} = 930 \text{ B.t.u.}$$

Then by Equation [1], $WxL = 12.725 \times 2000 \times 930 = 23{,}668{,}500$ B.t.u. per hr. Setting this equal to $Q\,(t_1 - t_0)$, the weight of circulating water per hour is

$$Q = \frac{23{,}668{,}500}{95 - 70} = 946{,}740 \text{ lb.}$$

Equation [5] now gives the surface directly:

$$S = \frac{946{,}740}{40.63 \times 8^{0.6}} \left\{ (102 - 70)^{\frac{1}{8}} - (102 - 95)^{\frac{1}{8}} \right\}$$

$$= \frac{946{,}740 \times 0.2669}{40.63 \times 3.485} = 1780 \text{ sq. ft.}$$

If 1-in. No. 18 B.W.G. tubes are used, Equation [6] gives

$$n = \frac{946{,}740}{1220 \times 8\,(0.902)^2} = 119$$

as the number of tubes per pass. From Equation [7]

$$l = \frac{3.83 \times 1780}{1 \times 119} = 57.3 \text{ ft.}$$

This is the aggregate length of all passes. In the absence of certainty as to values of U and t_m, no effort is here made to choose a *best* ratio of l to d. Usual ratios are between 30 and 50, indicating that two passes will be desirable in this condenser.

(The value of U for these conditions works out $325 \times 8^{0.6} \div 16.64^{\frac{1}{4}} = 796$.)

It is perhaps unreasonable to expect that definite and generally satisfactory values of U will ever be established. The transmission will inevitably decrease in service, the air-richness influence is doubtful, and the question of steam circulation has not been (perhaps cannot be) included. The rate of heat transmission in a surface condenser is a variable quantity, like the rate of transfer in a

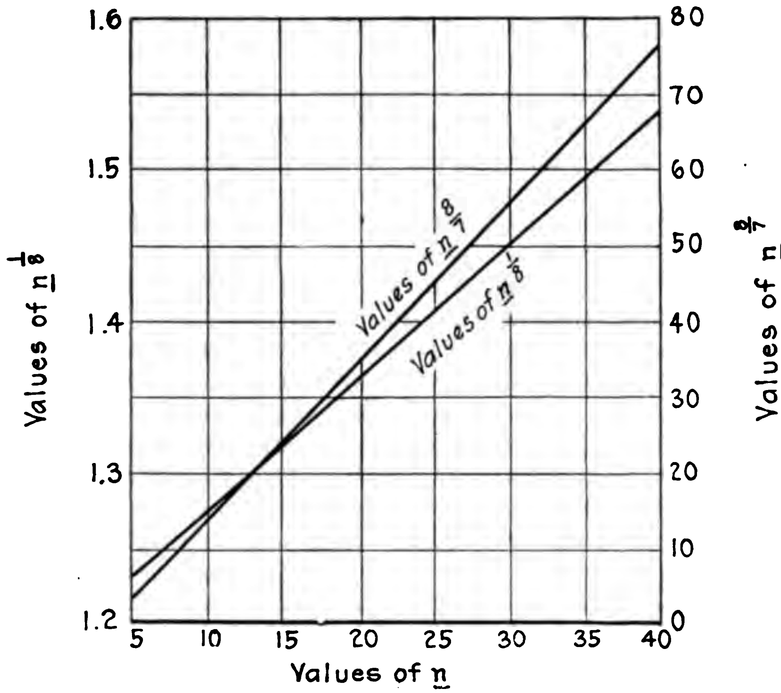


FIG. 2 GRAPH FOR EVALUATION OF EQUATIONS [4] AND [5]

steam boiler. For safe design to meet anticipated operating conditions U should probably be taken at about 400, with due allowance for factors known to influence its value. Perhaps Weir's relation, $U = 250\ V^{0.6}$, may be found safe if divided by a *hedging* factor of 2.

The mean temperature difference, which is only a slight factor in the determination of U, is itself a nearly linear factor in the total transfer. Accurate knowledge of its value is therefore of first importance. The weight of evidence points toward an exponential law for the rise of water temperature in the tube, but the logarithmic mean given by Equation [4a] represents very closely the res of such law for any ordinary conditions. Moreover, in Equati [4], exact values of the exponents are not fully settled.

A quick method for evaluation of Equations [4] and [5] is afford by Fig. 2.

In an oral discussion of the paper, Professor Ennis said that the author would explain the statement: "The law of temperatu rise in a condenser tube is of the exponential form, with 0.875 (or 0.8, perhaps) as the exponent," the surface capacity in a proble like that of Mr. Kent's could be readily determined.

THOMAS C. MCBRIDE called attention to Par. 14, in which it stated that the final temperature of the condensing water approach to within 5 or 10 degrees of the temperature corresponding to vacuum, and said that this close working had at one time be considered by some an index of good condenser design, but that now days much larger circulating-water ratios were customary, with consequent increase in mean temperature difference and increas amount of steam condensed per square foot.

Attention was also called to the fact that length of tube seem to be considered only incidentally, whereas the length assumed required by local conditions, because it determined the number tubes and, therefore, the velocity of the water in them, seemed be a most important feature to be considered in condenser desi With this thought in view, he submitted an equation showing relationship between the length of water travel through the tul and the amount of surface required, and pointed to the fact that t gradual increase in circulating-water ratio of the last few yea together with the desire to increase the velocity of the water in tl tubes without excessive power consumption of the circulating pum would lead to a more general adoption of single-pass condensers i the future, particularly in large units.

GEO. H. GIBSON and PAUL A. BANCEL (written). The autho has rendered a valuable service by gathering together in this an former papers all that is known about the influence of water velocity temperature and temperature head upon rate of heat transfer throug the walls of tubes, the sum of which knowledge he has added by his own experiments. He has, however, evidently given mo thought to the devising of algebraical formulæ which would expre these relations than to explaining the fact, which is clearly pater that single-tube experiments and practical condenser results are 100 per cent or more apart.

For example, in Par. 10 values of the coefficient K varying from 470 to 630 are mentioned as experimental results. Upon proceeding to put down principles which can be used in designing, however, the author says "The value of K will then be reduced to somewhere between 300 and 400 for design purposes, say 325 for average good working." As a matter of fact, coefficients equivalent to 300 have been attained for years in old-fashioned condensers without the grace of new-fangled formulæ and with tubes packed 100 or more to the square foot and only 3 or 4 ft. per sec. water velocity, whereas the author recommends about 60 tubes per sq. ft. and 8 ft. per sec. water velocity. If all the new formulæ get us no farther than this, why formulate?

The conclusion is unescapable that what takes place in an individual tube surrounded by pure steam is, in the actual condenser, submerged and obscured by some other factor or factors. Has sufficient study been given to the flow conditions on the steam side of the tube? Some tubes, as those near the inlet, are bathed in pure steam at all times, others receive more or less according to load and vacuum, and still others perhaps never see any real steam at all. The author recognizes this fact, when he admits that single-tube experiments are not approached with lighter loads or colder water. His explanation, however, that it is "because only so much heat is transferred as is present," obscures the truth. In an ideal condenser lighter loads and colder water ought to be accompanied by vacua proportionately nearer the limit set by the circulating-water temperature; in other words, if heat transfer is dependent only upon water velocity, temperature difference, and air, the same high coefficient of transmission should obtain as at lower vacua. The reason that a high average coefficient is not realized with cold water is undoubtedly that, due to the great specific volume of the steam at high vacua, the steam is throttled in its flow through the tube bank and not all, nor nearly all, of the tubes are active. As the author states in Par. 13, this throttling may amount to as much as 0.2 in. of mercury in well-designed condensers. It is apparent that increased specific volumes, accompanying higher vacua, will be accompanied by still greater throttling.

Sufficient consideration has not been given to the proposition that just as there is a best tube length for the water flow, so there is a best tube-bank depth for the steam flow. As condensers grow in

size, it is not correct merely to make them deeper in propo
they are made wider. The best depth must be determi
increased capacity obtained by spreading out in the other
sion. Steam lanes in the tube bank do not solve the probl
returning to the water-flow analogy, they are like by-passin
directly from the cold-water box to the warm-water box.
the heat-transmitting surface is by-passed by the lanes and
stagnant and inactive because of accumulation of air.
solution which has been proposed, viz., the attempt to kee
tube in the condenser bathed in pure steam summer and
heavy loads and light loads, by pulling a larger proportio
steam through into the air pump, leads to an excessive pov
sumption by the latter.

Mr. Bancel supplemented the foregoing written discus
considering the effect of the size of condensers on efficie
increasing the size of condensers, as is constantly being d
said, it has been assumed that the condenser would be more
in large sizes. One reason for this seems to be that the tu
more efficient in large sizes; but the problems which come i
denser design are very different from those which enter i
case of the turbine.

The turbine exhaust area and the area for flow of steam i
condenser are increased with the capacity. If the cooling
and number of tubes are increased in proportion to the cap
is necessary then, assuming the tube arrangement to rem
metrically similar, to increase the depth of the condenser
square root of the capacity. Suppose that the capacity h
doubled. Then the flow is doubled but the steam-flow
creased only by $\sqrt{2}$, making the velocity $\frac{2}{\sqrt{2}}$ as great, and t
sure loss for the same depth approximately $\left(\frac{2}{\sqrt{2}}\right)^2 = 2$ times
But the depth of the tube bank is $\sqrt{2}$ as great, which means
erably more than twice the pressure drop if the steam is to re
added tubes below the original depth. It is evident, therefo
the layout of the tubes in a condenser is of great importance
is of little benefit to know what heat transmission will be o
with given conditions of water velocity, etc., in a given-size
the steam never reaches the tube.

M. C. Stuart (written). Undoubtedly the data presented in the author's 1910 paper [1] show conclusively that the heat-transfer factor did vary as a function of the mean temperature difference between steam and water. The runs used to reach this conclusion, however, included only those in which the steam temperature was constant and the inlet-water temperature varied.

Gibson and Bancel [2] have raised the question, "What would be the effect if the water temperature remained practically constant and the steam temperature varied over a considerable range?" My analysis of the author's groups of runs n which the steam temperature was approximately 115, 126 and 187 deg. fahr., respectively, and the inlet-water temperature approximately 50 deg., shows a constant heat-transfer factor at constant water velocity of 8.7 ft. per sec. In other words, the heat-transfer factor did *not* vary with temperature differences when the variation in temperature difference was obtained by varying the steam temperature. This same conclusion was reached by Gibson and Bancel in their analysis of data presented by Loeb. A set of runs may show that the heat-transfer factor varies with the temperature difference, but this does not mean necessarily that the variation is *because* of the temperature differences. At constant steam temperature, small temperature differences correspond to high actual water temperatures, and it is altogether possible that the variation as obtained by the author was due to the variation in actual water temperature rather than to the variation in temperature difference. I agree with the author that for the narrow ranges of temperature existing in condenser work (from 70 to 90 deg.) the effect of viscosity need not be considered. It would also appear that the effect of varying temperature differences need not be considered, especially in view of the evidence that the heat-transfer factor varies with actual water temperature and not with temperature differences.

Plotting velocity against heat-transfer factor on log paper gives a curve which indicates $k = 0.5$ to 0.6 in the formula $U = CV^k$ over the velocity range from 2 to 8 ft. per sec. The formula connecting heat-transfer factor and velocity proposed by Carrier in his paper on air-conditioning apparatus,[3] was $U = 1/[a + (b/V)]$. This form of equation fits the author's data remarkably well over the entire range of velocity.

[1] Trans.Am.Soc.M.E., vol. 32, p. 1139.
[2] Trans.Am.Soc.M.E., vol. 37, p. 975.
[3] Trans.Am.Soc.M.E., vol. 33, p. 1095.

Another question in the design of condensers and heat-transfer apparatus is the choice of velocity to be used. In order to show the effect of velocity on the other variables of the design, the author's method has been applied for velocities from 2 to 16 ft. per sec. inclusive, the conditions assumed being those given in Par. 16 of the paper, namely, steam temperature at 90 deg., inlet water at 70 deg., and outlet water at 80 deg. The effect of the various velocities on unit tube lengths, total tube lengths, and friction losses shown in Table 1.

TABLE 1 EFFECT OF WATER VELOCITY ON CONDENSER TUBE LENGTHS AND FRICTION LOSS

Velocity, ft. per sec.	Unit tube length, ft.	No. of tubes per million lb. water per hour	Total length of tubing, ft.	Friction drop, feet of water
2	14.94	502	7500	0.58
4	19.71	251	4947	2.48
6	23.18	167	3871	5.87
8	26.00	125	3250	10.73
10	28.44	100	2844	17.28
12	30.59	84	2570	25.24
14	32.53	72	2342	34.80
16	34.32	63	2162	46.10

The values in column 2 for unit tube lengths were determined from the author's formula, and those in column 3 were obtained from a consideration of velocity and tube area. The total tube length required is the product of unit tube length and number of tubes. The friction drop was computed from the most authoritative data for friction drop in brass tubes,[1] and includes velocity losses at entrance and exit to tubes in a two-pass condenser. Each horizontal line of Table 1 represents a possible design which will satisfy all of the assumed conditions. The length of the condenser will be proportional to the unit tube length. The diameter or cross-sectional area will be proportional to the number of tubes. The size, weight and cost will be proportional to the total length of tubing. The power required to pump the circulating water will be proportional to the friction drop. The choice of velocity must evidently depend upon a careful consideration of each of these factors. The use of a high velocity results in a smaller condenser at the expense of increased friction drop. The exact relation between the

[1] Saph and Schoder, Proc.Am.Soc.C.E., vol. 51 (1904), p. 253 *et seq.*

size of condenser and friction drop is shown by the curve of Fig. 3, on which curve the velocities and unit tube lengths are also indicated. At a velocity of 8 ft. per sec. the friction drop amounts to 10.73 ft. of water and the total length of tubing amounts to 3250 ft. per million pounds of water per hour. By increasing the velocity to 12 ft. per sec. the total length of tubing may be reduced to 2570 ft. at the expense of an increase in the friction drop to 25.24 ft. of water. I am not advocating the use of any particular velocity,

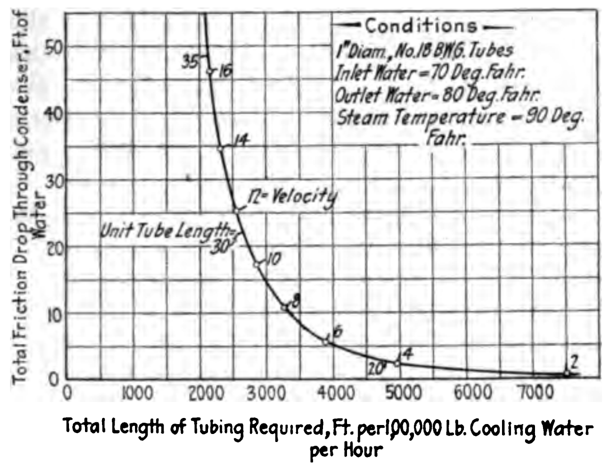


FIG. 3 RELATION BETWEEN SIZE OF CONDENSER AND FRICTION DROP

but simply placing emphasis upon the fact that the proper choice of a velocity should be given more consideration.

G. L. KOTHNY (written). The author emphasizes the fact that the heat transmission is reduced due to the presence of air inside of the condenser, and that the reduction is approximately covered by the use of a term $(P_s/P_t)^2$ as a reducing factor of the constant K. He also states that the difficulty is in obtaining a figure for the amount of air present and its temperature. In Par. 12*g* he gives the value of K as 325.

It would be interesting to hear what amount of air leakage in pounds per hour should be assumed per 100 sq. ft. of surface when using 325 as the value of K.

The paper has for its object the proportioning of surface condensers, and as the amount of air coming into a condenser through leakage is a very important factor when determining the heat transmission, it would seem that an attempt should be made by the different condenser manufacturers to agree on the maximum and minimum amounts of air leakage permissible for certain sizes of condensers. In other words, just as rules have been laid down for stresses in different materials, so should rules or curves be established or agreed upon which will show the minimum and maximum amounts of leakage for a certain size of condenser. It would be an easy matter to determine these figures from installations in operation by making readings when the apparatus is under best working conditions and again after it has been in service for a certain length of time.

The establishment of such standard figures for air leakage would be of assistance in the proportioning of surface condensers, and also would give a better basis for comparison of the efficiency of different types of condensers.

It would be interesting to hear from the author if he has any data referring to the maximum and minimum amounts of air leakage in pounds per hour for different sizes of surface condensers, such as those with 5000, 10,000, 25,000 and 50,000 sq. ft. of cooling surface.

The discusser would suggest that to avoid misunderstandings the symbols used in making condenser calculations should be agreed upon or determined by a standard committee of members of the Society engaged in condenser work.

The Author. I am greatly pleased with the form which the discussion of the paper has taken. Professor Ennis has amplified the suggested method given in the paper by introducing factors which are usually slurred over. In the actual tests N was always known by the weight of circulating water times its temperature rise, and the steam was rarely wet, as it nearly always is in an actual condenser.

Mr. Stuart's discussion emphasizes some points which perhaps have not been brought out in sufficient detail. Mr. Kothny is referred to my paper on Air in Surface Condensation (Trans.Am.Soc.M.E., Vol. 34, p. 713) for data on air leakage, and I wish to reiterate my statement that eternal vigilance is the price of freedom from air leakage. From 1 to 3 cu. ft. of free air per min. is very good practice with any commercial condenser, and this result can be obtained with condensers of the largest sizes.

Mr. Gibson mistakes K for U, and U is the coefficient of heat transmission in the fundamental formula $N = SU\theta$, while K is the reduced coefficient as given in Par. 10 $\left(K = \frac{U\theta^{\frac{1}{8}}}{V_w^{0.6}}\right)$ with a further reduction for air richness.

A reference to Trans.Am.Soc.M.E., Vol. 32, p. 1210, where the "three-dimensional" diagrams are shown, should make this distinction clear as well as Professor Ennis' question about the exponent ($\frac{1}{8}$).

Mr. McBride brought up the very interesting point that frequently it is necessary to assume primarily the tube length and diameter in some commercial designs. I have met this case in my own practice a number of times, but have always solved it by trial and error instead of using the arithmetical mean.

Mr. Bancel has again brought up the question of the depth of the tube bank, and incidentally the velocity of the steam among the tubes. This is a very complicated subject, being mixed up with the prevalence of the water film on the steam side of the tubes. There has been some little mathematical work along these lines, but in our experiments the results were negative, as was also the case with the observed results on commercial condensers. At the present time, however, there seems to be no reason to doubt the statements regarding this subject in Trans.Am.Soc.M.E., Vol. 32, p. 1160. Since writing the paper John E. Bell has read a paper on this subject before the Engineers' Society of Western Pennsylvania, in which he shows that a definite relation exists between the transfer rate and the cube root of the water condensed per hour per square foot divided by a function of the film temperature. More work is being done along these lines, but all I can say at present is that it is probable that we may be able to show a physical basis for the exponential formula of temperature rise.

No. 1545

THE TESTING OF HOUSE-HEATING BOILERS

By L. P. Breckenridge, New Haven, Conn.
Member of the Society

and

D. B. Prentice, New Haven, Conn.
Junior Member of the Society

During the last twelve years one of the authors of this paper has directed the testing of many house-heating boilers. For the most part these tests have been made for, or coöperating with, several organizations, as follows:

a The Bureau of Mines, at St. Louis, Norfolk and Pittsburgh.
b The Engineering Experiment Station of the University of Illinois, Urbana, Illinois.
c The Research Laboratory of the H. B. Smith Company, Westfield, Massachusetts.
d The Mason Laboratory of Mechanical Engineering of the Sheffield Scientific School of Yale University, New Haven, Connecticut.

2 These tests, numbering possibly four hundred, have included boilers of different types designed for anthracite coal, bituminous coal and coke; the coke-burning boiler being imported from Germany. A large number of briquets have also been tested.

3 The house-heating boiler has been rapidly developed into a very economical fuel-burning device, and efficiencies between 60 per cent and 70 per cent are very common.

4 The authors desire to present to this Society the results of some of these tests, but before doing so it has seemed desirable to propose for discussion, in the paper here presented, a standard

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

method for testing boilers of this type, as well as a consideration of some of the units which should be adopted in reporting tests of house-heating boilers.

5 The very complete facilities for testing house-heating boilers which have been installed in the Mason Laboratory of Mechanical Engineering have been rendered most effective in operation by the careful and ingenious planning of Professor E. H. Lockwood, whose attention to small details has made it possible to secure almost complete automatic records of the results of the tests. To him the authors extend acknowledgment for many helpful suggestions.

6 With these preliminary words of introduction and explanation we will proceed with the paper itself.

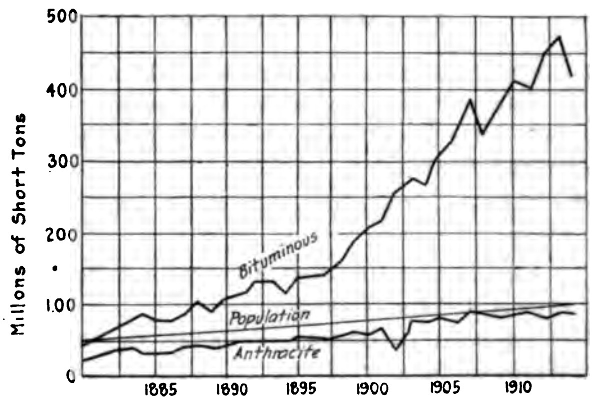


FIG. 1 COAL PRODUCTION OF THE UNITED STATES

OBJECT OF PAPER

7 The Power Test Committee of The American Society of Mechanical Engineers has formulated codes carefully setting forth standard methods of testing many types of engineering apparatus in common use. The methods are "standard" in the sense that they represent the opinions of prominent engineers as to the best practice of to-day, and are recognized by the Society as the proper form in which to present data for its records.

8 This paper proposes to discuss a standard method for testing house-heating boilers, a type of apparatus in very common use, but which, so far, has not received the attention of the Power Test Committee. If all boilers are classified as either power or heating, we define a house-heating boiler as one of the latter group designed

to serve 2000 ft. of radiation or less. This is equivalent to a boiler of less than 14.5 h.p. Such a boiler is usually operated with little attention, infrequent firings, low rate of combustion, and automatic pressure regulation by damper control.

9 The satisfactory testing of house-heating boilers by a method which admits of fair comparison is important for three reasons:

a On account of the increasing cost of anthracite coal it is desirable to determine efficient types of heaters and the most efficient method of burning this fuel.

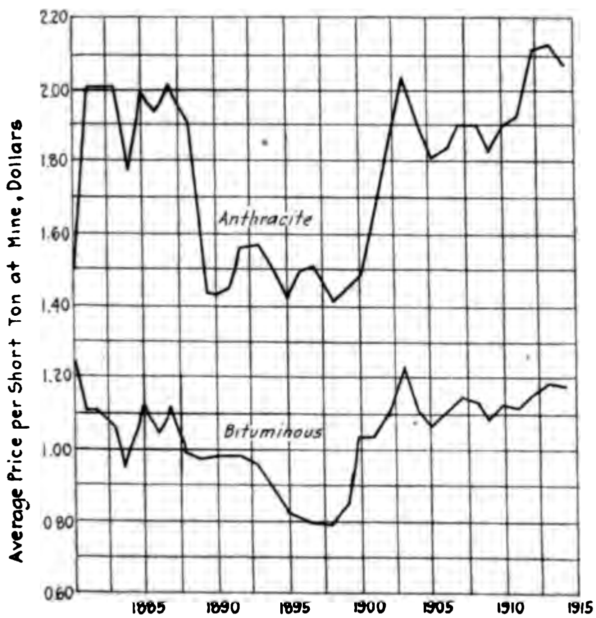


Fig. 2 Coal Prices at the Mines, United States

b It is desirable to develop efficient types of furnaces for other fuels, such as bituminous coal and coke.

c By means of standard tests it is desirable to establish a satisfactory rating system which will eliminate the uncertainty of the present manufacturers' ratings of house-heating boilers.

10 The annual coal production of the United States over a period of thirty years is presented graphically on the accompanying diagram, Fig. 1. With increasing population it is evident that the

price of anthracite coal must rise, even disregarding the greater cost of mining. The mine prices of the two kinds of coal are given in Fig. 2.

11 The commercial ratings of house-heating boilers as given by the manufacturers are seldom comparable and are often optimistic. This is suggested by the following quotation from a standard professional handbook, The Architects' and Builders' Pocket-Book: "These ratings (house-boilers) are commonly made pretty high, so that it is a safe rule to use a boiler having a rating of 40 per cent in excess of the actual direct radiation when the mains are covered, and 50 per cent when they are not covered."

REQUIREMENTS IN TESTING HOUSE-HEATING BOILERS

12 The authors of the present paper believe that a method of testing any apparatus to give results of value should meet two requirements:

a The apparatus should be tested with as many conditions as possible equivalent to those under which it normally operates (except for particular investigations under special conditions).

b Errors should be so far eliminated or reduced that results can be reproduced.

13 To meet the first requirement with a house-heating boiler it is necessary to operate at a moderate, reasonable pressure, controlled automatically, as would be the condition in an actual installation. It is proper to fire the fuel in rather heavy charges and to give attention to the boiler only at intervals of several hours. The fire should not be frequently shaken or poked. The temperature of the feed water should be comparable to the returns from a radiator system, and the boiler should be supplied with water continuously and uniformly. Perhaps the greatest essential is that the test should be started under average running conditions, with the boiler and covering thoroughly heated to bring the radiation loss to equilibrium, and with a fire neither freshly kindled on bare grates nor clogged with ash; in brief, a good, thin, recently cleaned fire. A method of starting which the authors have found very satisfactory and which secures nearly uniform conditions at the beginning and end of a test is described in Par. 17.

14 The elimination or modification of errors so that results can be verified by several tests presents some difficulties. The measure-

ment of coal fired in the boiler is a matter of careful weighing. Determining how much coal has been *burned* is necessarily somewhat uncertain. The method of starting and stopping described in Par. 17, the authors believe, decreases the error of measurement to a

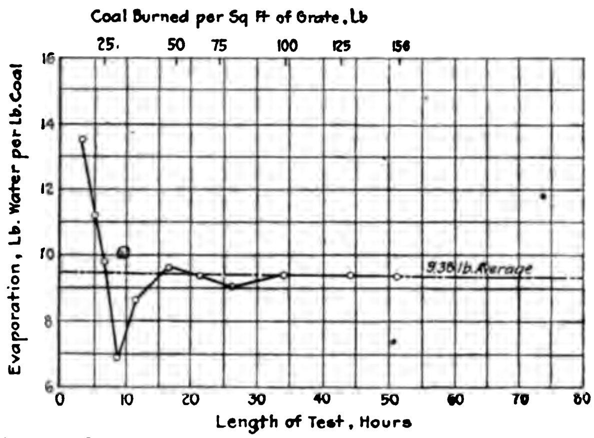


FIG. 3 SPECIAL TEST TO DETERMINE PROPER LENGTH OF HEATING-BOILER TESTS

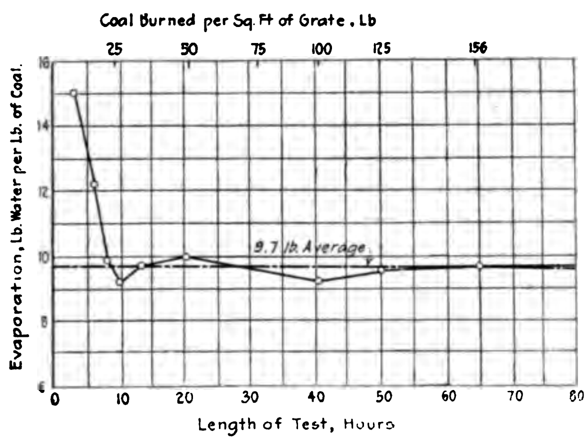


FIG. 4 SPECIAL TEST TO DETERMINE PROPER LENGTH OF HEATING-BOILER TESTS

minimum, yet makes the test under normal running conditions. The low rate of combustion in house boilers and the small amounts of fuel involved make long tests essential. It is necessary to burn at least 40 lb. of coal per sq. ft. of grate area, or to test for at least 12 hours, to secure results that can be approximately duplicated.

The error in coal measurement is then distributed over an amount large enough to make it a small percentage of the total. That shorter tests are unreliable is shown by the accompanying diagram of two special tests, Figs. 3 and 4. In these tests the starting conditions were carefully reproduced several times. From the start to any one of these endings may be considered a complete test. The evaporation per pound of coal has been calculated for the several periods from the start to the successive closing points, and these performance values are plotted against the elapsed time, that is the durations of the corresponding periods. The heavy horizontal line represents the average evaporation for the longest period. The vertical distance of any point from this line is the error in the calculated rate of evaporation which would result from closing the test at that time. Very little gain in accuracy is secured by tests longer than 20 hours.

CONTROL OF BOILER OUTPUT

15 The output of a boiler during a test may be controlled in one of several ways or it may be allowed to vary. The latter condition reproduces, strictly, house operation, where the heating load is uneven, depending on wind and weather. But testing on such a plan would be very unsatisfactory; for the duplication of a test would be a matter of chance, and innumerable experiments might be necessary to produce a performance curve for a fair range of output. Some control is desirable, preferably by a constant and accurate method. Condensation in radiators or piping, the amount of surface that is open to steam being regulated, is too variable. An orifice of some kind is a better control. However, the pressure in the boiler is not even, and the discharge of steam directly through a certain opening would not be uniform. An excellent arrangement seems to be to maintain a pressure of 2 lb. per sq. in. in a receiver by an automatic pressure-reducing valve, the boiler carrying 3 to 8 lb. The discharge from the receiver through a given area is then constant. The authors believe that a tapering needle valve is more satisfactory than a set of changeable orifices, as it facilitates varying the load during operation. A bank of valves of varying size, opening from the receiver, is another convenient and suitable method of control. The exhaust steam may be used to warm the feed water or it may be condensed and used again. The first scheme, of course, requires less water.

16 The method of feeding the water to the boiler is not important provided a steady, easily controlled flow is secured. The

measurement of boiler output may be made either before or after the water goes to the boiler. An extremely convenient method is to draw the water supply from a tank fitted with a long gage glass and scale. The level is quickly noted and no "weighing back" is necessary. The tank should be carefully calibrated, by weighing, over the entire range between the extreme levels reached. A duplex water-driven pump is a very convenient device for feeding the boiler wherever city water pressure is available.

AUTHORS' METHOD OF STARTING TESTS OF HOUSE-HEATING BOILERS

17 The authors have found the following method for starting house-heating boiler tests very satisfactory. A preliminary fire is carried until the boiler is well heated and operating at the usual pressure, about 5 lb., and the load at which the test is to be made. When conditions are considered stable the preliminary fire is allowed to burn down, the pressure begins to fall, and the fire is shaken, which retards the pressure drop temporarily; but soon a pronounced fall is noticed. The fire is now thin, clean, burned out, but still hot enough to kindle fresh fuel. When the pressure reaches a given point, perhaps 3 lb., the necessary measurements are made, including a careful determination of the thickness of the fuel bed, the first charge of weighed coal is fired, and the test is considered started. When the test is drawing to a close these same conditions are readily duplicated, and at a point, after shaking, when the pressure falls to 3 lb., the test is concluded. The ash should be shaken out until the thickness of the fuel bed is the same as at the start. The events outlined are characteristic, and a recording pressure gage pictures them very clearly. The accompanying diagram, Fig. 5, was copied from a pressure chart made during an actual test. The periods of falling pressure immediately before the beginning and ending of the test, as well as before each firing, are quite distinct.

18 Before presenting a proposed "standard method" for testing house-heating boilers the authors wish to discuss the customary unit employed for stating the load or capacity of this type of apparatus. Such a unit has never been defined exactly by either the American Society of Heating and Ventilating Engineers, which would naturally be most interested, or by The American Society of Mechanical Engineers. The former organization, however, has had the subject under consideration at various times.

PROPOSED UNIT OF CAPACITY FOR RADIATORS AND HEATING BOILERS

19 The capacity or commercial rating of a heating boiler h always been given in terms of the direct radiating surface which would serve. The unit usually employed is the "foot of radiation commonly understood to be that surface which will condense quarter of a pound of steam per hour at 2 lb. pressure when in a

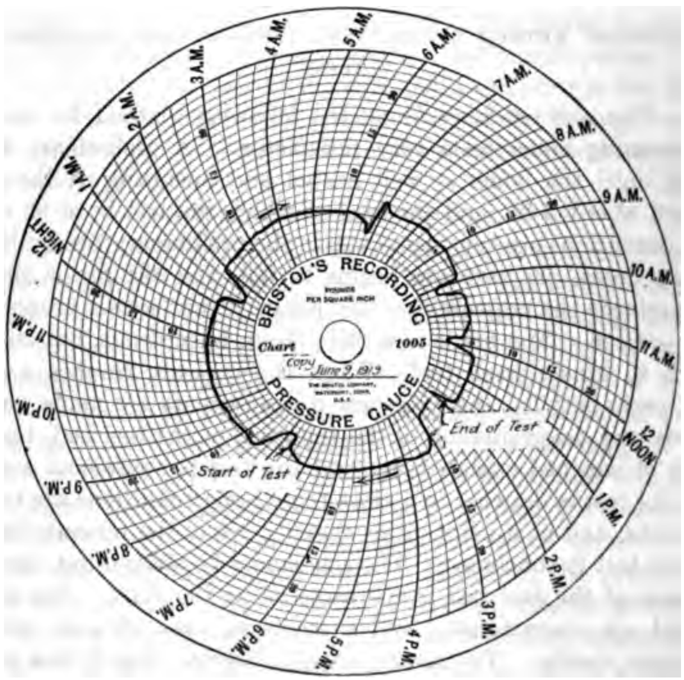


FIG. 5 PRESSURE CHART MADE DURING AN ACTUAL TEST OF A HOUSE-HEATING BOILER

air at 70 deg. fahr. Originally the foot of radiation meant a squ foot of radiating surface, 144 sq. in., but improvements in des and arrangement enabled manufacturers to secure this condens tion with less surface, and consequently less weight of iron. T result has been a variable and decreasing value in square inches the "foot of radiation." In fact, the unit has become, as it shou dependent entirely on condensation of steam, which means a h transfer, rather than on any particular area of metal.

20 If the unit is considered as a certain number of heat units it is applicable to boilers directly, as well as to radiation. There are many reasons why it would be convenient to have a capacity unit for heating boilers comparable to the capacity unit for power boilers, *i.e.*, the boiler horsepower. The authors propose, therefore, the following definition of a unit for stating the capacity of radiators and heating boilers:

> The "foot of radiation" shall be a quarter of a pound of steam condensed from and at 212 deg fahr. per hour.

This is equivalent to a heat transfer of 242.6 B.t.u. The condensation of a quarter of a pound of steam at 2 lb. pressure is equivalent to 241.5 B.t.u. There is, therefore, very little change in the heat equivalent of the unit. Although there is no direct connection between the unit as defined above and a square foot or any particular area of radiating surface, yet the authors favor the retention of the old name for the same reason that boiler horsepower was continued although that unit bears no relation to the standard horsepower.

21 The advantages of the above definition of the "foot of radiation" are:

- *a* The heating- and power-boiler capacity units are based on the same physical conditions.
- *b* They are mutually convertible by the factor 138.
- *c* Radiation is rated without reference to surface, so that efficient designs and arrangements are benefited.
- *d* The unit is applicable to hot-water radiators and heaters by determining the heat transfer. (The result may be called "equivalent feet of radiation.")

22 The authors recommend that radiator tests to determine rating be made with steam at 2 lb. pressure and in "still air" at 70 deg. fahr., and suggest that boilers under test be operated at from 3 to 8 lb. pressure.

PROPOSED HOUSE-HEATING BOILER TEST CODE

23 The following paragraphs are proposed as an Individual Code for house-heating boilers, to be used in conjunction with the First Section (General Matters) of the Report of the Power Test Committee of the Society.[1]

[1] Trans. Am.Soc.M.E., vol. 37, pp. 1273–1458.

RULES FOR CONDUCTING EVAPORATIVE TESTS OF HOUSE-HEATING BOILERS

OBJECT AND PREPARATIONS

[1] Determine the object, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions, Part I, and make preparations for the test accordingly.

FUEL

[2] Determine the character of fuel to be used. For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the fuel should be some kind which is commercially regarded as a standard for the locality where the test is made.

[3] A fuel selected for a maximum efficiency or capacity test should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

[4] For guarantee and other tests with a specified fuel containing not more than a certain amount of ash and moisture, the fuel selected should not be higher in ash and moisture than the stated amounts, because any increase is liable to reduce the efficiency and capacity more than the equivalent proportion of such increase.

[5] 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.

[6] The size of the fuel, especially where it is of the anthracite class, should be determined by screening a suitable sample.

APPARATUS AND INSTRUMENTS

[7] The apparatus and instruments required for a heating-boiler test are:

(a) Platform scales for weighing coal and ashes.
(b) Graduated scales attached to the water glasses.
(c) Tanks with full-length gage glasses and graduated scales for measuring water.
(d) Pressure gages, thermometers, and draft gages.
(e) Calorimeters for determining the calorific value of fuel and quality of steam.
(f) Gas-analyzing apparatus.

[8] Full directions regarding the use and calibration of the above-mentioned appliances are given under the heading Apparatus and Instruments, Part I, ¶ 7 to 9. For particulars regarding the best location of the various instruments and apparatus, see Appendix No. 24.

OPERATING CONDITIONS

[9] Determine what the operating conditions and method of firing should be to conform to the object in view, and see that they prevail throughout the trial, as nearly as possible. Determine the amount of fuel for each charge and verify the output regulation so that the desired capacity is maintained. During the preliminary run set and test the damper regulator and the pressure-reducing valve for the desired pressures.

DURATION

[10] The duration of tests to determine the efficiency of a house-heating boiler should be at least 12 hours of continuous running, or such time as may be required to burn a total of 40 lb. of fuel per square foot of grate. The duration of tests which include periods of banked fires 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 3 hours.

STARTING AND STOPPING

[11] The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live fuel 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.

[12] To secure the desired equality of conditions the following method should be employed:

The furnace should be well heated by a preliminary run of at least an hour under all the conditions at which it is proposed to test. A suitable pressure for running the boiler while under test is from 5 to 8 lb. During this run the operation of the pressure-reducing valve and damper control should be tested, and the setting of the output-control valve should be verified. The fire should now be allowed to burn low, which will result in a falling pressure. When the pressure reaches a chosen point (3 lb. is recommended), the fire should be shaken clear of ash, the thickness of fuel bed noted, the water levels in boiler and feed tank recorded, together with steam pressure, draft, flue temperature and the other usual data. The first charge of weighed fuel should now be fired and the time recorded as the starting time for the test. The test should be ended when the fire has burned low and it is impossible to check the pressure fall by shaking, as at the start. When the pressure again reaches the chosen point the final readings should be made, the ash shaken out until the fire is the same thickness as at the start, and the time recorded as the stopping time. It is advisable to regulate the feed so that the water in the boiler is brought to the starting level and held there, towards the end of the test. All ash should be removed from the ashpit immediately after the fire is shaken down at the beginning of the test and again at the end. The second lot, with any that may have been drawn during the test, should be carefully weighed.

RECORDS

[13] The records of data should be obtained as pointed out in the General Instructions. 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. The fuel should be weighed in amounts equal to the desired firing charge.

QUALITY OF THE STEAM

[14] If the boiler delivers wet steam the percentage of moisture in the steam should be determined by the use of a separating calorimeter. If the boiler delivers superheated steam the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well.

SAMPLING THE FUEL

[15] During the progress of the test the fuel should be regularly sampled for the purpose of analysis and determination of moisture.

ASHES AND REFUSE

[16] 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.

CALORIFIC TESTS AND ANALYSIS OF FUEL

[17] The quality of the fuel should be determined by calorific tests and analyses of the 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.

ANALYSIS OF FLUE GASES

[18] For approximate determinations of the composition of the flue gases, the Orsat apparatus, or some modification thereof, should be employed.

SMOKE OBSERVATIONS

[19] In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made at from 5- to 15-second intervals during the smoking period, noting at the same time the furnace and firing conditions.

CALCULATION OF RESULTS

[20] The method to be followed in expressing and calculating those results which are not self-evident or stated in detail in the Boiler Code, is as follows:

The capacity of the boiler in "feet of radiation" served is found by dividing by 242.6 the total heat absorbed per hour by the water in the boiler, expressed in British thermal units.

DATA AND RESULTS

[21] The data and results should be reported in accordance with the House-Heating Boiler Test Form (Table 1) given below, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

CHART

[22] 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.

REPORT

TABLE 1 DATA AND RESULTS OF EVAPORATIVE TEST OF HOUSE-HEATING BOILERS

(1) Test of...............heating boiler located at...........................
to determine.........., conducted by.................................
(2) Kind of furnace...
(3) Grate surface: Width........length........Area.....................sq. ft.
(4) Water-heating surface: Direct........, indirect........, total..........sq. ft.
(5) Superheating surface..sq. ft.
(6) Date..............................Test No.........................
(7) Duration...hr.
(8) Kind and size of fuel..

AVERAGE PRESSURES, TEMPERATURES, ETC.

(9) Steam pressure in boiler by gage......................................lb.
(10) Steam pressure in receiver by gage....................................lb.
(11) Temperature of feed water entering boiler..............................deg.
(12) Temperature of escaping gases leaving boiler...........................deg.
(13) Force of draft between damper and boiler...............................in.
(14) Percentage of moisture in steam, or number of degrees of superheating..per cent or deg.
(15) Percentage of CO_2 in escaping gases.............................per cent

TOTAL QUANTITIES

(16) Weight of fuel as fired...lb.
(17) Percentage of moisture in fuel....................................per cent
(18) Weight of dry fuel consumed...lb.
(19) Weight of ash and refuse..lb.
(20) Percentage of ash and refuse in dry fuel..........................per cent
(21) Weight of water fed to the boiler.....................................lb.
(22) Weight of water evaporated, corrected for moisture in steam...............lb.
(23) Equivalent evaporation from and at 212 deg............................lb.

Hourly Quantities and Rates

(24) Dry fuel consumed per hour.......................................lb.
(25) Dry fuel consumed per sq. ft. of grate surface per hour....................lb.
(26) Water evaporated at 16.7 lb. absolute pressure per hour..................lb.
(27) Equivalent evaporation per hour from and at 212 deg.....................lb.
(28) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water-heating surface.......................................lb.

Capacity

(29) Evaporation per hour from and at 212 deg...............................lb.
(30) Feet of radiation served (1 ft. radiation = 242.6 B.t.u.)................ft. rad.
(31) Rated capacity in feet of radiation.......................................ft. rad.
(32) Percentage of rated capacity developed.............................per cent

Economy Results

(33) Water fed per lb. of fuel fired.......................................lb.
(34) Water evaporated per lb. of dry fuel..................................lb.
(35) Equivalent evaporation from and at 212 deg. per lb. of dry fuel............lb.
(36) Equivalent evaporation from and at 212 deg. per lb. of combustible.........lb.
(37) Fuel, as fired, per hour per 1000 ft. of radiation..........................lb.
(38) Dry fuel per hour per 1000 ft. of radiation..............................

Proximate Analysis of Fuel

(39) Moisture..per cent
(40) Volatile matter..per cent
(41) Fixed carbon..per cent
(42) Ash...per cent

100.00 per cent

Efficiency

(43) Calorific value of 1 lb. of dry fuel...................................B.t.u.
(44) Calorific value of 1 lb. of combustible...............................B.t.u.
(45) Efficiency of boiler, furnace, and grate.............................per cent
(46) Efficiency of boiler and furnace......................................per cent

Attendance

(47) Weight of fuel fired per charge.......................................lb.
(48) Weight of fuel fired per charge per sq. ft. of grate surface.................lb.
(49) Average interval between firings.......................................hr.
(50) Minimum period of operation without attention..........................hr.
(51) Maximum period of operation without attention..........................hr.
(52) Nature of attention..
(53) Character of fuel as regards handling, etc..............................
(54) Character of ash as regards handling, etc...............................
(55) Smoke observations..
(56) Soot deposits..

Cost of Evaporation

(57) Cost of fuel per ton of ....lb. delivered in boiler room.................dollars
(58) Cost of fuel required for serving 1000 ft. of radiation one hour..........dollars
(59) Cost of fuel required for evaporating 1000 lb. of water from and at 212 deg..dollars
(60) Notes:

DISCUSSION

GEO. H. BARRUS (written). In laying this matter before the Society, Professor Breckenridge directs attention not only to the particular subject presented, but to the Code of Rules already accepted which forms the basis of his paper, and incidentally to the entire report of the Power Test Committee, dealing as it does with matters of such widespread interest.

I think it may be taken for granted that the Committee will welcome any suggestions for amendment to the report such as Professor Breckenridge brings forward, and that they will be given thorough consideration. Furthermore, I hope that his action will lead other members of the Society to study the Codes as they now stand, and if any Code is found incomplete, or unsuitable, the fact may be made known to the Committee. I am sure that the Society is desirous that the members interest themselves in these Codes, to the end that so far as possible deficiencies may be pointed out and the Codes made most acceptable as standards.

C. M. GARLAND (written). This type of boiler and furnace has been notoriously uneconomical in the appropriation of the heat liberated by the fuel, and it is largely due to Professor Breckenridge's initiative along these lines that the inefficiencies of these boilers have been brought to the attention of the manufacturers, engineers, and to the government experiment station. The result is that manufacturers have given the matter of efficiency serious thought, and house-heating boilers of excellent efficiency are now built.

The testing of such boilers is radically different from the testing of the high-pressure boilers as used for power and larger heating work. It is therefore desirable that a different code should be provided for the testing of these boilers. The one proposed by the authors would seem to meet the requirements of the different conditions. A heat-value unit for a standard measure for one foot of radiation would be a most desirable standard.

ROY E. LYND (written). The titles of the paper and of the proposed testing code both confine themselves to house-heating boilers, and the paper states that the class of boilers indicated by the authors under this heading includes only boilers designed to serve 2000 ft. of radiation or less. It seems that we make a mistake in thus limiting this code. The same boilers which we use in our houses are used very extensively to heat schools, churches, and other large buildings, and several makes of low-pressure cast-iron sectional boilers are designed to serve as much as 10,000 ft. of radiation. We would do well to eliminate the term *house-heating boilers* from the title, the paper and the code, and substitute therefor *low-pressure heating boilers;* and include all low-pressure heating boilers instead of those only which are designed to serve 2000 ft. of radiation or less. The larger boilers of this class are covered by Par. [9] of the code, which states that the test conditions should be as nearly as possible like the ordinary operating conditions for the boiler to be tested.

In the A.S.M.E. Boiler Code of 1914, boilers are divided into two classes, — Power Boilers, Section 1, and Boilers used exclusively for Low-Pressure Steam and Hot-Water Heating and Hot-Water Supply, Section 2. This division should be borne in mind in any new testing code. As the proposed testing code is essentially a code for evaporative tests, we are not concerned with boilers for hot-water heating and hot-water supply. It would seem therefore that the new code should cover all boilers used exclusively for low-pressure steam heating, and should be so entitled. The A.S.M.E. Boiler Code, in Section 2, does not limit low-pressure heating boilers to 2000 ft. or less, and we should not so restrict the testing code.

The second point is in regard to the definition given for a *foot of radiation.* The authors seem to think that the amount of steam condensed per foot of radiation enters more largely into ordinary heating calculations than the B.t.u. They have assumed a convenient average amount of steam per foot of radiation, and have then converted this into an awkward B.t.u. value. This, to my mind, is wrong. We figure practically everything in connection with heating installations in B.t.u., and it is very rare that the question of the amount of steam involved is raised. I would suggest that the *foot of radiation* be defined as a transfer of heat equal to 250 B.t.u. per hour. This figure, and its reciprocal, 0.004, are both very convenient, and would be far preferable to the figures given by the authors.

It has been my practice to test low-pressure boilers at atmospheric pressure, keeping a record of the steam temperature as indicated by a mercury thermometer placed in an oil well directly in the steam chamber in the top of the boiler. The pressures at which these boilers are operated are as a rule so nearly atmospheric, if the heating system is conservatively designed, that a test made at atmospheric pressure comes about as close to actual operating conditions as it can be got. The great advantage of the atmospheric-pressure test is, of course, its simplicity, it being unnecessary to use the reducing valve, receiver, and bank of valves spoken of by the authors.

One of the functions performed by this system of pressure control suggested by the authors is in the determination of the time of starting and stopping the test. The test is started by establishing normal running conditions with a pressure of, say, 5 lb. on the boiler. Then the fire is cleaned and thinned until the pressure drops to, say, 3 lb., when the test is assumed to start. The same conditions are reproduced at the end of the test, the test being over when the pressure drops to the same 3 lb. This would all be out of the question with a test made at atmospheric pressure. I have employed for some time a system which is very similar, and which gives practically the same accuracy, and which is applicable to tests made at atmospheric or any higher pressure. Normal running conditions are established before the test, and then the fire is cleaned and thinned just as outlined in the paper, but instead of depending on the pressure dropping to a certain starting pressure, the temperature of the flue gases is used as an index. When the temperature of the flue gases falls to a predetermined point, the test is assumed to be started, and at the close of the test the starting conditions are reproduced until the flue-gas temperature taken at the same point in the flue falls to the starting temperature. This method seems preferable, as the flue-gas temperature is more intimately connected with the condition of the fire than the steam pressure is, and at the same time it enables us to use the very much simpler atmospheric-pressure conditions.

It has also been my practice, for some time past, to keep accurate records of the draft in the flue, in the firebox, and in the ashpit, by means of differential draft gages. These data sometimes indicate differences in the draft conditions which may explain differences in test results.

C. B. Thompson[1] (written). Referring to Par. 9c, I cannot find in this paper any proposed method of rating boilers that will be any improvement over existing conditions; in fact, there is no hint of what a given boiler's rating should be.

If I have read this paper understandingly, the argument is that boilers are overrated because there has been no well-established or standard testing code, and hence errors in tests have led to errors in computing working capacities. With this proposition I cannot

[1] Thompson Heater Corporation, Buffalo, N. Y.

agree. The code presented by Messrs. Breckenridge and Prentice is only one of many by which boilers may be tested that will show practically the same results. It is when the tests are computed to determine the boiler's capacity in terms of square feet of radiation that guesswork is employed. The computer can give the capacity at which the boiler will operate for any unit of time if he knows the fuel capacity. Without this all-important item he is adrift in a fog.

The most important items omitted in the proposed code are the fuel capacity, the recoaling reserve, and the fuel available. Thus: Fuel capacity, 300 lb; reserve, 80 lb; fuel available, 220 lb. The latter will vary for different types of boilers and for the various kinds of fuel from 60 to 80 per cent of the full fuel capacity.

Since the heat energy is derived from the fuel burned through the medium of the boiler, and since the quantity burned per hour determines the boiler's capacity, is it not essential that the fuel available plus the reserve for igniting the recoaling charge be clearly and accurately stated? Without this information how is it possible for one to determine the necessary firing interval for any given capacity?

If the available fuel is 200 lb. and 25 lb. per hour is burned, the capacity, using the unit of 0.25, is about 875 sq. ft. and the coal will last for eight hours; chimney 8 in. × 12 in. × 35 ft. Twenty pounds per hour gives 700 sq. ft. for 10 hours; 15 lb. 525 sq. ft. for 13.3 hours, and 10 lb. 350 sq. ft. for 20 hours, chimney 8 in. × 8 in. × 30 ft. The efficiencies for these varying rates in properly proportioned boilers will vary so little that they may be disregarded.

If the capacity of house-heating boilers were computed something like the following there would be less complaint of overrated boilers, because many buyers do not understand that the rated capacity is at the boiler outlet:

	Sq. Ft.
Capacity cast-iron radiation	800
Add for covered pipe	150
Total boiler load	950
Hours coal will last	9

If the chimney size required be then given, the purchaser can select what he wants without outside assistance.

In regard to the testing code suggested, I do not like any small boiler tests in which the weight of the fuel at start and finish is determined by measurement. Where this method is employed it is only by accident that errors in the weight of the fuel are avoided.

If the boiler is covered, as it will be in actual use, and is fired with wood until thoroughly heated and the water in the boiler at the boiling point, the unburned wood may then be withdrawn and a weighed quantity of kindling wood and coal used to start the test.

At the close of the test, the unburned fuel remaining in the firepot may be quenched with a weighed quantity of water — just sufficient to kill the combustion. The unburned coal may then be carefully freed from ash and weighed. The water used for quenching the fire should be added to the water evaporated in the usual manner. The fire may be quenched by turning water into the firepot without danger of breaking the castings. A boiler's capacity and efficiency at any point in the test may be determined by setting the boiler on a scale which shows the combustible burned per unit of time. The chemical ash may be employed to show the coal consumed.

Many tests have been made employing various methods of stopping and starting, at different steam pressures, and with feedwater at temperatures varying from 40 to 200 deg. fahr.; the duration of these tests ranged from four hours to 21 days.

As a result of these tests I am convinced that a test started and stopped about as described above will show the boiler's capacity and efficiency as accurately as will tests that entail much more time and expense. It is understood that boiler efficiency in its narrow sense means the ratio of output to input.

There are other efficiencies or deficiencies of more vital importance to the user of boilers than is the "ratio of output to input," but these are seldom or never heard of until the boiler is in actual use. Of the useful heat produced per pound of fuel the owner knows nothing, but he will object strenuously if the draft control is inadequate and the boiler burns fuel far in excess of requirements. If the grate is poorly fitted and falls into the ashpit when it is shaken, if the doors warp and stick so they cannot easily be opened or closed, if the water leaves the glass with a rising fire, or if the boiler makes a noise at certain temperatures, the owner will have a poor opinion of the installation, though the boiler may be capable of a laboratory performance showing 75 or 80 per cent efficiency.

The tests should be run at rates that will burn the available fuel in periods of 6, 8, 10, 12 and 16 or 20 hours, and the efficiency for each rate of combustion noted.

It should also be noted for the benefit of the purchaser of the boiler, at what rate of combustion the ashes fuse and form slag and clinker, and at what lower rate this objectionable feature disap-

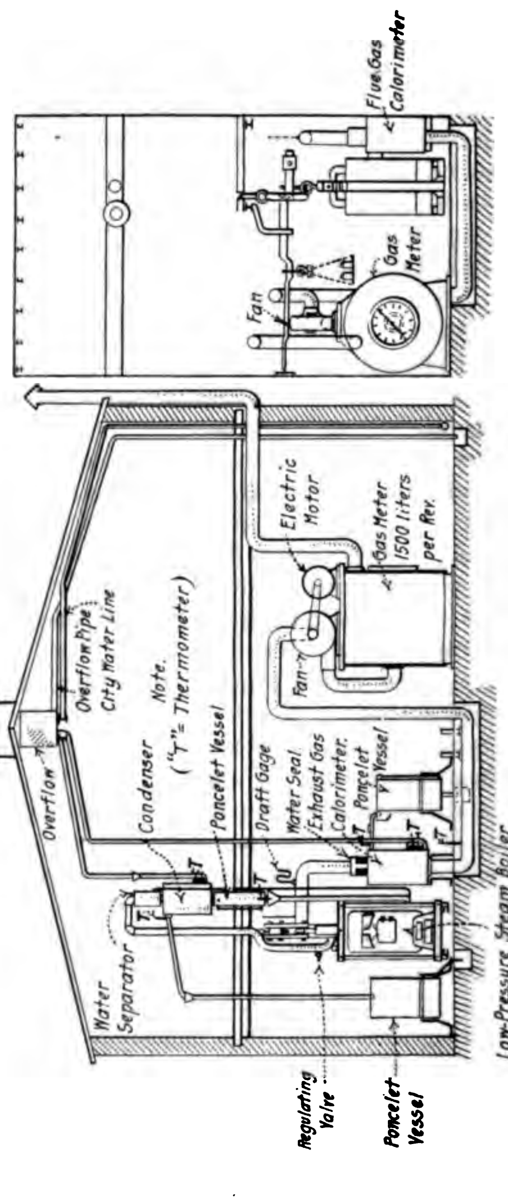


FIG. 6 TESTING LOW-PRESSURE BOILERS AT THE ROYAL TECHNICAL COLLEGE AIX-LA-CHAPELLE, GERMANY

pears. The fine coal known as buckwheat can only be successfully burned at very low rates, otherwise the excessive ash content forms quantities of clinker.

For each rate of combustion the area and height of chimney should be given, this item being of vital importance to the purchaser.

The heat energy per hour should be shown in terms of B.t.u., which may be divided by 242 for a load condensing at the quarter-pound rate (0.25 lb. per sq. ft. per hour). For a boiler load consisting of radiation 75 per cent and covered piping 25 per cent, the divisor will be about 205; for factory work it will be 300 and for greenhouse heating probably 350.

I do not see the necessity for preheating the feedwater. It is so easy to calculate the temperature rise of feedwater that its heating outside the boiler seems to be an unnecessary detail and expense, while certainly nothing is gained in accuracy. If very close accuracy in this item is an essential, we must abandon the "from and at 212" and figure the condensate entering the boiler at a temperature considerably below 212 deg. Probably 180 deg. more nearly represents the actual temperature of the condensate entering the boiler in actual house-heating practice at 2 lb. pressure.

In so-called "vapor" heating where the initial pressure is from 1 to 3 oz., the condensate will enter the boiler at or about 110 deg. fahr. This type of apparatus, which is an open-circuit system without air valves, is fast displacing the 2-lb. to 5-lb. closed-circuit systems, and will have to be considered in future boiler and radiator tests.

Referring to Figs. 3 and 4, the results shown therein are so unusual that I would like to ask the authors if they can explain why the evaporation in a 3-hour run is 13.5 lb. and in an 8-hour run less than 7 lb. Is this variation due to errors in calculating by measurement the weight of coal burned?

In measuring the fuel bed there is always the possibility of including ash in a fused state, which in that condition cannot be distinguished from the glowing fuel.

Max Friedlander (written). A new method for the *continuous* determination of the heat balance of house-heating boilers, suggested by Prof. H. Junkers, the originator of the Junkers calorimeter, was tried out and applied by the writer in a series of actual tests on a steam-heating boiler in 1911 when he was his assistant at the Technical College of Aix-la-Chapelle, Germany. The idea was to measure all items of a complete heat balance in a continuous way during operation, and for this purpose the boiler was suspended upon a sensitive balance so the smallest amount of fuel burned off in the boiler could be weighed very exactly at shortest intervals, thus giving a continuous determination of the fuel consumption and the incoming heat. The arrangement is shown in Fig. 6.

The entire steam generated was condensed in a condenser and the condensed water carried back to the boiler. In this way the useful heat could be determined continuously by continuously measuring with a Poncelet vessel the quantity of cooling water used in the condenser and, with the thermometers, the increase in its temperature.

The flue gases were drawn out by a ventilator and carried through a flue-gas calorimeter, in which their entire *sensible heat* was determined by cooling them down to the room temperature by a water jacket, the quantity of cooling water being measured continuously with a Poncelet vessel, and its rise in temperature also being measured. The volume of the flue gases was recorded with a gas meter of 1500 liters capacity per revolution.

A quite novel feature was the continuous determination of the loss of heat due to incomplete combustion by a new calorimetric method in which the heat value of the flue gases was measured in a calorimeter fitted with a specially designed burner for which a patent is pending. This method for the calorimetry of flue gases has been developed by the writer in separate experiments and tried out in a great number of actual tests and applications on boilers and combustion engines, and it has been described in detail in a dissertation (not yet published), where all these experiments and tests are also reported. The arrangement for this is also shown in the illustration.

The heat loss due to incomplete combustion was very variable and, with the boiler mentioned, wavered between 8 per cent and 23 per cent of the incoming heat when the operation and combustion were normal, and increased to over 45 per cent when the boiler was operated with insufficient excess of air or otherwise in bad condition. In all cases, however, the heat value of the flue gases decreased continuously, or, in other words, the combustion improved steadily in the proportion as the layer of coal was burning off, thus indicating that the boiler was working in the beginning like a gas producer.

Heat radiation and conduction were determined by temperature measurements of the outer surface of the boiler by use of individual coefficients of heat transfer.

ALLEN HUBBARD (written). If boiler manufacturers would rate their boilers on the basis outlined in the paper, the public and business generally would be greatly benefited. The important thing is to know what a given boiler will do under the usual handicapping conditions. It is safe to say that the flues of the average house-heating boiler are not cleaned oftener than once a month. It seems as though this handicap should be recognized in making the tests and rating the boilers.

WILLIAM KENT (written). The method of rating a house-heating boiler proposed by the authors seems to leave out a most important factor of such a rating, viz., the grate surface, or the amount of coal that should be burned per square foot of grate surface.

The authors say, "The capacity or commercial rating of a heating boiler has always been given in terms of the direct radiating surface which it would serve." The capacity of such a boiler thus defined, that is, the amount of radiating surface which it will serve, is an exceedingly variable quantity, depending chiefly upon the amount of coal that is burned under it per hour, which in turn depends on the size of the grate and the rate of combustion. A certain boiler with 1 sq. ft. of grate and say 20 sq. ft. of heating surface may supply 150, 300 or 450 sq. ft. of radiating surface, depending on whether the coal is burned at the rate of 4, 8, or 12 lb. per sq. ft. per hour. It is evident then that no satisfactory rating of a house-heating boiler can be made that does not take into consideration the rate at which the coal is burned. I therefore would amend the authors' definition of a unit for stating the capacity of a heating boiler so as to make it read as follows:

> The *foot of radiation* (more properly square foot of radiating surface) when used as a measure of the capacity of a house-heating boiler shall be $\frac{1}{4}$ lb. of steam per hour condensed at 212 deg. fahr. and discharged as water at 182 deg. (equivalent to 250 B.t.u. per hour) when the coal is burned at the rate of 4 lb. per sq. ft. of grate surface per hour.

I use the figure 250 instead of the authors' 242.6 because it has long been used by heating and ventilating engineers as a standard equivalent for an average square foot of radiation. A radiator generally discharges its return water at a temperature somewhat below the temperature of the steam, and the figure 250 is therefore more nearly equivalent to the actual conditions of condensi of steam than is 242.6.

In 1909 the writer presented a paper on The Testing and of House-Heating Boilers to the American Society of Heat Ventilating Engineers, which is published in the Transactions society. Some of the points in that paper are pertinent discussion of the present paper.

S. B. FLAGG and R. L. BEERS (written). The writers ha engaged during the past two years in planning and carryin extended series of tests which the Bureau of Mines is conduc one of the Government departments. The principal pu these tests has been to obtain information as to the relati for domestic heating purposes of a large number of fuels used department, including a number of Canadian and foreign co the same time a comparison is being made of steam and h boilers.

The reasons given in the paper why it is important to satisfactory method of testing house-heating boilers are indorsed. The writers would add to these reasons by poin that in many localities anthracite coal is practically unobtai can be used only at a much greater heating cost than for son fuel. Consequently methods of firing and testing such boil the ratings established for them should take account of the fuels, especially bituminous coal.

It is agreed that the lack of clearness as to the meaning *of radiation* is undesirable and should be corrected. In r results of tests of boilers of the hot-water type, the employ a unit such as the paper describes would be especially desir comparison with steam-boiler tests.

In the development of plans for the tests which the Burea conducting the following considerations governed:

The average residence-heating boiler operates during the the heating season at a load less than 40 per cent of its ratin sults were desired showing the comparative values of the fuel average load conditions, and the tests were therefore run proximately this load.

Conditions of attention were to be comparable to those i service so far as possible. For this reason with most fuels ch relatively large size were fired so as to give a firing period from 6 to 12 hours.

In the case of the steam boiler installed in a residence, neither the rate of delivery of steam nor that at which the condensation returns is uniform. The boiler output was therefore allowed to vary, but the valve controlling the delivery was so set that with automatic damper regulation an average load of approximately 40 per cent of rating was maintained. This corresponds with the authors' requirement in Par. 12*a*.

So far as possible the test data were mechanically recorded and some of the data so recorded by a second piece of equipment.

In order to reduce errors of starting and stopping the duration of the tests was made approximately 48 hours.

The authors concede that when the load is allowed to vary the conditions of house operation are reproduced, but they feel that under such circumstances it would be difficult to duplicate results. Tests conducted at the Bureau's experiment station do not justify such a conclusion, and the writers' opinion is that the real purpose of the test should be to learn what the boiler will do under operating conditions. If it be agreed that such is the purpose the effort should be to approximate the operating load. Observations taken at the time the Bureau's plans were being developed showed that the actual delivery of steam by a residence-heating boiler varies through a considerable range, even in moderate winter weather, and is affected principally by the times and conditions of firing, and the character of the fuel.

It was the effort when the Bureau's tests were first started to employ a method of starting and closing tests similar to that proposed in Par. 17 of the paper. The duration of the full test was then and still is made approximately 48 hours, and at the end of the first 24 hours the fires were brought to a closing condition and readings taken. Experience with this method, particularly with anthracite coals, showed so great variations between the two 24-hour periods and also between different tests that it was abandoned for what was formerly known as the "standard" method of starting and closing. With the latter method of test the overall efficiencies are usually lower, but results can be more easily checked with 24-hour tests by this method than by 48-hour tests by the former method. The results in the table on page 516 illustrate the variation.

The proposed method of starting and stopping can probably be used with a fair degree of success with anthracite coal of stove or chestnut size, but the writers' experience with other sizes was anything but encouraging. The evaporative performances quoted by different manufacturers, nearly all of which are believed to be for anthracite coal, appear high, and it may be due to the effort to get results with a short test or to the use of a method of starting and stopping which does not give correctly the quantity of fuel actually consumed.

Test	Coal	Method of Starting	Duration (hours)	Total Coal per sq. ft. of Grate	Overall Efficiency, %
761	Anth. Egg.	First.	47.07	89.6	77.6
761 (a)	"	"	24.28	51.8	80.0
761 (b)	"	"	22.78	37.8	74.5
868	"	Second.	24.25	80.3	62.8
853	"	"	49.37	128.3	62.9

It is obvious that the proposed method of starting involves less work in both the conduct of the test and the analysis of fuel and refuse samples than does the other method wherein the test is started with a new fire and analysis is made of the material remaining on the grate at the close of the test. It was because of this difference that the effort was first made to start with a fire which had been burning for 3 or 4 hours, but the writers were not able to carry out the method successfully with anthracite, lignite and some of the sub-bituminous coals, and they are of the opinion that others would experience similar difficulty. A duration of test sufficient to show a total fuel consumption of 40 lb. per sq. ft. of grate is, however, believed to be adequate if the so-called "standard" method is used.

The feeding of water to the boilers may be done as described in the paper or it may be done in another way if the output rather than input is measured. The latter course is followed in the Bureau's tests.

Connection may be made by a small line from a source of water under pressure to the return outlet of the boiler. In this line a small orifice may be placed and the pressure drop through the orifice read from a manometer graduated to read in rate of flow or simply in pressure difference. The manometer shows at any time at what rate the water is being fed, and this feed can be adjusted to keep the boiler water level practically constant.

Measurement of output can readily be made in either of two ways. One way is to send the steam delivered by the boiler through a closed-type feedwater heater, the condensing water circulating in the coil, and measure the condensate. The other way, which would obviate the use of calorimeter readings in computing results, is to measure the quantity and rise in temperature of the condensing water, the condensate returning to the boiler. Selection of equipment for either method may be made from a wide variety, and nearly any desired degree of accuracy obtained in measuring the output.

The Authors. The authors appreciate the discussion and suggestions presented by the different members. It is evident that the time has arrived for adopting a plan for testing house-heating boilers. It is probable that the Power Test Committee will be able to use the suggestions made in the discussion when they come to give consideration to this paper.

At no place in the paper do the authors discuss a method of rating boilers. That is a matter for future consideration. We still believe that in spite of the convenience of 250 B.t.u., the reason already given in the paper is of sufficient weight to justify asking that the most careful consideration be given to the authors' proposed definition of a foot of radiation.

No. 1546

STEAM SAFETY VALVES

CONSIDERATIONS LEADING TO THE DEVELOPMENT OF A NEW TYPE

By George H. Clark,[1] Boston, Mass.
Non-Member

The object of this paper is to place before the Society a record of the experimental work, and conclusions drawn therefrom, of an investigation into the theory and design of a safety valve. In the course of this work at least fifty valves have been designed, built and tested in an attempt to get at some of the fundamental facts as an aid to the designer and in verification of the theory presented later.

OBJECT OF PAPER

2 The experimental work had for its main object the development of a valve of high discharge capacity at the greatest possible efficiency and with the least possible weight of valve per pound of discharge.

3 In order to realize the conditions stated, it is clear that for any given diameter of valve seat the one giving the greatest lift will have the greatest discharge capacity if the seat angles are equal, and with equal lifts and equal diameters the flat seat (at right angles to perpendicular axis) is the most efficient. The flat seat was adopted at the start, due to its greater efficiency and to other considerations to be mentioned later, and effort was directed toward methods of increasing the lifts, at the same time retaining all the advantageous factors obtaining in low-lift valves.

4 The two most serious objections to high-lift valves are the shock produced at closure and the tendency to lift water. There has been almost endless discussion over these points, resulting in more or less misunderstanding. The shock of closure depends on the

[1] Instructor, Mech. Engrg., Mass. Inst. of Technology.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers. Contributed by the Boston Section.

height from which the disk drops to its seat and the velocity with which it is moving at the time it seats. By reducing the height from which the disk drops, the shock may be reduced so that the valve having an enormous lift after popping may close without shock if that lift decreases gradually with the receding pressure until it becomes small at the point of closure. This point has been very clearly brought to the experimenter, although no record is available, since the shock cannot be measured. The second objection to high-lift valves, that of lifting water, is in reality less serious. It is useless for anyone to say that this objection is unfounded, because the tendency to lift water is almost entirely dependent upon the capacity of the valve, and the higher the lift the greater the capacity. It must, however, be remembered that every steam boiler has a certain required safety-valve capacity which is the same for all classes of valves, so that for proper relief the total capacity of the low-lift installation and the high-lift installation is one and the same. Now if the capacities are the same, the velocities in the boiler nozzles are the same and one has no greater tendency than the other to lift water, since the rise in water level is due to velocity toward the nozzle. The two objections spoken of above may exist in high-lift valves, if, in the first case, the valve lacks the proper action, and in the second case if judgment is not used in the installation.

5 The results of shock are injury to the boiler and a short valve life, and if it is felt necessary this may be avoided, not by eliminating the high-lift valves, but by legislating against their bad feature, high closing lift. The lifting of water seldom occurs with stationary boilers and it can be prevented by the proper choice of valve nozzle and of discharge capacity per valve.

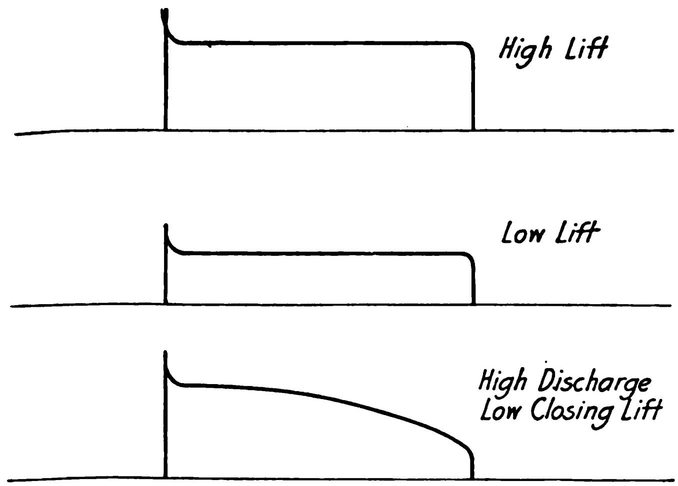
6 Fig. 1 shows the characteristic of certain high-lift valves which are objectionable, due to the high closing lift. Fig. 2 shows that of a low-lift valve in which the high closing lift has been obviated at the expense of discharge capacity. Fig. 3 shows the characteristic of an ideal valve having high capacity and low closing lift, and this was decided upon as the aim of the investigation.

RELATION BETWEEN DISCHARGE AND PRESSURE UNDER VALVE DISK

7 In order to investigate the subject thoroughly it is necessary to take into account two forces which produce valve action, not the least important of which is the variation in pressure on the spring-load

area of the disk — in other words, the pressure which exists below the disk in advance of and during the period of blowing. By means of certain assumptions we can apply the formula of flow proposed by Napier, and after the calculations are made they may be tempered by the actual conditions which exist, should they deviate from the assumptions.

8 Steam in the course of its passage from the boiler to the atmosphere encounters two reductions in area of passage which affect the



FIGS. 1, 2 AND 3 DIAGRAMS OF LIFT CHARACTERISTICS

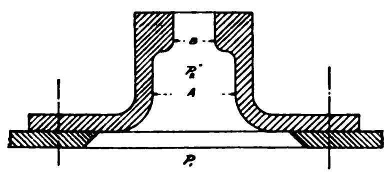


FIG. 4 REDUCTION IN AREA OF PASSAGE BETWEEN BOILER AND VALVE

pressure under the disk, one at entrance to the valve or the nozzle, and the other at the seat. The existent area conditions may be graphically represented by Fig. 4, in which the reduction of area from boiler to valve is denoted by the section at A, and over the seat by the section at B. The edges of the passage which form the approach to the section at A are rounded to a radius equal to the diameter of the passage, in order that the formula that we shall use may fit the case at hand. Let P_1 represent the boiler pressure, P_2 the pressure unc the disk, and then assume that by some means the area at B can changed as desired. When flow is established and conditions ha become constant, the weight of steam passing the section B is t same as that passing the section A, and if the weight passing B varied, that passing A goes through the same variation. Now t

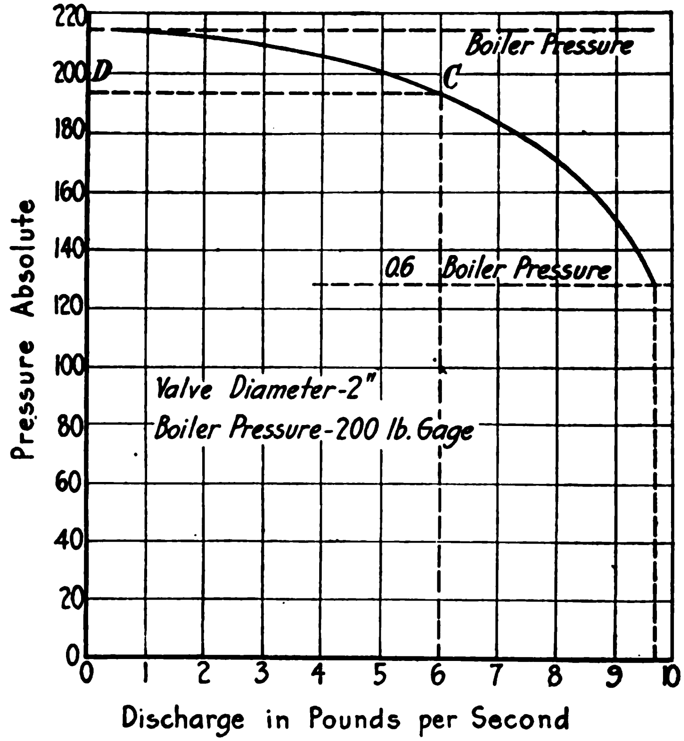


FIG. 5 RELATION BETWEEN PRESSURE UNDER DISK AND WEIGHT OF STEAM FLOWING

pressure P_2 depends on the weight flowing and the pressure Assign a constant value to P_1, say 200 lb. absolute, and by mean the following formula calculate the pressure P_2 with different weig flowing. So long as the discharge capacity at B is less than that a the formula

$$W = 0.029\, a \sqrt{P_2 (P_1 - P_2)}$$

holds. When P_1 is assumed to be constant the formula may written:

$$P_2 = \frac{P_1}{2} \pm \sqrt{\frac{P_1^2}{4} - \frac{W^2}{(0.029\,a)^2}}$$

after the quadratic equation has been solved.

9 Assume now, for an actual case, that the diameter at A is 2 in., giving an area 3.1416 sq. in., and that P_1 is equal to 200 lb. absolute boiler pressure. Then calculate different values of P_2 with different weights flowing. These values of P_2 may be plotted against the weights, and the curve shown by Fig. 5 will result. This curve is interesting, since it represents quantities which may be arrived at by calculation only, and which must exist. Its significance is as follows:

10 With a safety valve of 2 in. nominal diameter and adapted to discharge a certain weight of steam, the pressure under its disk may be found from the curve in the following manner: Suppose its discharge is 6 lb. per sec.; then the points C and D are located and the pressure under the disk may be read. It is clear then from this curve that the pressure under the disk varies as the weight of flow varies; and that for valves of low lift the reduction of pressure is not great, but that for high lifts it increases rapidly. This curve is discontinuous when the pressure under the disk reaches a point where it is equal to $0.6 \times$ boiler pressure, under which condition the area at B has become equal to that at A; the pressure P_2 is the minimum possible, and to all intents and purposes we have a standard orifice, so-called, the characteristics of which are shown by Fig. 6.

11 Our only assumptions up to the present are the neglect of friction and the rounded edges. It is clear that friction and square edges will tend to increase the drop in pressure, since both tend to reduce the discharge capacity of A, and it also follows that maximum capacity is not available with square edges at A. In other words, the full advantage of the hole in the boiler is not available unless this hole has rounded edges.

12 There results from the further consideration of the curve in Fig. 5 a simple explanation of why a safety-valve disk settles back after the first sudden pop. Since the reduction of pressure on the disk does not take place until the valve is open or until flow is established, the lifting force is, at opening, that due to full boiler pressure, and the instant flow is established this is reduced and the disk must necessarily settle back to a position of equilibrium. The design of a valve must then incorporate means to provide lifting force outside the seat area, to make up for the reduction of upward force on the

load area due to drop in pressure over and above that necessary to compress the spring.

13 In any safety valve there are three factors which affect the rate of flow over the seat, which are characteristic of the seat itself. They are (1) vertical lift, (2) seat angle, and (3) the contour of the passage which approaches the seat. The flat seat is adapted to open the largest area per unit of vertical lift, and hence should be employed, all other things being equal. Since the approach to a standard orifice makes the throat sectional area 100 per cent efficient, the characteristics of such an orifice should be employed. Professor E. F. Miller, of the Massachusetts Institute of Technology, was first to round the edges of a safety valve in order to increase the efficiency of discharge per unit of lift, and in so doing took a long step in the direction of high

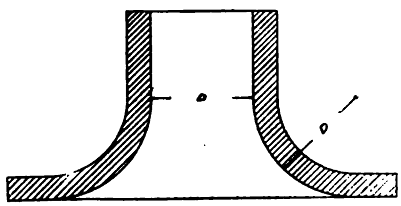


FIG. 6 STANDARD ORIFICE

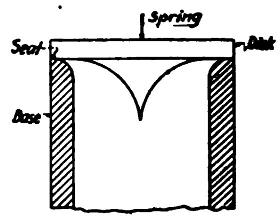


FIG. 7 VALVE INCORPORATING STANDARD ORIFICE EQUIVALENT

safety-valve efficiency. The valve shown in Fig. 7, in which the standard orifice (or better, its practical equivalent) is incorporated, follows the theory proposed by him several years ago. The full benefit of the standard orifice would be available, regardless of the type of lip provided, if the passage area from the seat outward increased; but to produce a high-lift valve the maximum available pressure between the lips consistent with maximum discharge must be used.

STEAM FLOW AS AFFECTED BY SHAPE OF ORIFICE

14 A consideration of the pressure conditions within a standard orifice, of which Fig. 6 is an example, will give valuable information. In this type of orifice the area of each section beyond the throat section is equal to that at the throat section; and it has been found that at the throat section and at each section of equal area thereafter the pressure is very closely equal to 0.6 × boiler pressure, and that the

back pressure into which this orifice discharges has no effect on either the weight discharged or the existent orifice pressures unless this back pressure exceeds 0.6 × boiler pressure. If it does exceed 0.6 × boiler pressure, the discharge capacity is reduced and the pressure throughout the orifice is increased. With the orifice of a shape shown in Fig. 8, but of equal throat-sectional area, the weight discharged is the same as before, but the pressure after the throat section varies inversely as the area.

15 It follows from the above consideration that in order to produce maximum pressure between the lips there must exist at full lift an equality of area at each section outward from the throat of the orifice which is located at the seat. Since the effective disk diameters are constantly increasing, the effective lift must be reduced as the passage is lengthened. This can be done by giving the passage an upward slope, as shown by Fig. 9, which illustrates graphically the

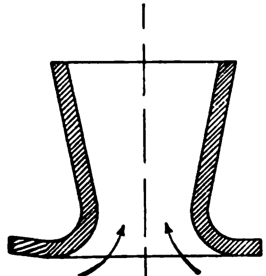


Fig. 8 Orifice with Pressure beyond Throat Varying Inversely as Area

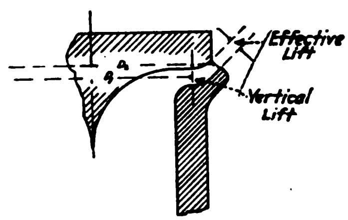


Fig. 9 Valve with Maximum Pressure between Lips

reduction of effective lift with increasing diameters such as D_1 and D_2, at which the vertical and effective lifts are measured.

16 If a valve of the above characteristics is constructed and blown, it will be found to have an extremely high lift, and the opening of such a valve is so sudden as to be truly explosive. A very large percentage of the pop lift is sustained if the popping pressure is held constant, and when such a valve starts to blow down, the lift gradually decreases during the blow-down period until a certain point is reached where the pressure becomes such that the valve disk drops suddenly to closure. This construction will produce a blow-down which is likely to be a third of the boiler pressure, and forces its designer to

incorporate means of regulation which will give a small blow-down without interfering with the capacity or other features of valve action.

REGULATION FOR WARNING AND BLOW-DOWN

17 The question of adjustment presents the real difficulty in safety-valve design. Means for regulation should in general perform their functions with as little effect as is possible on the lift, and, per unit of lift, should produce no effect on the discharge capacity. Other than lift, the two functions to be controlled by regulation of the variations in lifting force are warning and blow-down. A valve which chatters or vibrates at the instant of opening, or sizzles at that time, shows a lack of lifting force at low lifts, and one which shows an excessive blow-down demonstrates that the lifting force at low lifts is too great to allow of closure, and it follows that the requirements of warning and blow-down are somewhat combative. If it is

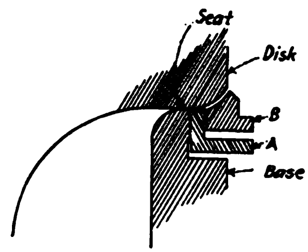


Fig. 10 Valve Adjustable for Variable Lifting Force

possible, means should be provided for making, in so far as is possible, the means adopted to regulate one independent of the other. Blow-down is an absolute necessity on the spring-loaded type, due to the existence of lifting force outside the spring chamber, and it is clear that attempts to reduce the blow-down will tend to decrease the lift and relieving capacity. In general, the real secret in the design of a perfect safety valve lies not in the production of excessive lifting force but in controlling and adjusting it; the action of a valve is dependent not on the quantity of lifting force but on variations of lifting force with discharge capacity and pressure.

DESIGN OF NEW VALVE

18 Fig. 10 shows a valve having means for adjustment which are based on the production of variable lifting force as stated above. The two annular rings shown at A and B carry on their upper sur-

faces curves which approximate in shape the curve of the disk. The inner ring A, which I choose to call the warning ring, is threaded upon the base of the valve just outside the flat seat and carries, threaded upon it, the ring B, which is adapted to control the blow-down. These rings may be lowered away from the disk, or given what I choose to call initial clearance.

19 Now it is clear that if both of these rings are lowered an extreme distance, when the valve tends to open no lifting force will be available. If the ring A is raised to a position near the valve disk, a lifting force will be provided which will depend (1) on the pressure produced between it and the disk and (2) on the extent of the disk area on which this pressure acts. As was shown in the discussion of the standard orifice, the pressure depends on the area of the passage, and the area in the valve is a function of the initial clearance and the shape of the curve. These factors are built into the valve to produce the desired result, which is to provide lifting force at small lifts to overcome excessive warning; and it has been found that with the proper shape of ring, on its initial clearance depends the warning of the valve. Since this warning ring alone produces only small lifts, it has no appreciable effect on the blow-down.

20 Now as the blow-down ring B is raised from its lowest position, there is a tendency to build up pressure between it and the disk when the valve is open, which tendency increases as the disk is approached. If a reasonably large clearance is left, it is clear that this ring produces almost no lifting force at low lift, since the area of the passage along this ring is large in comparison to that over the seat; and it follows, also, that as the disk lifts the effect of this initial clearance on the area of the passage becomes less and less as lifting continues. This can be illustrated by a simple calculation. Suppose that, measured at the extreme outside edge of this ring, the initial clearance is 0.03 in. and that the angle at this point is 45 deg.; then if the disk lifts 0.1 in. in a vertical direction, the height of the passage over the seat will be 0.1 in., and at the extreme edge the increase in passage height perpendicular to the curves will be 0.07 in., since with a 45-deg. angle the vertical lift is only seven-tenths effective — so that the total height of the passage is 0.1 in., the same as that over the seat. Suppose now that lifting continues; the area at the edge of this ring gradually approaches that over the seat and can be made to become equal to it by proper design at any chosen lift. It is clear then that for all of the initial clearance of the rings we have at full lift gotten back to the area conditions of a standard orifice.

21 Tests of this valve prove the value of this method of re tion, in that all the requirements of practice are afforded by it f of the high lifts. Any one sufficiently interested can lay out a of this type, and he will find that if the characteristics of the s to be used are known, he can calculate with reasonable accurac lifting force at any lift and also an approximate value of the tota

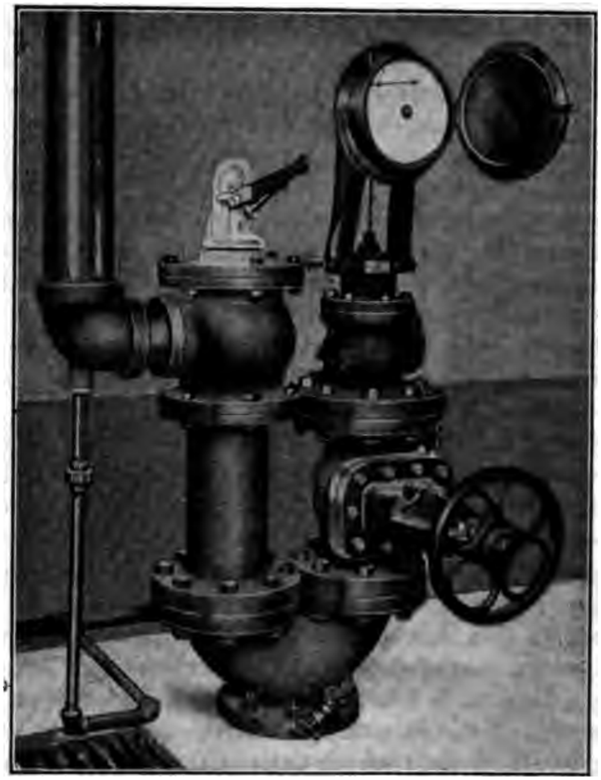


FIG. 11 VALVE ARRANGED FOR TEST

In calculations of this nature the reduction of pressure on the sp load area, as previously explained, must be taken into considera

DIAGRAMS SHOWING PERFORMANCE OF EXPERIMENTAL VALV

22 I have with some difficulty succeeded in getting some grams from valves under service conditions which, I believe, sh detail the results of the methods of regulation which are empl

Fig. 11 shows the arrangements of the valves for testing. The tests from which the diagrams were secured were carried on at the power plant connected with the new buildings to be occupied by the Massachusetts Institute of Technology.

23 One of the regular 4-in. valves was removed and a Y-base substituted in its place; the regular 4-in. was then placed on one of the branches and on the other was placed a special by-passed gate valve, and finally, on top of this, the valve on test was connected. Under the above conditions the valve on test was well above 3 ft. higher than the top of the drum. Those experienced in safety-valve

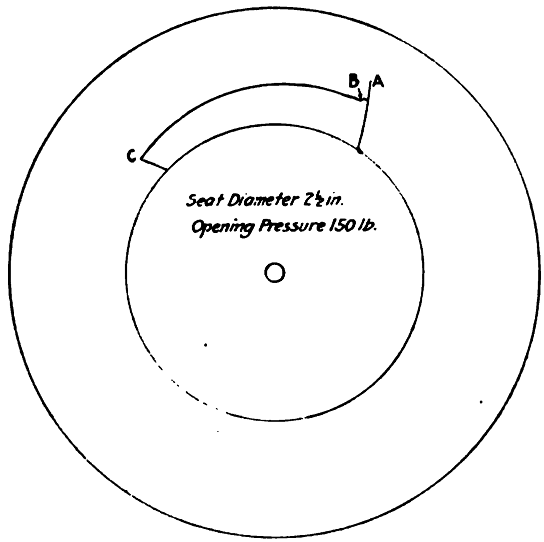


FIG. 12 PERFORMANCE CHART OF 2½-IN. EXPERIMENTAL VALVE

action will realize that this height is apt to affect seriously the action of the ordinary valve. The recording apparatus used consisted of a clock motion adapted to turn a chart once in 15 minutes. Bearing on this chart was a pen rigidly connected to the spindle of the valve and adapted to leave a record thereon of lift with respect to time, the lifts being multiplied by a ratio of approximately four to one.

24 Fig. 12 shows a chart of the performance of a valve with a 2½-in. seat. This valve was blown at 150 lb. gage boiler pressure and showed, under the existent regulation, 5 lb. blow-down and almost no warning. In order that it may be clearly understood, it has been transferred to a diagram having rectilinear axes, on which is plotted

the scale of lifts which resulted from the calibration of the recording lift gage with an accurate micrometer. In Fig. 13 the sudden pop lift is shown at *A*, and the corresponding sustained lift at *B*. The variations in lift at the point *B* are further evidence of the readjustments of pressure which determine the pop and sustained lifts, an idea of which may be gained from the theoretical considerations previously stated in this paper. The discharge capacity of this valve was such that the popping pressure was not maintained, and the curve from *B* to *C* shows the reduction in lift, which in this type always corresponds to a reduction in boiler pressure. At *C* the valve closed suddenly, with less than half its initial sustained lift prevailing.

25 In other words, the drop lift of this valve, which is numerically 0.09 in., is about equal to that of a low-lift valve, while its

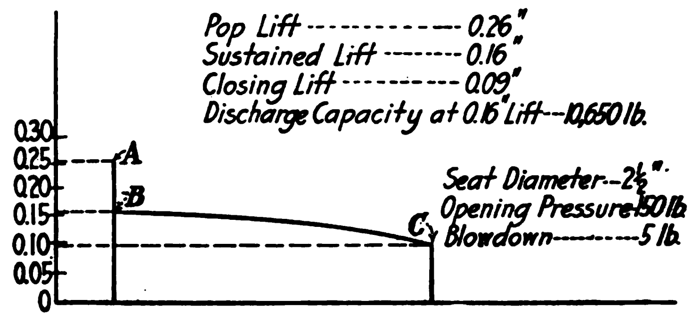


Fig 13 Analysis of Diagram Shown in Fig. 12

discharge capacity at popping pressure is nearly twice as great as regards lift, and is even more when seat angle and efficiency are taken into account. This diagram shows the action of the valve when adjusted to give action somewhat similar to that to be found in some types of valves now procurable, and is in a good many ways ideal. Attention is called to the smoothness of the curve between *B* and *C* as evidence of the equilibrium between lift and boiler pressure existent as the boiler pressure is reduced.

26 Fig. 14 shows precisely the same valve with its outside or blow-down ring lowered slightly. This diagram, like the one preceding, is reproduced from a photograph of the actual diagram, and has not been transferred to rectilinear axes because of the difficulty in showing the fluctuations in lift. The section of the curve at *A* shows the pop and sustained lifts as before. From *B* to *C* is shown the

gradual reduction of lift corresponding to decreasing boiler pressure, until the point C is reached, where the disk suddenly falls from a reasonably high lift to a much lower one. This action is due to the large initial clearance of the blow-down ring, which throws it out of action entirely when the lift value at C is reached. The fluctuations in lift at the point C show the lack of equilibrium at this point in the action, which results finally in the drop to the lower lift. At the lower lift, represented by D, all the supporting force is that due to the inner or warning ring.

27 The downward slope of the curve from D to E shows the tendency of the valve to decrease its lift with decreasing pressure, and

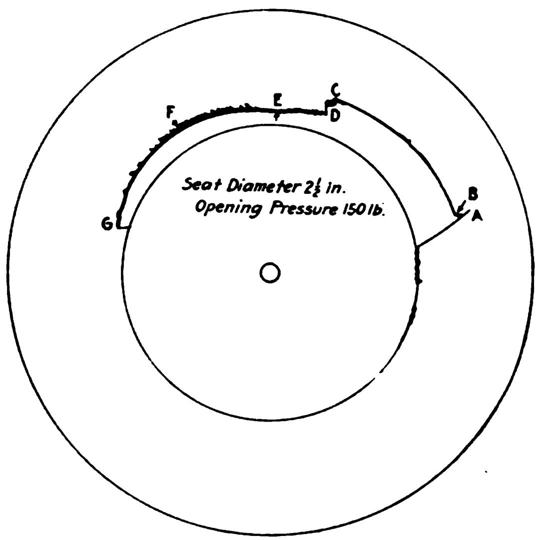


FIG. 14 PERFORMANCE CHART OF VALVE OF FIG. 12 WITH BLOW-DOWN RING LOWERED

closure would doubtless have taken place at D had not the fans been started at this point in an attempt to force the pressure back to the popping pressure. The section from E to F shows an increase in lift corresponding to the increase in pressure. The fluctuations in lift during this period show the tendency of the blow-down ring to come into action, it being prevented from doing so only by the lack of pressure. The fans were shut off without orders at F, and the valve immediately blew down and closed with 0.04 in. closing lift at G. If we suppose the diagram to have been completed by closure at D, as would have been the case without the attempted accumulation, this diagram would then represent the action of a high-lift valve with a closing lift less than that obtainable from any low-lift valve of ordinary design. That this action can be and is continually repeated by valves of this type, is a fact to which every one who has been a witness of the blowing of this valve will testify. To those who have heard such valves blow, no diagrams are necessary.

28 It was decided after the two preceding diagrams were secured to change to a smaller valve of lower discharge capacity, under which the pressure could be regulated as desired. It is my purpose to bring out by the three diagrams which follow the faculty this type

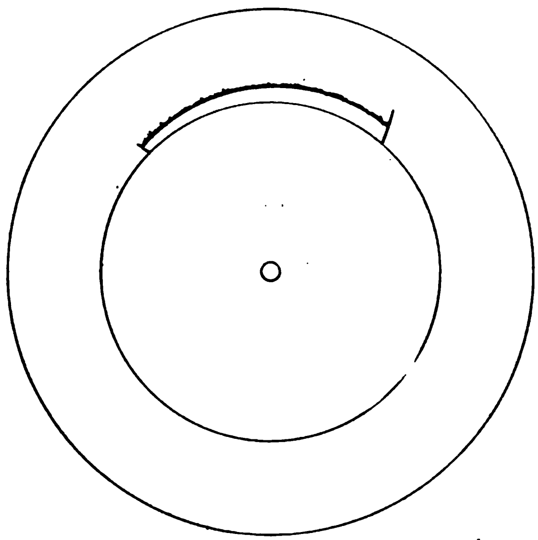


Fig. 15 Performance Chart of 2-in. Valve: Sustained Lift and Low Closing Lift

of valve has to respond, under ordinary conditions of regulation, to accumulated pressure above the opening pressure. The new Boiler Code rules, if put into force, will allow 6 per cent accumulation of pressure, and I wish to show by the diagrams what advantage may be taken of this accumulation.

29 Fig. 15 shows a chart taken from a 2-in. valve in which the pressure was not allowed to accumulate. The low closing lift in comparison to the sustained lift is to be noted. Fig. 16 shows the same valve with an accumulation of pressure over the opening pressure of 4 per cent. This is under the same regulation conditions

as the next preceding chart. The substantial increase in lift and corresponding discharge capacity is to be noted, together with the same closing lift.

30 In Fig. 17 the valve was regulated to give a low sustained lift, and was then given the full 6 per cent accumulation. It is clear that the lift at *B* is more than 100 per cent greater than that at *A*, so that the discharge capacity at 6 per cent accumulation is more than 100 per cent greater than that at the opening pressure. The low closing lift is to be noted as before. The blow-down on this valve was about 2 lb., so that this chart was made between a maximum pressure

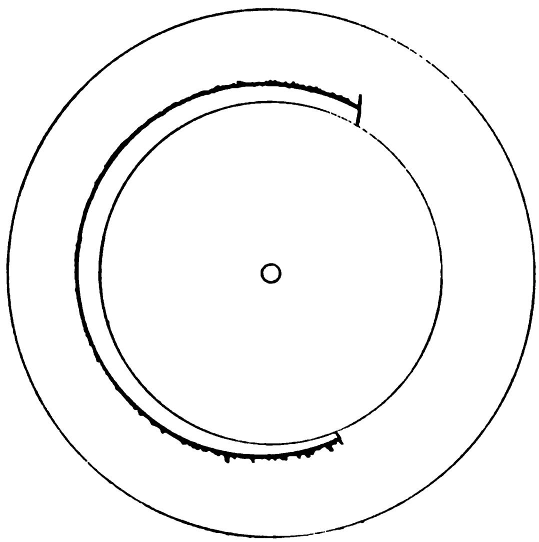


Fig. 16 Performance Chart of 2-in. Valve: Pressure Accumulation between Opening and Closing

of 159 lb. and a minimum pressure of 148 lb. The new Boiler Code specifications rate valves on their sustained lift at the popping pressure, and so it is clear that with this type the basis is amply safe.

31 In presenting the diagram shown in Fig. 18 for consideration, I believe that I am bringing forward one of a safety valve which is absolutely new, both in conception and results. It has been the aim of safety-valve designers to produce a valve which relieves a boiler with an action that is smooth and contains as few sudden fluctuations in pressure as is possible. The Boiler Code Committee recognized the capabilities of standard valves to respond to accumulated pressure

by incorporating the 6 per cent accumulation rule. The ideal safety valve for practical purposes is, I believe, one in which a reasonably low lift is maintained at the popping pressure, but one which will on the slightest accumulation respond with an extremely high lift to take care of the emergency. If a low lift is available at the popping pressure for ordinary relief, a low blow-down may be incorporated, the condition imposed by absolute safety being that at slightly above the popping pressure the lift will have increased to such an extent that any possible emergency will be taken care of.

32 Fig. 18 shows a diagram from a valve which produces the

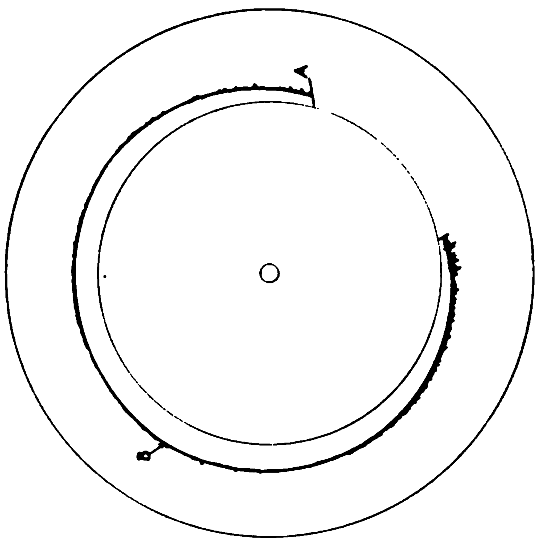


Fig. 17 Performance Chart of 2-in. Valve: Low Sustained Lift with Pressure Accumulation

above action, and is what may be called a second-opening valve. Many of this type have been built, and it has been found that with the proper regulation of rings, all valves having two rings will produce this action in modified form. Imagine, first, one of the ordinary type blown with the outside ring removed. This would result in a low lift and a very low blow-down. Now suppose that the blow-down ring is in place, but that it is given such a large initial clearance that the valve will act, provided that it has sufficient capacity to relieve the boiler, exactly as before, but if it has not the required capacity the pressure will accumulate and the increase of lift resulting with only the inner ring in action will finally bring the outer ring into action and a second pop will occur. This pop will result in a high lift, with corresponding high discharge capacity, and the valve size may be proportioned to give the required safety.

33 That this action can be accomplished in actual practice is shown by the diagram in Fig. 18, which resulted from the action of a 1¼-in. valve on a boiler, or rather boilers, of sufficient capacity to produce the required accumulation. The first pop and sustained lifts are shown at *A*, the sustained lift being very low. The increase in lift between *A* and *B* finally results in the second pop, shown at *B*, which is the result of accumulated pressure. Between *B* and *C* the lift gradually falls as the pressure is reduced, until at *C* the outside

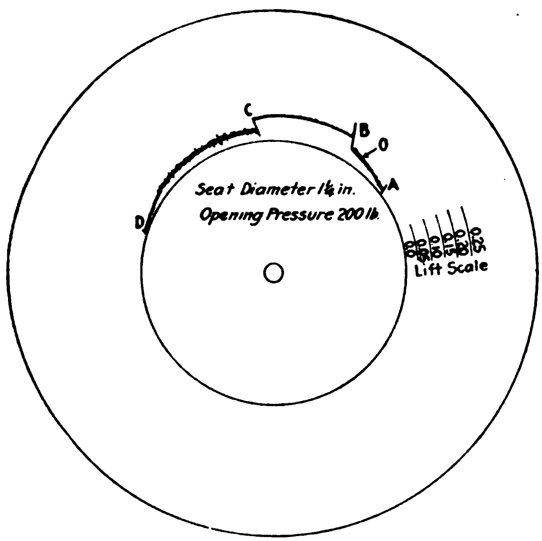


FIG. 18 PERFORMANCE CHART OF A SECOND-OPENING VALVE

ring goes out of action and the disk drops to a low lift, which is at the given pressure just sustainable by the inner ring. As the pressure is reduced the lift decreases, and after a slight blow-down the valve closes at *D* with a very low closing lift. This valve had a seat diameter of 1¼ in. and lifted at its highest sustained point about 0.11 in. This lift was the result of about 4 per cent accumulation, and the calculated discharge capacity is 5100 lb. per hour, which is approximately 170 boiler h.p.

FEATURES OF DESIGN

34 Careful design will allow of bringing the second pop down very near the first one, and the limits of regulation may be so built into the valve by limiting the possible up and down movement of rings, so that this second pop will always come within a given range of pressure. This type of action has many advantages to recommend it. The absence of sudden excessive changes in relieving capacity and the low lift at closure are evident from the diagram. It is clear that if a boiler had two valves of this type, the second might be set to blow at a pressure corresponding to the point *O* on the diagram, so that the two valves would be able to handle the boiler at all ordinary times without the second pop, but in the case of emergency both would rise to the second pop lift. The installation would then be essentially a low-lift installation, but would have all the advantages of high lift in the case of emergency.

TABLE 1 RESULTS OF DISCHARGE TESTS ON VALVES OF THE NEW TYPE

Test No.	Valve Diameter, In.	Opening Pressure, Lb. per Sq. In., Gage	Blowing Pressure, Lb. per Sq. In., Gage	Percentage Accumulation	Discharge, Lb. per Hour	Efficiency, Per Cent	Lift, In.
1	2.5	150	158.8	5.86	17,325	99.43	0.252
2	2.5	200	212.9	6.44	16,892	78.88	0.238
3	2.0	200	203.4	1.70	10,176	101.57	0.145
4	3.0	248.5	248.5	0.00	24,482	96.26	0.304

DISCHARGE TESTS

35 In the course of the experimental work very few discharge-capacity tests could be run, since most available plants have no means of measuring the water used. Four tests were run at Annapolis, Md., at the United States Naval Experiment Station, which establish the fact that the standard orifice type is nearly 100 per cent efficient in discharge capacity per unit of lift. The first three tests were made early in the experimental work, the fourth being run at a later date. The reported results of these tests are shown in Table 1.

36 Test No. 2, when compared with the other tests, shows so low an efficiency that it obviously includes some error in the reported data, so it seems fair to assume that the efficiency is very nearly 100

per cent, or, in other words, within the limits of the experimental error in a test of this nature. As a comparison it may be of interest to note that the 3-in. valve assembled for test weighed only 85 lb.

37 It is always interesting to compare the results obtained from any piece of apparatus with those which we know to be theoretically possible. In the case of safety valves the full-opening valve serves as a convenient basis of comparison, in that its capacities may be calculated without reference to the type of valve used. If we attach a standard orifice to a boiler we have then the most efficient way of discharging steam, and this on a basis of throat diameter may be characterized as the full-opening valve.

38 In line 1, Table 2, are shown the capacities of full-opening valves of several diameters figured directly from Napier's formula. Line 2 shows the discharge capacity of valves of the same diameters

TABLE 2 CALCULATED CAPACITIES OF FULL-OPENING VALVES

Opening Pressure, 150 Lb. Gage

	Nominal Valve Diameter, In.	2	2½	3	3½	4
1	Discharge capacity of a full-opening valve, lb. per hr.	26,600	41,600	59,900	81,600	106,500
2	Discharge capacity of A.S.M.E. valves (from table), lb. per hr.	2,529	3,613	5,419	6,954	8,670
3	Discharge capacity of proposed type (from tests), lb. per hr.	5,310	10,640	15,400	......	......
4	Spring load of (1) per 1000 lb. discharged, lb.	17.7	17.7	17.7	17.7	17.7
5	Same for A.S.M.E., lb.	185	204	196	208	218
6	Same for proposed type, lb.	88.5	69.2	68.9	......	......
7	Percentage of full opening attained by proposed type.	26.6	25.6	25.7	......	......
8	Same for A.S.M.E.	9.51	8.67	9.05	8.50	8.15

as read from the table in the recent Am.Soc.M.E. Boiler Code. I have assumed that these values represent standard conservative practice. Line 3 shows the discharge capacity of the type of valve which I have described, and is the result of calculations involving the sustained lift as measured from the diagrams at the popping pressure.

39 It seems reasonable to believe that the spring load of a safety valve is to some extent a measure of the shock at closure, which may be expected. Since all valves of the same diameter have approximately the same spring load, some other basis than diameter must be chosen for comparison. Discharge capacity is the first requisite in all valves, and so the spring load has been calculated per 1000 lb. of steam discharged per hour. Line 4 shows the spring load per 1000 lb. discharge of full-opening valves, which is, of course, constant.

Lines 5 and 6 show the same quantity calculated for Am.Soc.M.E. valves and for the proposed type. Lines 7 and 8 show the percentage of full opening attained by the two available types.

40 There is reason to believe that we shall soon have boilers which will operate at 600 lb. pressure. Consideration of some of the values given in this table is interesting in this connection. A 4-in. valve set at 600 lb. pressure will have a spring load of approximately 7540 lb., or about 3¾ tons. Now if a 2-in. valve can be constructed to give twice the lift of the 4-in., it will have the same discharge capacity, but its spring load will be only one quarter as great, or 1885 lb. In other words, the discharge capacity varies as the first power of the diameter and the lift, but the spring load, which is independent of the lift, increases as the square of the diameter. It seems clear from this that high-pressure boilers will of necessity require small-diameter high-lift valves.

41 Table 2 shows the net result of the development in spring-loaded safety valves, and serves to emphasize the opportunities which may be grasped by the designer.

42 The question of the effect of back pressure in the casing of a safety valve has been much discussed. It has been said that it may operate in two ways to reduce the discharge capacity. First, the effective lifts may be reduced, and, second, it has been held that back pressure reduces the efficiency of discharge of the orifice. If any of the disk area is exposed to back pressure the lifts will be reduced to some extent, but consideration of the theory of flow through an orifice, and particularly a standard orifice, brings out the fact that as long as the back pressure in the casing is less than 0.6 × boiler pressure, the discharge of the orifice is not affected.

43 The discharge-capacity tests made some years ago by P. G. Darling, Mem.Am.Soc.M.E., went some distance in establishing this fact, although no general conclusions were drawn. It remained for a section of a Committee of the American Society of Refrigerating Engineers, with Prof. E. F. Miller, Mem.Am.Soc.M.E., as chairman, to establish this fact beyond question. In refrigeration installations long discharge pipes are required, and the pipe friction produces so much back pressure that either the ordinary type of valve is useless or its discharge pipe must be of an excessive size. Experimental work resulted in a valve which was so constructed as to take advantage of back pressure to increase its lift without in any way having its discharge capacity per unit of lift reduced. That valve was due to the ingenuity of F. L. Fairbanks, Mem.Am.Soc.M.E., of the Quincy

Market Cold Storage and Warehouse Company, and has been adopted as the standard in Massachusetts for refrigeration work.

44 The possibility of working with back pressure operates to reduce the discharge pipes to reasonable sizes in spite of their length.

EXAMPLES OF THE NEW VALVES

45 Fig. 19 shows a complete view of a valve which has the desired advantages. In the first place, valves of high lift, and by that I mean higher lift than any of those obtainable at present, must be guided from above, as guides in the base will cause too great a drop

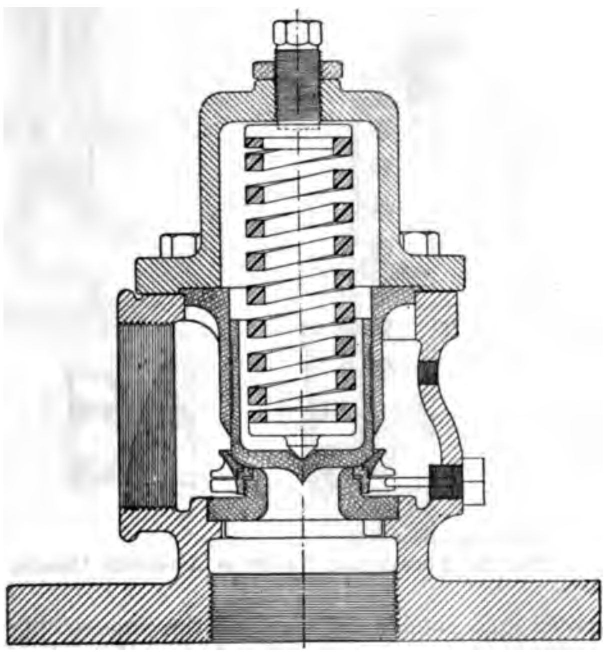


FIG. 19 STEAM SAFETY VALVE OF APPROVED DESIGN

in pressure on the disk area. In this view the disk is made in the shape of a hollow piston, which is guided by a cylindrical sleeve that encloses the spring chamber. The fit existent between these is extremely loose, and no attempt is made to prevent leakage by the disk. This loose fit insures that the disk will not bind because of expansion or contraction between the members due to temperature changes, and is long enough in relation to the diameter of the disk so that it will never cock. Its looseness alone insures that the disk will come to a uniform bearing on the seat, and it has been found that under no conditions of back pressure has the leakage by it been suffi- cient to produce pressure in the spring chamber, which is vented. With this type, the back pressure may be built up to nearly 0.6 boiler pressure without hampering the discharge capacity in the least, and since the back pressure is a result of the effective opening of the valve, it has no effect whatever on regulation if it is brought about by the means shown.

46 In the case of duplex or triplex valves having a common discharge pipe, the opening of one will tend to open the other, due

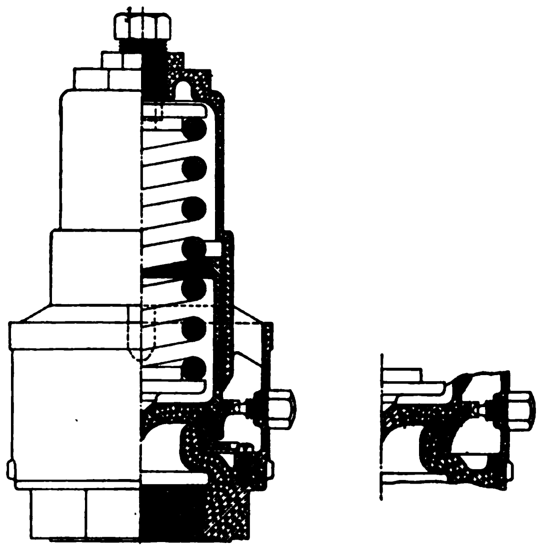


FIG. 20 LOCOMOTIVE VALVE OF APPROVED DESIGN

to the exposed area outside the seat and the impossibility of back pressure acting downward on the disk. It must be understood that the commercial types of valves which are shown in the following illustrations are not dependent on back pressure for any part of their action, but are in every case adapted to work with it without harmful effect. Fig. 20 shows a locomotive valve of approved design, and Fig. 21 the proposed marine type of exceedingly large capacity.

47 Having shown a valve in which there is substantial increase in discharge capacity over those in use at present, and having demonstrated that this may be accomplished with lower closing lifts than are now the rule, attention is called to a possible improvement in this type. It will have been noted by this time that in all the types shown

the passage turns upward from the seat region, and it is also clear that so far as the area conditions go it might as well be turned downward. If it is turned downward the area conditions will remain constant, as illustrated in Fig. 22, but there will be available as lifting force a factor which would act upward on the disk, and which is commonly known as a jet action. Some few valves of this type have been constructed, and the results gained have been convincing of their possi-

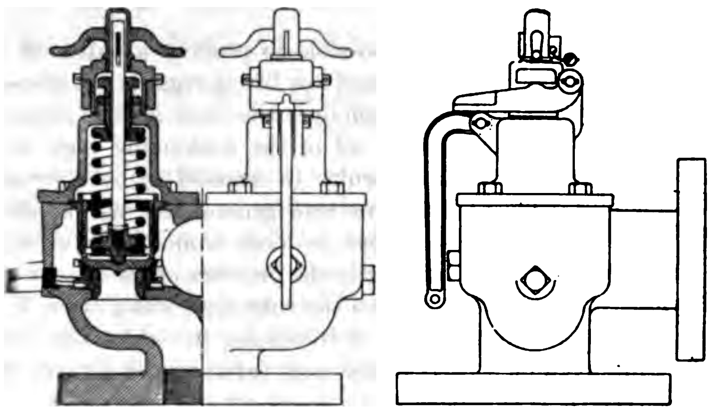


FIG. 21 MARINE VALVE OF APPROVED DESIGN

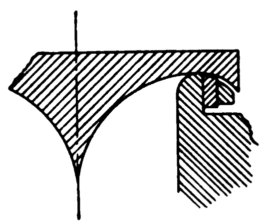


FIG. 22 VALVE WITH JET ACTION

bilities. Enormous lifts have been attained with the same absence of shock at closure as is the case with valves herein described. The problem has been by no means solved, but advances have been made in a direction which tends to indicate that it will be solved finally, and will result in enormous discharge capacities per inch of valve diameter, with the same efficiency per unit of lift. It is clear from a consideration of the diagram that the force due to jet action will not be destroyed by back pressure, as the velocity of the issuing jet of steam is a function of the orifice pressure, which pressure is not affected by the back pressure.

48 In closing this paper, attention is directed to the fact that this valve is not essentially a high-lift valve. It may be designed to give low lifts as well as high lifts, and is easily adaptable to the proposed Am.Soc.M.E. specifications. It will, of course, under the specifications, benefit by reason of its flat seat and by a discharge capacity about 30 per cent greater than the 45-deg. seat now in common use.

49 I have shown that it is possible to produce a valve of high lift as regards discharge capacity and low lift as regards the shock at closure. This one feature alone removes the most serious objection to high lift. It is clear that for all of the reasonably high initial lift the valves have a marked faculty to respond to over-pressure. Finally, high-lift valves must come into general use, as the safety requirements in boiler practice must be more economically met than is the case at present. It is distinctly the province of the experienced and unbiased engineer to lay down the rules that will govern future safety-valve installations, but in determining the rules the valves must be considered on a basis of discharge capacity and action rather than lift.

No. 1547

EFFICIENCY OF PROPULSIVE MACHINERY AND LATE DEVELOPMENTS IN NAVAL ENGINEERING

By H. C. Dinger,[1] Washington, D. C.
Non-Member

In January, 1912, Capt. C. W. Dyson presented a paper before the Engineers' Club of Philadelphia, entitled Propulsive Machinery and Oil Fuel in the U. S. Navy, which was followed later in the same year with a paper entitled Engineering Progress in the U. S. Navy, read by the then Engineer-in-Chief, Rear-Admiral H. I. Cone, before the American Society of Naval Architects and Marine Engineers at their annual meeting in New York City.

2 The first of these papers contained a comparison of the performances of the reciprocating-engined vessel *Delaware* and the Curtis turbine-engined vessel *North Dakota*, while the second paper contained a comparison of the performance of the *Delaware* and the Parsons turbine-engined vessel *Utah*. In both of these comparisons the decided superiority in economy of the reciprocating engine over the turbine for battleship propulsion was fully demonstrated.

3 That this apparent superiority was due to faulty practice in the design of propellers in the case of the Parsons turbine, and to both turbines and propellers in the case of the Curtis turbine, has been demonstrated by the trials of later vessels, in which the turbine revolutions have been very considerably reduced; and in the cases of Curtis turbines the turbine design has been radically changed in quite a number of particulars. In the cases of the latest two turbine-engined vessels, the economy has been still further improved by fitting cruising turbines driving the main shafting through mechanical reduction gear.

4 Since the trials of the *Delaware*, *North Dakota*, *Utah* and her

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TABLE 1 HULL CHARACTERISTICS OF VESSELS

	Delaware	North Dakota	Utah	Arkansas	New York	Oklahoma	Nevada	Pennsylvania
Length on load water line	510 ft. 0 in.	510 ft. 2½ in.	510 ft. 0 in.	554 ft. 0 in.	565 ft. 0 in.	575 ft. 0 in.	575 ft. 0 in.	600 ft. 0 in.
Beam	85 ft. 2⅝ in.	85 ft. 2½ in.	88 ft. 2½ in.	93 ft. 2⅝ in.	95 ft. 2⅝ in.	95 ft. 2⅝ in.	95 ft. 2⅝ in.	97 ft. ½ in.
Draught on trial	27 ft. 2 in.	26 ft. 11⅜ in.	27 ft. 10⅜ in.	27 ft. 11$\frac{1}{16}$ in.	27 ft. 9⅜ in.	28 ft. 6 in.	28 ft. 3$\frac{1}{16}$ in.	28 ft. 9⅝ in.
Displacement on trial, tons	20,209	20,051	21,268	25,496	26,300	27,500	27,222	31,400
Coefficient of fineness, block	0.6000	0.6000	0.5837	0.6220	0.6100	0.6170	0.6080	0.6460
Coefficient of fineness, mid-section	0.9798	0.9810	0.9792	0.9850	0.9900	0.9840	0.9820	0.9820

TABLE 2 MACHINERY CHARACTERISTICS OF RECIPROCATING-ENGINED VESSELS

	Delaware		New York		Oklahoma	
Engines, type	Vertical, inverted, 4-cylinder, triple-expansion					
Engines, number	2		2		2	
Cyl. diam., in., high pres.	38.5		39		34.98	
Cyl. diam., in., medium pres.	57		63		59	
Cyl. diam., in., low pres.	(2) 76		(2) 83		(2) 78	
Stroke, in.	48		48		48	
Clearances (total per cent):	Top	Bottom	Top	Bottom	Top	Bottom
High-pres. cyl. (H.P.)	15.57	16.77	14.29	13.54	7.226	9.586
Med.-pres. cyl. (M.P)	12.71	14.43	12.18	14.85	9.284	12.499
Low-pres. cyl. (L.P.)	11.91	13.04	13.75	16.30	8.900	12.400
Ratio H.P.: M.P	2.22		2.61		2.890	
Ratio M.P.: L.P	3.58		3.47		3.508	
Ratio H.P.: M.P	7.92		9.06		10.120	
Valves, type	Piston, with short, straight ports.					
Valves, number	1 H.P., 2 M.P., 4 L.P.		1 H.P., 2 M.P., 4 L.P.		1 H.P., 2 M.P., 4 L.P.	

TABLE 3 MACHINERY CHARACTERISTICS OF TURBINE-ENGINED VESSELS

	Arkansas, Utah and Florida	North Dakota	Nevada	Pennsylvania
Type of turbines	Parsons reaction	Curtis, wheel stages only	Curtis, wheels and drums	Curtis, wheels and drums
Number of shafts	4	2	2	4
Rotor diameters, in.:[1]				
High-pressure	71	144	132–138½	132–120
Low-pressure	97	......	120	138
High-pressure cruising	71	......	38–41	30
Intermed.-pres. cruising	70	......	....	....
Low-pres. cruising	....	......	....	30
High-pressure astern	71	......	137.55	84–108
No. of Stages, high-pres.	6 (91 rows)	9	5 wheel 25 drum	4 wheel stages 30 rows on drums
No. of Stages, low-pres.	6 (39 rows)	......	38	55 " " "
Revolutions per minute	320	280	213	210
Steam pres., lb. per sq. in.	200	250	250	250

[1] ... except in second column.

sister ship *Florida*, the following vessels have been completed and tried and commissioned: *Arkansas*, *Wyoming*, *New York*, *Texas*, *Oklahoma*, *Nevada* and *Pennsylvania*. Of these vessels the *Arkansas* and *Wyoming* are sister ships and are fitted with Parsons turbines on four shafts. The *New York* and *Texas* are alike and have reciprocating engines on two shafts. The *Oklahoma* and *Nevada* are also sister ships, the first having reciprocating engines while the other has Curtis turbines with cruising turbines and reduction gear. Both vessels have two shafts. The latest vessel, the *Pennsylvania*, is also fitted with Curtis turbines and with cruising turbines and reduction gear, but the power is divided among four shafts.

5 In tracing the advance in economy in turbine propulsion, that vessel of each pair of sister ships showing the better trial performance will be considered in order to eliminate favoritism of one type over the other, and with this object in view the following vessels will be used in the argument:

a Reciprocating-engined vessels *Delaware*, *New York*, *Oklahoma*.

b Parsons-turbine-engined vessels *Utah*, *Arkansas*.

c Curtis-turbine-engined vessels *North Dakota*, *Nevada*, *Pennsylvania*.

6 In order to furnish a proper understanding of the conditions of operation of the machinery, the characteristics of the vessels' hulls, their machinery and their propellers must be at hand for reference; and these characteristics are given in Tables 1–3.

7 The following is a detailed description of the turbines of the *Nevada* and *Pennsylvania:*

8 *U. S. S. Nevada Turbine Data.* The high-pressure turbine has five wheels and two drums, the 5th-stage wheel being secured to the forward drum, the steam being divided after leaving the wheel stages. The wheels are separated from each other by diaphragms containing the steam nozzles. There is one stage on each wheel with two rows of buckets each, except the 1st-stage wheel which has four rows, all having a pitch diameter of 132 in. The forward drum, 6th to 14th stages, inclusive, has nine rows of buckets with a pitch diameter of 119 in., and the after drum, 15th to 30th stages inclusive, has 16 rows of buckets with a pitch diameter of 118¾ in.

9 The low-pressure turbine has one drum of 38 stages and 38 rows of buckets for ahead conditions, and one wheel and one drum for backing. The backing wheel, one stage, has four rows of buckets

with a pitch diameter of 137.55 in., and the drum, nine stages, has 11 rows of buckets with a pitch diameter of 120 in. The backing wheel and drum are separated by a diaphragm containing steam nozzles.

10 *U. S. S. Pennsylvania Turbine Data.* The high-pressure ahead turbine has three wheels and two drums, the 3d-stage wheel being secured to the forward drum. The wheels are separated from each other by diaphragms containing the steam nozzles. There is one stage on each wheel, with three rows of buckets (132 in. P. D.), except the 1st-stage wheel which has four rows of buckets. The forward drum has 18 rows of buckets and the after drum 21 rows of buckets (both 120 in. P. D.).

11 The low-pressure ahead and astern turbine has two drums. The forward drum has 42 rows of buckets for ahead conditions (138 in. P. D.), and the after drum 13 rows of buckets for astern conditions (120 in. P. D.). The high-pressure astern turbine has three wheels and one drum; each wheel has three rows of buckets except the first stage, which has five rows (108 in. P. D.). The drum has 17 rows of buckets (84 in. P. D.).

12 The high-pressure cruising turbine has four wheels, two rows of buckets on each, except the first has three (30 in. P. D.). The low-pressure cruising turbine has six wheels, two rows of buckets on each (30 in. P. D.).

13 The cruising turbines for the *Nevada* were designed to give requisite power to pass steam through them up to a speed of 15 knots. Those for the *Pennsylvania* were designed to pass steam through them up to a speed of 12 knots. They were, however, used up to 15 knots by making use of special by-passes. In both vessels the cruising turbines may be used at higher speeds than those originally designed by keeping the cruising turbines in operation and at the same time admitting some live steam into the main high-pressure turbines.

COMPARISON OF ECONOMICAL PERFORMANCES

14 The curves in Figs. 1 and 2 have been plotted in order to establish a definite basis of comparison of the economical performances of the different vessels, and it is believed that the comparison thus made is very easily comprehended. The curves are plotted from the trial-trip data of the various vessels, which are believed to be reliable, but which cannot be taken with the refinement of a laboratory experiment.

CONSTRUCTION OF CURVES

15 Fig. 1, lower part, gives the effective horsepower and revolution curves for each of the several vessels. Fig. 1, upper part, gives the effective-horsepower and indicated- or shaft-horsepower curves for

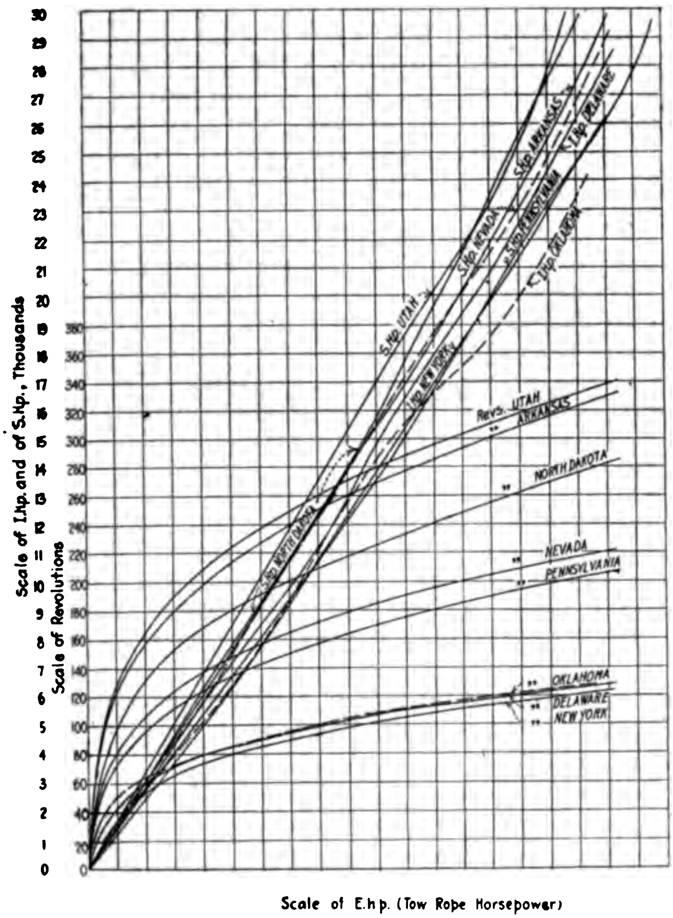

FIG. 1 COMPARATIVE PERFORMANCES OF NAVAL VESSELS

each of the several vessels. As plotted they are really curves of relative propeller efficiency, and the closer the curves come to the horizontal the greater the propeller efficiency. The curves indicate graphically the general improvement in propeller design made in succeeding vessels. It will be seen that curves of the two latest vessels, the *Oklahoma* and *Pennsylvania*, run fairly close togethe to an effective horsepower of 12,000, though one is a turbined v running at 181 r.p.m., and the other a reciprocating-engined v running at 118 r.p.m.; but above this point the reciprocating-en propellers are decidedly superior.

16 Fig. 2 gives the effective horsepower and B.t.u. utili obtained by plotting the value of $WH \div$ (E.H.P.), where $W =$ t water consumption per hour at point where the effective horsep (E.H.P.) mentioned is used, and $H =$ total heat of one poun steam at boiler less feed temperature. These curves, therefore, a comparison of combined engine and propeller efficiency, elimina the boilers. Similar curves might be plotted showing the en

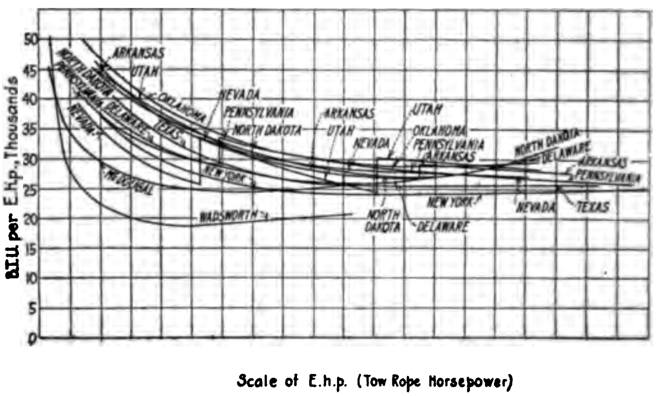

FIG. 2 COMPARATIVE PERFORMANCES OF NAVAL VESSELS

efficiency only. But as the type of machinery affects the prop efficiency, it is a better comparison to have the combined efficienc engines and propellers.

17 In plotting these curves, where saturated steam is used steam is assumed to be dry. Where superheated steam is used temperature of steam recorded is employed. It is believed that curves are truly comparable graphs of the economical efficiency of machinery of the different vessels, and are unique in this partic since they take into account propulsive work performed per l unit supplied in the steam to the engine. They also take into acco water for all purposes, except for the *North Dakota* the water dynamos is excluded.

18 Any other vessel for which the water consumption, st pressure, feed temperature, shaft horsepower and effective ho

power are known, can be compared in the same way and the relative efficiency of the machinery shown.

19 The disclosures from the curves of B.t.u. per effective horsepower are extremely interesting. The *New York* here shows the best performance, and this represents the combination of efficient propellers, good reciprocating engines, and the use of a moderate degree of superheat. In comparing the *Delaware*, *New York* and *Oklahoma*, we find their propeller efficiencies stand in the following order: 1 *Oklahoma;* 2 *New York;* 3 *Delaware.* The order of their economical efficiencies is: 1 *New York;* 2 *Delaware;* 3 *Oklahoma.* Both the *New York* and *Delaware* have superheat; the *Oklahoma* has none. The relative standing of the *New York* and *Delaware* is largely accounted for by the difference in propeller efficiency. The *Oklahoma* has better propeller efficiency and fully as good, if not better, engines, hence one is easily led to the conclusion that the falling off in the economical efficiency of the *Oklahoma* is due to the non-use of superheat. Similarly the performance of the *Nevada* and of the *Pennsylvania* could have been improved nearly 10 per cent by using superheat.

20 The *Nevada* and the *Pennsylvania* are fitted with geared cruising turbines. The superior economy secured by these over other turbine vessels is clearly shown by the curves. The curves also indicate that the Parsons vessels at high speeds give about the same economical result as the Curtis-turbined *Nevada* and *Pennsylvania.* The propeller efficiencies of the *Nevada* and *Arkansas* are about the same. However, within the range of the intermediate-pressure cruising turbine the Parsons vessels are superior, while at very low speeds the economy in the turbine vessels is about the same. The *Pennsylvania* at high power shows slightly better economy than the *Nevada*, this being expected since it is a later and somewhat better design, and better propeller conditions due to the use of four screws running at same revolutions are obtained. At low power the *Nevada* gives better economy results. The curves of the *Utah*, *Arkansas*, *Nevada* and *Pennsylvania* are really a series of curves, as they jump from one combination of turbines to another.

21 In plotting the curves, that for each condition is laid down and carried up to the point where the next combination of turbines comes in. Actually, by means of by-passing steam, the results will not follow the separate lines; but, as the limit of one combination is reached, by-passing is resorted to and a better result secured than if a complete shift to the next combination were made.

22 As far as economy at the different speeds is concerned, there

is not much choice between the different types of machinery w they have proper propeller efficiency. At very low powers the ciprocating engine is about as economical as the geared cruising t bine, both of these being more economical than directly connec cruising turbines. The use of superheat would improve the perfo ance, however, especially that of the reciprocating engine and pulse turbine.

23 The foregoing indicates that when all advantages of de are used, about the same economical results are secured with eit reciprocating engines, Parsons turbines or Curtis turbines, and t other matters besides mere economical performance should de the choice of prime mover as between the several types. It also g to show that when the losses are properly guarded against it is range of temperature that really counts in securing efficient propul machinery. Therefore, the most effective measures that can applied are those which increase the range of temperature that car used and those which conserve the use of the heat units suppl Hence we have reasons for use of superheat, the employment of highest possible vacuum practicable, the best possible use of fe heating apparatus, and economical operation of auxiliaries.

24 The curves of the destroyers *McDougal* and *Wadsworth* h also been plotted, Fig. 2. Owing to large differences in hulls a speeds, the destroyers are not directly comparable with the ba ships, but they are comparable with each other, and the curves i cate the general efficiency of the propulsive machinery of represe tive destroyers, and strikingly show the economical advant secured by use of gearing.

STEAM USED BY AUXILIARIES

25 A number of tests have recently been made to determine percentage of total steam used by the auxiliaries of naval vessels, following are given the approximate values obtained from averag these results:

Capital vessels....	Speed, knots...........	10	15	19	21
	Per cent of total steam..	38	30	18	14
Destroyers........	Speed, knots...........	12	16	20	25
	Per cent of total steam..	27	21	17	13.5

26 It will be noticed that the percentage of steam used by auxiliaries in the case of the destroyers is considerably less than used by the large vessels. This condition accounts for the fact the destroyer machinery proves to be more efficient than that of

apital vessels. The above are average results for certain average conditions, but it is believed that the steam consumption of the auxiliaries can, by special care and attention, be reduced from 10 to 25 per cent below the average conditions which obtained when the water consumption was measured.

27 The auxiliary exhaust that is not used in the feed heaters is used in the low-pressure turbines, so that under the conditions of actual operation the auxiliaries do not use the full extent of this percentage, and all over about 10 per cent is used in the low-pressure turbine, where it will give approximately half the return that the same amount of live steam entering at the throttle would give. Therefore, strictly speaking, the auxiliaries, except electric generators, should be charged with only about half of the above percentages, since they exhaust at about 10 lb. pressure, and this exhaust is used effectively both in the feed heater and in the turbines. The following table has been compiled to show the approximate steam consumption of various auxiliaries on large vessels for full-power conditions. The amounts of steam used are quite variable and depend not only upon the design of the auxiliaries but also on the method of operation.

	Per cent of total steam
Feed pumps	3.0 to 5.0
Circulating pumps	1.5 to 2.5
Air pumps and condensate pumps	0.2 to 0.6
Forced-draft blowers	1.5 to 3.0
Lubricating pumps	0.1 to 0.2
Oil-fuel pumps	0.2 to 0.4
Total for auxiliary machinery necessary to operate main engines	6.5 to 11.7
Bilge pumps, ice machines, steering engine and other miscellaneous machinery	1.0 to 4.0
Dynamo plant other than forced-draft blowers	1.0 to 2.0
Total	8.5 to 17.7
Evaporating plant, depending on rate of use.	

28 The principal steam users among the auxiliaries are the feed pumps, circulating pumps and forced-draft blowers; the other auxiliaries use a relatively very small percentage of steam.

29 The introduction of turbine-driven centrifugal feed pumps will reduce the steam consumption for feeding. The consumption of circulating pumps is reduced by the use of scoop injection for condensers on destroyers and scouts, and on other vessels where this is racticable. The consumption for forced-draft blowers is reduced by sing larger blower units and by having arrangements made whereby ro blower units are compounded. Compounding has actually been ade use of in several instances.

30 Electric forced-draft blowers have been used on a considerable number of large vessels. While this practice is fairly economical, the desired speed regulation is not secured, and electrically driven blowers are not well-suited to oil-burning installations. The objections to the electrically driven blowers are:

a Difficulty in obtaining as variable a speed as is desirable and necessary with oil fuel. With steam-turbine drive minute regulation is possible.

b Additional generating units have to be started when a considerable number of blowers are started.

c Starting and stopping of blowers creates a very variable load, and in order to accommodate this load extra generators have to be kept in operation.

d Electrically driven blowers cannot be efficiently operated at low speeds, and in many cases a low speed is desirable to suit the boiler output at any one time.

e Circuit breakers are likely to blow, and if this happens flare backs may take place and all steam generation be stopped until the blowers are again started. With steam-turbine drive it is very unlikely that all blowers would stop at the same time.

31 It must be remembered that the auxiliary exhaust is now invariably used for feed heating and in some instances for distilling, so that the actual heat used by the auxiliaries is really from 4.5 to 9 per cent of the total instead of from 8.5 to 18 per cent.

32 The auxiliaries can, by lack of care in design, condition and operation, become very wasteful, so that they do, in cases, use two or three times the amounts stated. But in our best performances we are reducing the percentage ot available heat used by the auxiliaries to approximately 5 per cent of the total at full power. By improvement in design and operation this figure can be still further reduced, but no very marked further saving in steam used for auxiliaries is practicable.

NEW TYPES OF MACHINERY

33 In the search for economy and efficiency there are new types of naval machinery to deal with which are causing both the directly connected turbine and the reciprocating engine to be superseded. These are the geared turbine and the electric drive. The hydraulic transmitter has also been considered abroad. These innovations, which have actually been tried in quite a number of cases both on

naval vessels and in the merchant marine, will secure an advantage in economy of 10 to 15 per cent as compared with the directly connected units, as well as a material saving in weight. In this connection note the position of the curves of the *Wadsworth*, our first U. S. geared destroyer. For certain types of vessels the geared turbine has the advantage over the electric drive. For other types the reverse is the case. For large capital vessels of high speed the electric drive appears to have the advantage for the following reasons:

a There is much more flexibility so far as arrangement and shaft and piping connections are concerned.

b Better economy at cruising ranges is secured.

c Generator rooms, boilers, piping, etc., can be more adaptably arranged. The necessity for bringing large steam pipes through boiler compartments and back to the engine room is avoided, electric conductors taking the place of these large pipes.

d Parts of machinery can be undergoing repair while other parts are in operation at a very large percentage of the total power of the vessel.

e Much better handling conditions are secured, and a greatly improved power for backing is present without the complication caused by the use of backing turbines.

f It is entirely practicable to use a high degree of superheat.

GEARING VS. ELECTRIC DRIVE

34 Gearing is more suitable than the electric drive for light, high-powered vessels of the destroyer or scout type, because it is more compact and, within the limitations of these vessels, is lighter than the electric drive. It is a little more economical at high power but not as economical at cruising and intermediate speeds. Gearing is more suitable for light, high-powered installations where weight and space are cut to the minimum and where the power is not so great but what the leads of steam piping and shafting can be satisfactorily handled. The turbine-reduction-gear layout does not permit of as much protection to the machinery, and the handling arrangements and flexibility are not as good. In the scout and destroyer designs, machinery of the following characteristics may be applied: Powers up to 25,000 shaft h.p. on one reduction gear set, two pinions being utilized. Water consumption of 10.5 to 11 lb. per shaft h.p. at full power; machinery weight, total, about 40 lb. per shaft h.p. Such machinery contemplates the use of scoop injection condensers, and

also the possible use of steam ejectors or kinetic tanks in place of air pumps. It also requires the reduction of all auxiliaries to the lowest practicable limit.

35 A comparison of gearing and electric drive on various points is given below. The advantage is not all one way, and the special conditions of vessel and her service requirements will govern the choice of one type or the other.

GEARING	ELECTRIC DRIVE
WEIGHT	
The initial advantage rests with the gearing, since the gear will not weigh as much as the generator and motor combined. As very large powers are used and a multiplicity of shafts, this advantage is lessened.	The saving of weight by using gearing will, however, not be a very large percentage of the total machinery weight, probably not more than 1 or 2 per cent. However, considerable weight in piping is saved by means of the electric drive, so that for large vessels the weight of machinery is about the same for either gearing or for electric drive.
SPACE	
The advantage will rest to a slight extent with gearing, so far as the cubical contents are concerned.	The space can, however, be in many cases more advantageously utilized. The generator can be placed at some distance from the motor. The shaft can be greatly shortened. This advantage shows up specially on large vessels. Superior boiler-room arrangements can be utilized.
RACING	
The gear and turbine will be subject to the racing action of the propeller.	There will be no racing. The motor will govern the propeller and maintain even speed. This fact may possibly produce a slight gain in propulsive efficiency.
HANDLING	
Handling is satisfactory.	The advantage rests with the electric drive, especially with regard to backing power. See Jour. A.S.N.E., February, 1916, Stopping, Backing and Turning Ships, by Lieut. S. M. Robison, U. S. N.

Adaptability to Best Conditions for Efficient Propulsion

The gear ratio having been adopted it cannot be changed. If the effective horsepower and propeller efficiency have not been accurately estimated, an improper ratio will be present which may prevent the turbine from securing its most efficient speed. Unless very high gear ratios are employed the most economical turbines cannot be used.

The relative speed of propeller and turbine can be altered to any extent. The turbine can therefore run at its most efficient speed without reference to the propeller speed. This enables a very-high-speed turbine to be used, which allows of greater economy. The design can utilize the very best propeller conditions without affecting the design or limitations of the turbine.

Facility for Repair

No special advantage. Whole engine plant has to be run when under way. At present large gears can only be cut by very few firms.

In case of accident generator and motor parts can be renewed more easily than gear can be repaired. One generator can be repaired while other works to full power. This will give about three-quarter speed.

Repairs to be Expected

Repairs to gearing will be little or nothing; but if damaged to any extent, however, gear will have to be replaced. A spare gear should be at hand to replace one damaged by accident.

The repairs to generator and motor may be of considerably greater extent and may cost more than those to gear, but they can be more readily made, and actual experience with the *Jupiter* does not indicate that there will be large repair bills.

Economy

With the same boilers and turbines gearing has an advantage of about 5 per cent at full power. The same turbines may, however, not be feasible, since the turbine to be used must accommodate itself to the gear that is being used.

Owing to ability to use only one generator at below half power, the ability to run generator at practically constant speed and the flexibility of the electric bond to adjust itself to any speed ratio, it is likely that the initial advantage of gearing in economy will disappear at cruising speeds, and at low speeds the electric drive will be more economical.

Reversibility — Backing Power

Requires a backing turbine, which complicates design and enables only a portion of full power to be used for backing. Limited to about 50 per cent backing power, even with very large backing turbines.

Full backing power is secured without disturbing turbine. Reversing is made particularly easy, and the full utilization of the steam supply is possible for backing, since there is about the same engine efficiency backing as going ahead.

Care in Operation	
Little or no special care required for the gear. Oil supply the essential matter. Some difficulty in lubrication has been experienced.	Considerable attention will have to be given to maintaining generator and motor in condition. This also requires a special knowledge of electrical features, for which personnel will have to be specially trained. Ventilation of motor and keeping it dry will require special attention.
Facility for Installation	
Location of turbines and gear fixed by position of shaft.	Generators can be located anywhere without regard to shaft. Best location of condenser, steam piping, etc., can be secured. This enables a more adaptable and simple arrangement to be made. Special advantage of this may be taken in making best provision for torpedo protection.
Certainty of Data on Which to Base Design	
Gears suitable for very large power have not yet been built in this country. Each installation practically necessitates a new design of the gear. Existing installations have been generally successful, but some trouble with wear on gear and breaking of gears has been experienced.	All portions of the electric-drive apparatus have been built and fully tested out. They have been applied to one marine installation and have been successful.

36 The Ljungström turbo-dynamos are a new feature in marine electric driving, but though there has been an installation actually built, the device is still experimental. If it stands up in service it will be a means of improving the now accepted type of electric drive in this country by about 10 per cent. Nothing definite has been done with this device in the U. S., though a great increase in economy is claimed for it and has been secured in numerous actual installations.

BOILER EFFICIENCY

37 The campaign for economy also makes it necessary to inquire into possible ways of improving boiler economy. Boiler efficiency has remained approximately at a standstill for quite a number of years, but the advent of oil burning has given a new phase to the situation. By means of improved burners continual improvement in combustion has been effected. A great deal of experimental work has been done by the fuel-oil-testing plant at the Navy Yard, Philadelphia, and marked improvement in burners has been secured.

38 It is believed, however, that we are about to secure a material improvement by arranging the heating surface of boilers so as to secure a proper relation between the flow of gases and the circulation of water. The actual course of circulation in water-tube boilers is a matter that bears study and investigation. A recent disclosure has shown that some generally accepted ideas are in error. It is believed that to obtain the best efficiency the gases must be so directed as to pass in the counter direction to the flow of water in the tubes, and that as the gases are cooled and the volume reduced the area of the gas passage should be decreased in order that the velocity of the gases may be maintained at a proper point for the requisite efficiency. Some experiments along these lines are under way.

39 The best boiler results that we have record of give from 77 to 81 per cent efficiency for well-baffled straight-tube boilers (with 0.1 to 0.3 lb. oil per sq. ft. of heating surface), and about 70 to 78 per cent for express boilers. With coal the boiler efficiency averages about 5 per cent less. It is believed that by special care based on proper experimental data a boiler efficiency of 85 per cent can be secured for continuous operation on board ship; this economy to be effected by means of proper baffling and proper relation of baffling to boiler circulation, and by supplying more efficient clothing and lagging. An oil furnace means a higher furnace temperature, and hence more boiler clothing is needed.

SPECIAL CONDITIONS OF DESIGN

40 The demands upon the engineer to secure in the new designs the necessary power for a great increase in speed, as well as for special and unprecedented protection from torpedo attack, have made it necessary to get every bit of power out of every pound of weight and cubic foot of space allowed. Everything non-essential must be left off, and every part must be designed to be adequate but without excessive margin. The principal method of saving both weight and space is to install machinery that will secure the highest practical economy, and in searching for the means of accomplishing this the naval engineer is looking to the engineering talent of the country to come to his assistance.

NEW DEVELOPMENTS IN NAVAL ENGINEERING

41 The present time is especially a period of development in naval engineering. The European war, and the developments caused by it, have emphasized and will tend to emphasize many new features

and point out special lines of advancement in naval engineering. We are now almost at the stage of discarding the reciprocating engine and the direct turbine drive for new fighting craft, and in their place are coming turbine reduction gear and the electric drive, with the oil engine for submarines and as a possibility for some auxiliary vessels. Coal burning is being superseded by oil burning on nearly all classes of vessels building for our Navy. There have been recently successfully tried three superdreadnoughts, the *Nevada*, *Oklahoma* and *Pennsylvania*, all of them equipped with oil-burning boilers exclusively. This marks a special step of progress in the evolution of naval engineering work, and the use of oil fuel has permitted many engineering and operating advantages to be secured. Special development is taking place in the field of aeronautical apparatus, and a great deal of this is being done by the coöperation between the Navy and the engineering industries of the country. Many new developments of condensing apparatus, pumps and other auxiliaries are coming on. New things are dawning all the time, and as fast as they give promise of practicability, reliability and proper adaptation to naval use they are tried out and adopted. There is continual progress toward greater efficiency, greater economy, compactness, and in reduction of weight, which enables the United States to take the lead in engineering development. Such developments are largely the result of coöperation between the industrial engineers and the naval engineers. Further to develop this, more experimental and laboratory work is necessary, and the country must be willing to spend some money for experimentation in order to save in actual service and to secure greater efficiency.

42 The Navy has now in service one destroyer, one destroyer tender and one submarine tender equipped with Parsons turbine reduction gear, and one collier equipped with Westinghouse reduction gear. There are building four destroyers with machinery of this type, three with Parsons turbines and one with Curtis. Geared cruising turbines are being used on fifteen destroyers and five battleships built and building. The electric drive has been tested out on one auxiliary vessel and is being installed on three first-class battleships building. In the matter of electric drive for propulsion of ships our Navy is the pioneer. This is entirely an American development, and the men responsible for its initiative and its development to a successful point are all Americans. The Navy Department has materially assisted in this development. An experimental Diesel engine, the largest yet to be installed on a vessel, has been completed at the Navy Yard

New York, and is now being installed on the oil-fuel carrier *Maumee*, which has been towed there from San Francisco. This two-cycle Diesel engine, of 5200 h.p., was built by the Government. The tender *Fulton* has been equipped with a two-cycle Diesel engine and has been in service for over a year.

43 Among the new things in the auxiliary machinery of naval vessels, it may be mentioned that the small steam turbine has been developed as the accepted drive for forced-draft blowers on nearly all our naval vessels. In this also the United States has been the pioneer. The use of gearing for turbine-driven generators, pumps and blowers has also been initiated. Rotary pumps are being used for oil-fuel service.

44 *Condensers.* The principal improvement in condenser practice has been the adoption of expanded tubes in place of packed tubes. Expanded tubes can be kept absolutely tight, while there is always some doubt about a packed tube. For some years expanded tubes have been used on destroyers. For large vessels the tubes have been expanded at inlet end and packed at outlet end. This prevents crawling of tubes.

45 Augmenters, dual air pumps, or some kind of dry vacuum apparatus are installed. Tube spacing to allow of least possible resistance to flow of steam is being employed.

46 Steam ejectors in combination with a condensate pump have been experimented with and their use is being considered. These ejectors save a great deal of weight and space, but they will have to be thoroughly tried out before they can be generally applied.

47 The importance of high vacuum is fully appreciated and no effort is spared to obtain all possible advantage within the limitations of the design. On turbine vessels the design contemplates securing a vacuum of about 28 to 28½ in. of mercury at full power, with circulating water at 70 deg. fahr. At lower powers a better vacuum is obtained. On destroyers and scouts scoop injection condensers are being used, with a small auxiliary circulating pump to furnish circulating water for backing and while vessel is standing still or maneuvering.

48 *Feed Pumps.* Turbine-driven multiple-stage centrifugal feed pumps are being installed on our latest vessels. It is expected that these pumps will be more reliable, furnish a more steady pressure, and be considerably more economical, especially in repairs, than the reciprocating plunger pumps that they supersede. It is also expected

that the centrifugal pumps will require far less overhaul and will lessen the danger of rupturing feed piping.

49 *Rotary Pumps* are being adopted for fuel-oil service. These pumps give a more steady pressure.

50 *CO_2 Refrigerating Machines* are being used in place of dense-air ice machines. Some of the later machines are electrically driven, but there is no very special advantage in the electric drive. Small electrically driven refrigerating machines of several types have been developed for use on destroyers and small craft, these being similar to machines used in yachts and private houses.

51 *Distilling Plants* of multiple-effect, low-pressure (operating under a vacuum), non-scaling type are being fitted, and auxiliary exhaust is being extensively used for operating evaporators. These steps have resulted in materially reducing the fuel consumption in port. The use of feed heating in connection with distilling plants is being extended and consequent gain in economy realized.

52 *Clutches* for connecting or disconnecting cruising units or motors while shafts are revolving, have been experimented with and installed in a number of vessels; some are hydraulic oil clutches, others magnetic. A thoroughly reliable clutch for this service is very desirable, and from the different kinds tried it is hoped that several will be developed beyond the experimental state in the near future. The clutches used are the Metten clutch, operated by oil pressure, the Turner mechanical clutch, and the magnetic clutch manufactured by the Cutler-Hammer Co.

53 *Forced-Draft Blowers.* Tests and experiments have been carried on for several years in order to develop forced-draft fans for destroyers that will be both strong and durable, and which will operate as noiselessly as possible. This is an extremely important matter, as the noise of these fans is one of the things that will disclose the presence of a destroyer to an enemy; it also indicates lack of efficiency.

54 *The Gyroscope,* besides its entirely successful application in the gyroscopic compass, which is now installed on all naval vessels of fighting value and which has made submarine navigation practical and reliable, is also being used in numerous other fields, such as a stabilizer for air craft, submarines and for other naval craft.

55 *Radio Apparatus.* In the field of radio development tremendous strides are being made. The range of apparatus on naval vessels has been doubled. Trans-Pacific stations are being built to connect the radio chain across the Pacific from San Diego to Cavite

via Pearl Harbor. The wireless telephone is also being developed in conjunction with the radio stations.

STANDARDIZATION OF ENGINEERING APPARATUS

56 Another vital matter involving naval engineering preparedness is the standardization of engineering methods, tools, gages, fittings, parts, etc. Comparatively little has been done in this matter, and the difficulty seems to be to get some one or somebody to start things. The American Society of Mechanical Engineers and other engineering societies have labored in this field, but definite progress is not satisfactory.

57 Standardization of mechanical apparatus is essential to the maximum efficiency of the naval engineer. Whenever standardized commercial articles can be used the Department is desirous of so doing, as a standard commercial article is (1) more quickly obtained; (2) allows increased competition; (3) will be cheaper; (4) will necessitate carrying a smaller stock of material; (5) and can be obtained readily when away from the source of supply.

58 Steps towards standardization have been made along the following lines: (1) Boiler-tube specifications of the American Society for Testing Materials, The American Society of Mechanical Engineers, and the Railway Master Mechanics' Association are recognized by law in several of the states; (2) spark plugs for gas engines are fast coming to a standard size with a standard screw thread.

59 The Navy Department has always been in favor of standardizing mechanical apparatus, and of late years we are trying hard to fix matters so that the best commercial practice in any field will be suitable for naval work. Standardization in many particulars is essential, and the manufacturers and mechanical engineers of the country should be making special effort to secure further standardization of methods, tools and parts. In the end the engineering industry of the country will benefit by this standardization, and engineering material and apparatus can be more economically supplied.

COÖPERATION OF ENGINEERING ACTIVITIES

60 The naval engineer, both in and out of the service, has many new and interesting problems before him, and it behooves him to be looking to the future to employ the best that science and mechanical ability can provide for fashioning the weapons of naval preparedness. The best development of naval-engineering matters will be accelerated

by the best possible coöperation between the naval and the industrial engineering talent of the country. Without this hearty coöperation and the support of the engineering facilities of the country, the Navy could not be properly placed or kept upon a war footing. In fact, without the utilization of the civilian engineering talent of the country we would not get very far with our material for the Navy. The Navy appreciates the need of this coöperation, and desires that our commercial engineering forces coöperate in every possible way to give it the best material that American engineering talent can devise and that its mechanical forces can fabricate.

61 But not only should there be coördination between the civilian engineer and the engineer in the Navy, but there must also be that coöperation and standardization of engineering means and efforts that this Society aims to foster and in which it has been so markedly successful.

No. 1548

THE DESIGN AND TEST OF A LARGE RECLAMATION PUMPING PLANT

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Non-Member

Until recently reclamation work along the rivers of the state of California has been accomplished more or less successfully in the small districts. During the last few years, however, there has been a very rapid development, with a decided tendency towards forming very large districts. To accomplish true reclamation it is necessary not only to exclude the flood, or overflow, waters, but also to provide a drainage and pumping system which will insure the removal of all surface water, whether from rainfall or seepage, as fast as it accumulates. In the smaller districts the matter of drainage is not of very serious moment, but in a very large district the amount of drainage water to be handled becomes enormous, and on account of the very great loss involved by a failure to remove this drainage water as fast as it accumulates, makes it imperative that the drainage and pumping system be designed in accordance with the best engineering practice. In this paper the writer will outline the problems involved and a solution of them in the case of one of the largest reclamation districts in the state of California, and which has resulted in the construction of the largest reclamation pumping plant in this country, having six 800-h.p. units.

2 In November, 1913, the writer was given the following problem: — Given a reclamation district located at a definite point in the Sacramento Valley; proceed with the design and construction of a drainage pumping plant that, when operated in connection with a certain drainage and levee system, will insure the removal of all accumulated drainage water. No restrictions were placed on the

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writer and all details were decided on their respective merits. The district under consideration is known as Reclamation District 1500, created by act of the state legislature in 1911, and commonly called Sutter Basin. Its geographical location is shown in Fig. 1. Lying as it does between the Sacramento and Feather Rivers at their junction, it forms a large basin, with the lowest point nearly in the center.

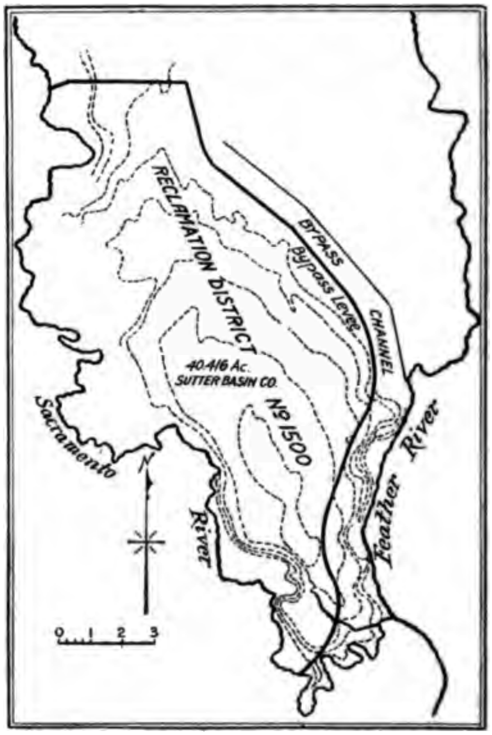

FIG. 1 SUTTER BASIN, RECLAMATION DISTRICT No. 1500

Its geographical location makes it subject to the full sweep of the flood waters of the above two rivers.

3 The District has an extreme length of approximately 19 miles, a maximum width of approximately 10 miles, and has an area of approximately 66,000 acres. The elevation of lowest point in basin is 19 ft. above datum, and maximum elevation of 30 ft. above datum. As the average maximum elevation of the flood waters at this point is approximately 39 ft. above datum during the winter time, the entire district is under water with varying depth up to nearly 20 feet.

4 To reclaim this tract of land it is necessary to build levees

around its entire circumference, about 70 miles, to exclude the flood waters of the rivers. The natural drainage of this tract is towards the center from the east and west and from north to south, terminating at the southerly end of the district in what is known as Sacramento Slough, which empties into the Sacramento River near its confluence with the Feather. On account of this natural condition it is obvious that this district should have only one drainage pumping plant, and its location should be near the point where the levees cross Sacramento Slough. This slough has now been utilized as part of the main drainage canal of the district. This canal extends in a general northerly direction through the center of the basin.

5 The essential steps in the determination of the size and character of the required drainage pumping plant, and its design, may be grouped as follows: (*a*) Determination of amount of drainage water to be pumped; (*b*) Pumping head; (*c*) Physical design of plant; (*d*) Construction; (*e*) Test and operation. The above steps will now be discussed in detail.

AMOUNT OF DRAINAGE WATER

6 As stated above, the amount of water to be handled by a reclamation pumping plant is made up of the run-off from rainfall and seepage through the levees or any other underground channel. The former, while influenced by many quantities impossible to predict precisely beforehand, such as rate of rainfall, duration of rain, state of land cultivation, etc., can, however, be approximated with a reasonable degree of accuracy. Since to produce true reclamation all drainage water must be pumped out of the district as fast as it accumulates, therefore the capacity of pumping plant will be determined by the *maximum* rate of accumulation of drainage water which has to be handled; otherwise a portion of district would become flooded. While the total amount of water to be handled may be made up largely of seepage, the maximum rate of accumulation is almost entirely determined by the run-off from rainfall, as explained later. It is, therefore, very essential that a very careful study be given to this phase, and as the maximum run-off may be expected at maximum rainfall, the essential point is to determine the conditions of run-off for this maximum rainfall.

RUN-OFF FROM RAINFALL

7 There has been in the past no actual observation of rainfall within this district. Therefore, the rainfall has been estimated from

the two nearest points of record, Marysville, located at the east and north on the Feather River, and Knights Landing on the west, just across the Sacramento River, giving twice the weight of the Knights Landing readings over those at Marysville. As these readings are a matter of record they are not given here. From an analysis of the estimated rainfall thus made, the maximum rainfall for any of the years of record occurred on January 12, 13, 14 and 15, 1911, when rainfall was 1.04 in., 0.95 in., 2.53 in. and 0.64 in., respectively. Other periods of severe rainfall which might be mentioned here are, December 12, 13 and 14, 1906, where 0.6 in., 2.37 in. and 0.37 in. of rain fell

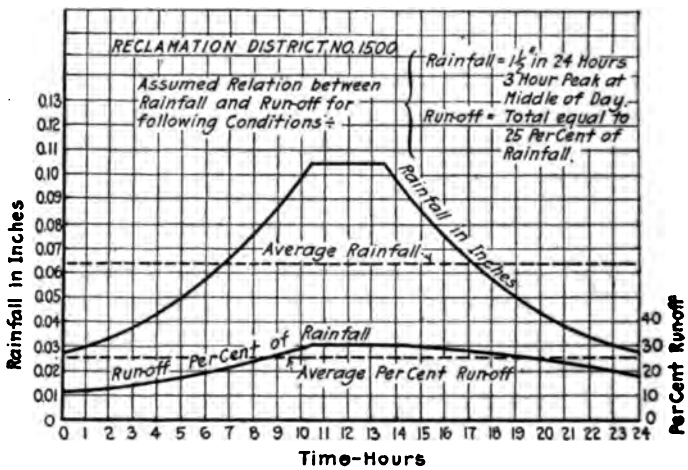

FIG. 2 ASSUMED RELATION BETWEEN RAINFALL AND RUN-OFF

respectively; also December 25, 26 and 27, 1906, with rainfall of 0.22 in., 2.01 in. and 0.20 in., respectively, and February 11, 12 and 13, 1909, with rainfall of 0.87 in., 1.40 in. and 0.69 in., respectively.

8 Rainfall equaling the above maximum values may be expected during any year, and it is not unreasonable to expect that at some time they might be slightly exceeded. However, taking into consideration the fact that they are estimated values, that if the above maximum values as first given are used in connection with a conservative value of percentage of run-off, the result will probably not be seriously exceeded at any time. Therefore, the pumping capacity of plant has been determined from the estimated run-off resulting from the rainfall of January 12, 13, 14 and 15, 1911.

9 The yearly run-off from a district of this character can be

estimated very accurately from the yearly rainfall, but in this case, since there are no storage facilities, we are concerned with the hourly run-off. It is a well-established fact that the run-off varies with many conditions, among which are the rate and duration of rain. In this particular case, as will be seen, the hourly rate of rainfall will have a very material effect on the size of pumping plant. It therefore becomes necessary to assign to the above rainfall a given hourly rate. After careful consideration of this point, Grunsky's method of estimating hourly rate of rainfall was adopted with some modifications.

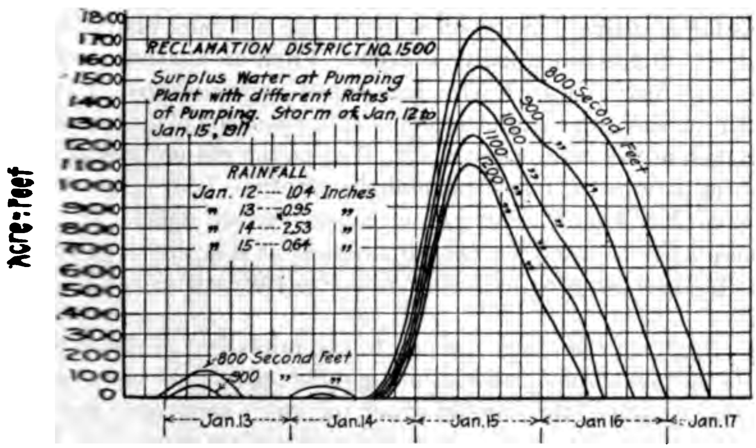

Fig. 3 Surplus Water at Pumping Plant

This, in effect, is that 50 per cent more rain falls in any one period than in the preceding period; that is, the ratio of fall for any period to the next of equal length would be 1 to $1\frac{1}{2}$.

10 A further assumption was made that the rainfall for any one day started at a minimum value at the beginning and increased at the above ratio to a maximum, which lasted for three hours during the middle of the day and then decreased to a minimum at the end of twenty-four hours in the above ratio. This rate of fall for a rainfall of $1\frac{1}{2}$ in. per day is shown in Fig. 2.

11 Having fixed the hourly rate of rainfall, the next step is to determine the run-off and rate of arrival of this run-off at the pumping plant. For purposes of this calculation the total run-off has been taken as 25 per cent of the total rainfall. The rate of run-off has been assumed arbitrarily to vary with rate and duration of rainfall, being greatest for greatest rate and also for equal rates of fall, being

greater at the end of the day's rainfall than at the beginning. This varying percentage of run-off for the rate of rainfall assumed above is shown in Fig. 2.

12 On account of the large area of the district and its extreme length of 19 miles, it is obvious that the run-off from the extreme northern end will arrive at the pumping plant several hours after the rain has fallen. The drainage system of this district as laid out contemplates a very extensive system of ditches. These ditches have been designed for a certain velocity of water and the area drained, depending upon slope and character of soil, so that it may be assumed that as far as delivery of run-off water to the pumping plant is concerned, all ditches are equally effective. From an analysis of these factors, the district was divided into zones, or areas, such that the run-off from each zone might be assumed to reach the plant in a given time, taken in this case as one hour. That is, the run-off of Zone No. 2 has been assumed to reach the pumping plant one hour after run-off from Zone No. 1, etc. There is, of course, certain delayed flow of the run-off before it reaches the ditches, but as it is the same for all zones, its effect may be neglected as far as this calculation is concerned.

13 Applying the above method to the rainfall of January 12 to 15, 1911, as given above, and assuming different rates of pumping at the pumping plant, the surplus water (given in acre-feet) accumulating at the pumping plant, above that pumped, is shown in Fig. 3. Fortunately, the main drainage canal is of very ample capacity, and it has in itself considerable storage capacity, provided that water is drawn down in the canal before rain sets in. Assuming that pumps can draw the water down to 10 ft. above datum, and that no serious flooding takes place until 20 ft. above datum is reached, the storage capacity between these two elevations is approximately 1150 acre-feet. Applying this value to the curves in Fig. 2, it is found that a pumping plant having a capacity of approximately 1000 to 1100 second-feet should be sufficient for the purposes of reclamation, and for this district this value has been taken.

SEEPAGE

14 The amount of seepage water which might be expected in this district is very difficult to estimate. With the character of soi that occurs here, and assuming well-constructed levees, such as ar being built, it is reasonable to assume that the rate of seepage will no be a very great amount. In other parts of the state, where the sub

peaty nature, seepage water often reaches a considerable here are practically no data available upon which to base l conclusion in this matter. It has been variously estifferent engineers that under normal conditions, with subring in this district, the seepage would average from ½ to et per mile of levee, or from 35 to 100 second-feet.

rate of seepage would be somewhat influenced by the duration of the flood waters outside of the levees. This

FIG. 4 EXTERIOR VIEW OF PUMPING PLANT

er is independent of the rainfall and will have a tendency e or less constant flow throughout the flood season, but n value would be relatively small and therefore not have l bearing on the maximum capacity of the pumping plant; effect on the amount of power consumed and the total water pumped may equal or exceed that of the run-off infall. Hence, in determining the capacity of the plant, been neglected.

PUMPING HEAD

head under which the plant must operate is extremely During the months of November and December, before there is any very great rise in the rivers, the head will be a minimum, and in this particular case will be about 10 ft. static. However, when the flood season occurs the plant will be required to operate under its maximum head. The calculation of the maximum flood plane is of very considerable moment, as not only does the pumping equipment hinge on it, but also the entire levee construction is based on the probable flood height. At the present time, with the district open to overflow waters, there was recorded this year a flood height of 39.2

FIG. 5 INTERIOR VIEW OF PUMPING PLANT

ft. above datum. But with the completion of levee system and the confining of the flood waters to a restricted channel, the flood plane will increase very materially.

17 On account of its bearing on the construction of the levee system, the calculation of the maximum flood plane was undertaken in very great detail, and very thoroughly, under the able direction of the chief engineer of the District, George N. Randall, who, from many years' experience with the flood conditions of the Sacramento River, was able to bring a great fund of information and data to assist in determinations of this point. The problem was attacked from every conceivable angle. In general, however, the methods consisted primarily in determinations of the maximum flow of water through the By-pass and Feather River channel. The width of this channel

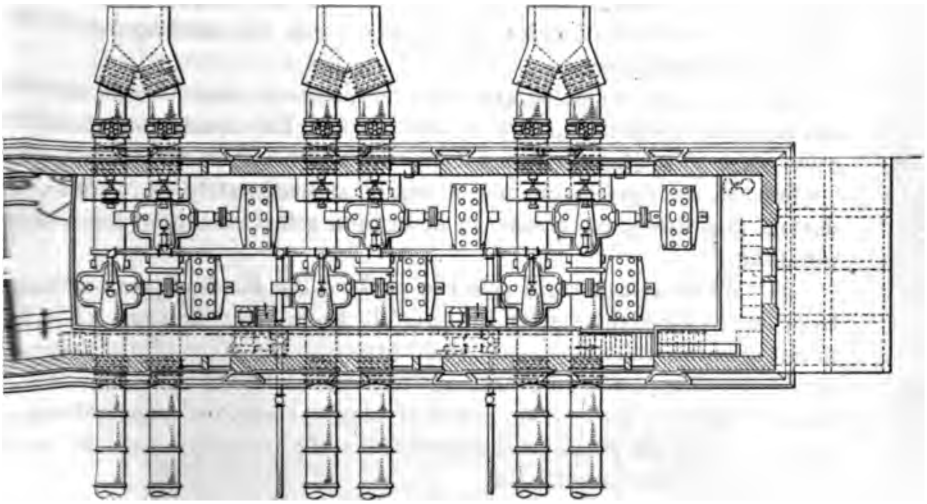

FIG. 6 PLAN OF PUMPING PLANT

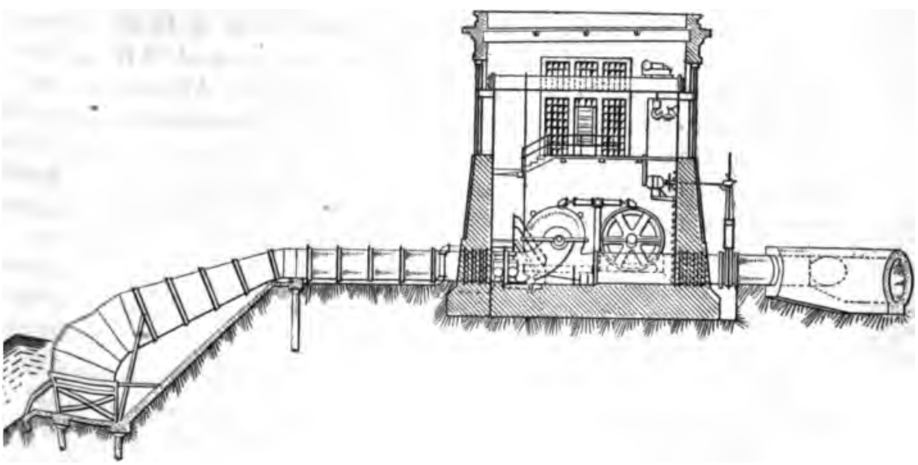

FIG. 7 ELEVATION OF PUMPING PLANT

being fixed and its general slope known from surveys, having estimated the maximum flow, it is a comparatively simple matter to determine the depth of water, or, in other words, the maximum elevation of the flood plane.

18 The data collected are too voluminous to present here, and are perhaps beyond the scope of this paper. The conclusions, however, were that the maximum elevation of the flood waters when confined to the By-pass channel would be approximately 42 ft. above datum, and this value is used in determining the maximum head of pumping.

19 While as a general rule the peak of the flood waters occurs from one to three days after the rainfall, there are conditions which might arise whereby this peak would arrive at the point of discharge of the pumping plant simultaneously with the arrival of the maximum run-off water at the suction sump of plant; therefore conservation demands that the plant be designed to handle maximum amount of water against maximum head.

20 While the above quantities determine the capacity of plant, in order to obtain an estimated yearly consumption of power the same calculations of quantity and head were made for each day of the winter season and also for the six years of record, 1906 to 1912. These detail calculations are not presented here, the estimated power consumption for pumping run-off only being from 700,000 to 1,200,000 kw-hr.

21 With an elevation of discharge water level of 42 ft. above datum and elevation of water surface in suction sump of 16 ft. above datum, the maximum static pumping head is 26 ft. Allowing for an estimated friction loss in discharge pipes of 3 ft., the total maximum pumping head was taken at 29 ft. Therefore, in conclusion, the capacity of the plant will consist of pumps that can handle 1000 second-feet against a 29-ft. head. With this determined, the only remaining feature is the physical design of plant.

PLANT DESIGN

22 The first consideration in the plant design is the motive power. In this connection electric motors, steam engines or turbines, and Diesel engines were considered. The fact that the plant was readily accessible to three power companies, the Pacific Gas and Electric, Great Western Power Company and Northern California Power Company, had a very great weight in forming the decision to use

tric-motor-driven centrifugal pumps, which have the great ad-
tage of lower initial investment, although the operating expense is
tle greater.

23 In order to select the size and number of pumping units, a
iber of plant layouts were made with four, five, six, seven and
t units, and after careful consideration, both of conditions to be
and manufacturers' requirements, a plant having six units, each
-in. centrifugal pump having a normal capacity of 175 second-
against a discharge head of 29 ft., driven by an 800-h.p. electric

Fig. 8 View of Discharge Pipes

or, was selected as offering the most satisfactory arrangement
best economical design.

24 At this point a consideration of the advisability of using
or two-speed motors was taken up. The use of a two-speed
or offers several advantages in a condition such as presented here
ariable head. The two-speed motor, if properly chosen in re-
t to conditions, will produce a higher overall efficiency, which in
i means a lower annual operating cost. On the other hand, it has
ral disadvantages, as higher first cost of motor, increase in cost
uilding on account of requiring more space, and great multiplicity
parts and operating gear. After careful consideration of the
ous advantages and disadvantages a final selection of the one-
ed motor was made, as the saving in annual cost of operation due
slightly greater overall efficiency did not warrant the increased
estment.

BUILDING

25 Inasmuch as the district would be flooded the first yea installing the plant on account of non-completion of levees, an as a protection against damage to pumps from flooding due to pc breaking of levee at high water, the design of the pumping building becomes of perhaps greater importance than the actu

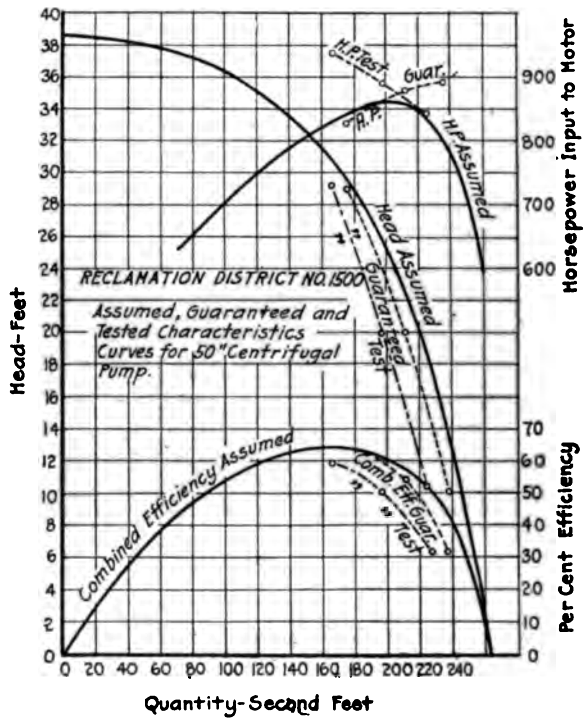

FIG. 9 RESULTS OF PRELIMINARY TEST RUN OF ONE UNIT

chanical layout of the pumping units. In designing the build must be so constructed that its walls and floor slab will safely the water pressure due to floods; also sufficient weight m present to prevent flotation.

26 The building as finally constructed is of reinforced co the walls so designed as to withstand a depth of water on the of 18 ft. above floor level. This necessitated walls 4 ft. 6 in. t the base of the floor line. Under the above water-pressure con the uplift pressure on the bottom of building amounts to near tons. This gives some idea of the problems that had to be f

hey are similar to those of dry-dock construction. On account of he water conditions to be met with, there is no opening into the buildng below elevation 42 ft. above datum, or 18 ft. above the floor line. he building rests on a concrete-pile foundation, approximately 300 5-ft. piles being used. No difficulties of any serious moment were xperienced during construction.

PUMPING UNITS

27 On account of the problems to be met in the building construction, every effort was made in the pump-unit layout to reduce the se of the plant. The writer believes that the layout finally selected

FIG. 10 GENERAL VIEW OF PUMPING PLANT

fers the most compact one consistent with good operating conditons. The units are dovetailed in such a manner that a minimum space is required. Each unit is independent of the remainder, ving a separate steel suction pipe, and the discharge of each group two pumps is brought together outside of building into a single pe. There are three reinforced-concrete discharge pipes, 78 in. in meter. After passing through the levee each of these pipes passes o a valve chamber which has a large wood-and-steel swinging ck valve and also a sluice gate.

28 On the suction sides of the pumps there are no valves of any racter. On each discharge side there is located just outside the ding a 50-in. gate valve, operated by a direct-current motor nted on the inside of building for flood protection, and connected eto by a shaft passing through a stuffing box in the wall; the valves were placed outside to reduce the size of the building. All

units and motors operating the gate valves are controlled from the switchboard located in one end of the building. Priming pumps are installed in duplicate, each capable of priming any one pump in ten minutes. These pumps are of the wet-vacuum type, and may also be used for removing any drainage water inside of building. One rather unusual feature is the installation of a storage battery for operating motors on gate valves. This was done so that the gate valves might be operated quickly if the main source of electric power should fail, and this feature has proved very successful.

29 Electric power is delivered to motors at 2200 volts from three 1500-kva. outdoor-type oil-cooled transformers (60,000 to 2200 volts). These transformers are mounted outside the building on a platform sufficiently high to be out of danger of flood water.

30 The total cost of this plant, exclusive of engineering, is approximately $170,000, of which the building, discharge pipes and sump represent $75,000 and the pumping machinery $95,000. Views of the plant are shown in Figs. 4 to 8 and in Fig. 10.

TESTS

31 The writer had hoped to give at this time results of final tests on units in this plant. However, the preliminary test run has been made on one unit and the results are presented here. These results are not to be taken as final, as the pump manufacturers are to make a slight change in the runner which it is expected will modify considerably the results of the preliminary test run.

32 The specifications covering the pumping machinery provided for the method of test. Guarantees of efficiency were made on the combined efficiency of motor and pump, also with all losses in suction pipe charged to the pump. The electrical input to the motor was measured by means of two indicating wattmeters. The discharge head on the pump was obtained by means of a U-tube of water, the total pumping head being taken as the difference in level between the water in the sump and height of water shown in the U-tube. The amount of water was obtained by means of a Cole and Flad pitometer, two right-angle traverses being made on the suction side of the pump. Care was taken to eliminate errors on account of use of this instrument on the suction side. The combined efficiency was obtained for three different heads, 10, 20 and 29 ft., as provided in the specifications. The results of the preliminary tests on one unit are shown by the curves in Fig. 9. In this figure are also shown estimated curves used by the writer in his original calculations, and also guaranteed

values. It is expected that changes in the runner, as proposed by the manufacturer, will bring about a closer harmony between guaranteed and actual test results.

DISCUSSION

James T. Whittlesey asked that the author state the value of the annual load factor of the plant described in the paper.

Thomas Morrin, discussing the matter of seepage, said that in one drainage district he had in mind the dry face of the levee was periodically mud-washed, and that this procedure was effectual in stopping all seepage for a year and a half or more.

Charles T. Hutchinson asked whether the author had taken into consideration the amount of evaporation during the period of the main flood.

W. B. Gregory (written). The paper is of great interest because of its points of similarity with drainage problems of the Gulf Coast country and especially because of its points of dissimilarity. The heads are much greater than in Louisiana and Texas and are more in line with those found in the reclamation projects of the Mississippi Valley.

With a normal head of 29 ft., perhaps one may leave out refinements in the discharge pipes. While the suction pipes seem to have been carefully designed, the discharge pipes as shown in Figs. 7 and 9 are not so easy to understand. In Fig. 8 there is shown a condition where the water level is at least 6 ft. below the bottom of the discharge pipe. In Louisiana it is usual to design the piping for drainage-pumping plants so that it forms, with the pump, a siphon; also to enlarge the discharge end until the final velocity is low. In this way the pumps, besides overcoming entrance losses, friction and discharge losses, have only to elevate the water through a height equivalent to the difference of level of suction and discharge basins, and the energy thrown away at the discharge is small.

When each of the units is pumping 175 sec-ft. the velocity in the 50-in. discharge pipes is 12.85 ft. per sec. and the velocity head about 2.56 ft., while at the final discharge, which is 78 in. in diameter, the velocity is 10.5 ft. per sec., corresponding to a velocity head of 1.72 ft. It would be interesting to know what the mean head pumped against amounts to or what it was assumed to be in designing the plant. Since for lower heads than 29 ft. the discharge velocities would be greater than those given above, the losses in the discharge pipe, whether running the pumps singly or in pairs, must be a fairly large per cent of the total head. It would seem to be worth while to have carried each discharge pipe down to such a level that the end would be water-sealed except possibly at extreme low water. It would also seem desirable to have large end areas and consequently low discharge velocity.

Each discharge pipe is provided with a 50-in. gate valve. Could not these expensive valves have been eliminated by carrying the discharge pipes over the levees instead of through them, and are the advantages of a straight pipe with a gate valve sufficient to justify the added expense?

It is evident that conditions are quite different in California from those in the South, also that many details necessary to a thorough understanding of the problem are omitted. For these reasons the above questions are raised in order to clarify the problem.

THE AUTHOR. Answering Mr. Whittlesey I would say that the plant operates only six months in the year and that the seasonal load factor varies from six to nine per cent, depending on the rainfall, which values may be brought up to from 10 to possibly 20 per cent by seepage.

As to the matter of evaporation, which Mr. Hutchinson brings up, I found that the evaporation between November and May was not large enough in amount to have any particular influence on the capacity of the plant.

If sufficient care is taken in the construction of a levee and proper soil conditions prevail, seepage can be cut down to a very small amount. One great cause for seepage, however, is gopher and squirrel holes, with which the upper portion of a levee will become honey-combed during the summer months. Before the flood plane has risen to their levels, therefore, the animals should be killed or driven out and the holes filled in.

Referring to Mr. Gregory's comments, I would state that in the particular plant under consideration the average pumping head has been estimated to be approximately 16 ft.; and that during the normal operation of the plant only three units would be running, one on each discharge pipe. Conditions of rainfall and run-off would be such that the occasion for operating more than three units would be comparatively rare. Therefore the design of discharge pipes has been governed to some extent by the normal operation — in other words, the conditions of operation which call for the greatest expenditure of power for pumping purposes. Further, the diameter of the discharge pipes is governed by the relation between the power cost and construction cost. Calculations in this particular case showed that the most economical annual cost, including operating and depreciation interest charges, was obtained with pipes approximately 78 in. in diameter.

During the winter months, when this plant will be in operation, the suction pipes will be under water. In Fig. 8 of the paper the water outside of the levee is at the end of a very dry season, and at a normal low period. It might be said that at no time during the ordinary course of operation of the plant will the suction pipes be exposed, as shown by this illustration, therefore the discharge pipes may be said to have a complete water seal.

In reference to advantages and disadvantages of carrying pipes through or over the levees, this question was gone into rather carefully, and it was decided to carry the pipes through the levee, primarily on account of construction difficulties. At the time the plant was installed the levees at this point were not completed, therefore pipes were very easily carried through levees and supported on firm ground. To carry pipes over levees would have meant considerable trestle work and would interfere seriously with the building of levees, particularly with additions to levees, which are necessary as time goes on, on account of wasting away due to action of the elements and other causes.

The installation of gate valves on the discharge pipes was not governed by whether pipes went through or over levees. The valves were installed for two reasons: (1) To reduce priming difficulties on account of the extremely large volume of discharge pipes, and (2) as a safety factor in case of breakage of pumps, etc., which might cause serious flooding before sluice gates and check valves could operate on the extreme end of discharge pipes.

[illegible]

[illegible]

[illegible]

No. 1549

STANDARDIZATION OF POWER-PLANT OPERATING COSTS

By Walter N. Polakov, New York, N.Y.
Member of the Society

The cost of manufacturing power or any other commodity, being the result of numerous factors involved in production, is in the end the chief criterion upon which the market price, range of use, legislation, future developments, social welfare, etc., depend. Few if any of these questions can be intelligently answered from a knowledge of the actual costs, owing to the effect of an unknown factor — the degree of perfection of the actual performance. The importance of an accurate knowledge of the meaning and significance of actual cost data to the financier warrants the development of a method whereby a cost report tells:

a What the power costs
b What it should cost
c Where the loss has occurred
d Why the loss has occurred.

These questions answered, elimination of waste is a comparatively simple engineering problem.

2 Standardization and predetermination of the cost of power production have never before been considered as possible undertakings, and their advantages were thought questionable. Predetermination of operating costs has not been made use of for other than estimates of probable future expenses prepared by promoters or contractors. These estimates are usually based, as is always the case in work of such nature, either on past performances modified by expectations, or on data obtained from the actual performance of another plant considered as similar. The accuracy of such estimates depends at least on the following conditions:

Presented at the Annual Meeting, December, 1916, of The American Society of Mechanical Engineers. Contributed by the New York Section.

a How reliable the cost records used were
b How near the possibilities were realized
c How close the similarity of the equipment of the plants under consideration is
d What effect the location has
e What effect the nature of load has
f What effect the labor market produces
g How completely the future factors were foreseen

Since, however, there is no assurance that in the *sample plant* the operating methods are perfect, it is not reasonable to expect that another plant is in every respect identical to the *sample plant*, and the value of such a guessed operating cost is highly problematical.

3 Realizing, on the other hand, that such estimates are inevitably colored by the personal sympathies and prejudices of the estimator, a demand for a class of disinterested counsellors has been created. As the financial and not the thermodynamic side of the question is more vital to the investors, the predetermination of the results of power production is often entrusted to public accountants who may or may not be fully equipped to account thoroughly for the influence of such factors as chemical and physical properties of available fuels on the efficiency of boilers and furnaces, the effect of load and machine factors, water rate of turbines under the predominating condition of load, rôle of power factor, wattless current, phenomena of electric transformation, transmission, drop of voltage in distributing lines, and numberless other factors affecting the cost of current either directly or indirectly.

PREVIOUS EFFORTS TOWARD STANDARDIZATION

4 The urgent need of a dependable measure for the financial efficiency of operation has prompted managers and owners to compare their operating-cost data with those available from other plants. An attempt to decide whether one's operation is as economical as possible by comparing it with operating data from other plants whose equipment and service are more or less radically different, would be absurd were it not for the want of a better method. Table 1 presents an example of such a footless effort to make use of cost data by comparing the monthly cost reports of seven central stations. Their equipments are widely different; no two of them use the same grade of coal; the arrangement of machinery requires in some cases double the number of attendants; one is generating electric current for a trunk railroad, whereas others supply suburban and tunnel traffic or

even private consumers, with consequent differences in the characteristics of current, distribution of load and peaks during the day, etc. Under such circumstances, to say from those data that one is operating more efficiently than another is at least too presumptuous.

5 The most interesting attempt to devise means for more rational cost studies was offered by Messrs. H. G. Stott and W. S. Gorsuch (Transactions A.I.E.E., 1913, Vol. II, p. 1619). Yet the use of various factors tending to compensate for differences in fuel, load factor, labor cost, etc., is evidently inadequate. Prices per heat unit in the fuel, if adjusted, do not distinguish the differences in cost between the inherent efficiency of the boilers, furnaces and stokers and the methods of firing or the personal element of attendance. The effect of load-factor corrections is entirely offset by existing differences in the water rates of the turbines and the influences of the auxiliary apparatus in the plants under comparison. Establishing pay-roll correction factors on the basis of mere pay rates is erroneous, because of the size of units, floor plans, automatization of certain operations, etc., requiring more or less men, not to mention the fact that generally low pay to attendants results in high cost per kilowatt-hour, usually on the coal item and often in maintenance. Even if these factors of correction were unquestionably correct, this method leaves the effect of the supremacy of the equipment efficiency unseparated from the efficiency of the methods of management. Finally, even if all factors are fully accounted for, *the fact that one plant is equally as economical as another does not tell how far each of them is from its possible degree of perfection.*

6 The idea was advanced at the Spring Meeting of the Society in Buffalo, in June, 1915, by David Rushmore, that the opportunity exists for standardizing cost accounting for industrial plants, and it was hoped that some member would lay before the Society a method and data for establishing such a standard. The proper tabulation and distribution of costs is, however, of lesser importance than a satisfactory method of analyzing the data collected. It is the writer's belief that the time is now ripe to consider these questions, at least in this one branch of manufacture, namely, the generation of power.

CLASSIFICATION OF EXPENSES

7 All expenses incurred in the course of power production fall under analysis into two main groups:

a Constant (within a certain range) for any output

b Variable in some direct proportion with the output.

TABLE 1 COMPARATIVE COST OF OPERATION AND MAINTENANCE OF SEVEN POWER PLANTS — JUNE 1915

EASTERN RAILWAY CLASSIFICATION — (NOTE THE IMPOSSIBILITY OF ESTABLISHING A CRITERION FOR JUDGING THE TRUE OPERATING EFFICIENCY)

			A		B		C		D		E		F		G	
			Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.
OPERATION	Labor	Boiler Room	2233.21	0.036	1639.45	0.056	2686.34	0.025	4967.27	0.0195	3076.18	0.0318	3283.50	0.0855	1133.95	0.052
		Turbine Room	1116.52	0.018	679.89	0.024	1682.89	0.016	3156.36	0.0124	1471.10	0.0153	1472.54	0.0383	756.10	0.035
		Electrical	1080.52	0.018	522.75	0.018	536.12	0.005	982.22	0.0039	1105.73	0.0114	264.28	0.0120	144.87	0.007
		Superv. — Janitors and Watchmen	563.14	0.009	531.99	0.018	601.26	0.006	1230.89	0.0049	460.85	0.0048	1061.43	0.0276	149.80	0.007
		Total Operating Labor	4993.39	0.081	3374.08	0.116	5506.61	0.052	10336.74	0.0407	6113 86	0.0633	6281.75	0.1634	2184.72	0.101
	Material	Coal	20535.00	0.335	7338.55	0.252	29945.89	0.285	73803.55	0.2908	35260.67	0.3648	11820.31	0.3075	8587.80	0.395
		Water	675.65	0.011	59.56	0.002	1618.32	0.015	475.42	0.0019	1358.96	0.0141	789.18	0.0205	30.69	0.002
		Lubricants	84.02	0.002	61.11	0.002	44.88	0.001	226.47	0.0009	270.81	0.0028	135.68	0.0035	68.13	0.003
		Miscellaneous Material	194.55	0.003	157.30	0.005	27.70	0.000	2043.91	0.0080	35.12	0.0004	555.73	0.0145	149.87	0.007
		Miscellaneous Charges	20.91	0.000	Cr. 1.55	0.000	757.96	0.007	......	....	......	....	......	....	129.87	0.006
		Total Operating Material	21510.13	0.351	7614.97	0.261	32394.75	0.308	76549.35	0.3016	36920.56	0.3821	13300.90	0.3460	8966.36	0.413
		Total Operation	26503.52	0.432	10989.05	0.377	37901.36	0.360	86886.09	0.3423	43034.42	0.4454	19582.65	0.5094	11150.86	0.514
MAINTENANCE	Labor	Building	40.95	0.001	129.36	0.004	190.19	0.002	1312.28	0.0052	140.91	0.0015	77.35	0.0020	2.35	0.000
		Boilers	291.36	0.005	319.27	0.011	703.13	0.007	641.67	0.0025	751.72	0.0078	279.49	0.0073	139.45	0.006
		Boiler-Room Auxiliary Apparatus	173.89	0.003	268.33	0.009	594.54	0.005	520.47	0.0020	280.40	0.0029	66.15	0.0017	31.49	0.002
		Turbines	1055.16	0.017	27.49	0.001	328.36	0.003	55.50	0.0002	361.42	0.0037	286.24	0.0075	1.84	0.000
		Auxiliary Apparatus	142.08	0.002	21.23	0.001	561.59	0.005	322.21	0.0013	176.86	0.0018	334.92	0.0087	6.83	0.000
		Electrical Apparatus	9.32	0.000	1.50	0.000	399.57	0.004	119.43	0.0005	353.12	0.0037	85.69	0.0022	1.23	0.000
		Piping	85.83	0.001	54.03	0.002	380.61	0.004	433.24	0.0017	119.59	0.0012	131.69	0.0034	136.05	0.006
		Miscellaneous	49.40	0.001	55.06	0.002	132.23	0.001	55.69	0.0002	396.04	0.0041	1.92	0.0001	23.78	0.002
		Total Maintenance Labor	1847.99	0.030	876.27	0.030	3299.22	0.031	3460.49	0.0136	2580.06	0.0267	1263.45	0.0329	343.02	0.016

TABLE 1 (Continued)

		A		B		C		D		E		F		G	
		Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.	Dollars, Total	Cents per kw-hr.
MAINTENANCE	Material: Building	212.17	0.004	99.98	0.004	799.08	0.008	3490.00	0.0137	2.19	0.0000	29.65	0.0008	15.14	0.001
	Boilers	295.16	0.005	66.56	0.003	895.22	0.008	1389.30	0.0055	702.81	0.0072	169.67	0.0044	457.45	0.021
	Boiler-Room Auxiliary Apparatus	50.22	0.001	40.28	0.001	836.39	0.006	529.10	0.0021	88.08	0.0009	73.00	0.0019	36.18	0.002
	Turbines	78.42	0.001	0.25	0.000	739.08	0.007	27.48	0.0001	18.35	0.0002	78.08	0.0020	5.54	0.000
	Auxiliary Apparatus	41.08	0.001	8.18	0.000	1344.22	0.013	68.04	0.0002	143.86	0.0015	1300.79	0.0338	28.15	0.001
	Electrical Apparatus	14.06	0.000	0.84	0.000	138.89	0.001	766.58	0.0031	137.70	0.0014	162.18	0.0042	......	....
	Piping	20.69	0.000	7.94	0.000	220.35	0.002	574.31	0.0023	81.34	0.0008	200.10	0.0052	149.53	0.007
	Miscellaneous	8.76	0.000	2.13	0.000	127.11	0.001	82.47	0.0003	209.01	0.0022	......	....	60.60	0.003
	Total Maintenance Material	720.56	0.012	226.16	0.008	5100.34	0.048	6927.28	0.0273	1383.34	0.0143	2013.42	0.0523	752.59	0.035
	Total Maintenance	2568.55	0.042	1102.43	0.038	8399.56	0.079	10387.77	0.0409	3963.40	0.0410	3276.87	0.0852	1095.61	0.051
SUMMARY	Total Labor	6841.38	0.111	4250.35	0.146	8805.83	0.084	13797.23	0.0543	8693.92	0.0000	7545.20	0.1963	2527.74	0.117
	Total Material	22230.69	0.363	7841.13	0.269	37495.09	0.356	83476.63	0.3289	38303.90	0.3964	15314.32	0.3983	9718.95	0.445
	Total Lab. and Material, Pow. Sta. Proper	29072.07	0.474	12091.48	0.415	46300.92	0.440	97273.86	0.3832	46997.82	0.4864	22859.52	0.5946	12246.69	0.562
	Other Items Charged to Power Station Accts.	676.55	0.011	676.55	0.023	294.42	0.003	2024.28	0.0080	......	....	966.73	0.0252	255.52	0.013
	Total	29748.62	0.485	12768.03	0.438	46595.34	0.443	99298.14	0.3912	46997.82	0.4864	23826.25	0.6198	12502.21	0.575
	Net Output in kw-hr.	6133065	....	2915417	....	10509685	....	25286650	....	9664081	....	3844419	....	2171400	....
	Total Power Generated, kw-hr.	6211195	....	2941018	....	10625545	....	25385400	....	9781610	....	4043803	....	2171400	....
	Lb. Coal per kw-hr.	2.68	....	2.72	....	2.18	....	2.062	....	2.682	....	3.78015	....	3.40	....
	Cost of Coal per 2000 lb., dollars	2.50	....	1.85	....	2.613	....	2.808	....	2.72	....	1.61	....	2.33	....
	Load Factor — Machine, per cent	59.6	....	74.5	....	64.56	....	73.17	....	86.6	....	49.0	....	64.0	....
	Load Factor — 15 Min. Max., per cent	45.3	....	23.8	....	36.37	....	90.44	....	58.1	....	35.0	....	112.0	....
	B.t.u. per net kw-hr. Output	38289	....	33513	....	30716	....	29561	....	38046	....	43637	....	49800	....

8 Expenses that are independent of the volume of output are at the same time independent of each other and do not characterize the efficiency of processes performed in the power plant. Their effect on the unit cost is represented by a parabolic curve decreasing with the increase of output. They are exemplified by interest on investment, depreciation, sinking fund, insurances, management, pay roll (in some cases), taxes, etc.

9 Expenses that vary with the output of the plant characterize the efficiency of operation, other conditions being constant, and their elements stand together in dependent sequence. If represented graphically, they show very irregularly shaped curves peculiar to each set of equipment. The unit cost has a general tendency to drop with an increase of output, as the efficiencies of boilers, turbines, etc., tend to improve with increased load; yet, since with higher degrees of overload the efficiency decreases, the unit cost rises. With further increase of load when an additional unit is started, the efficiency again begins to improve until their cumulative efficient capacity is exceeded, when the unit cost commences to increase again. Such waves are sometimes very pronounced, and generally, throughout the range of the plant's capacity, the number of waves on the unit-cost curve is equal to the number of generating units installed. Fuel, water, certain supplies, and, in less pronounced dependence, maintenance expenses, belong to this group of expenses, and to classify the expense account it is best to itemize them to correspond with the steps in which the energy is transformed during the generating process.

10 The criterion of economy is established by the interplay of three factors, *time*, *product* and *cost*. When only one factor varies, its effect on the economy can easily be foreseen. Thus a greater product, without a change of the time required or the cost, increases the economy. An increase of either the time of production or the cost of production reduces the economy. Generally, however, all the factors vary simultaneously, in which case an analysis of the equation for the economy criterion $e = \frac{P}{Ct}$ can be made for any influential element n, and the increase of economy for a unit increase of n, if differentiated, is

$$\frac{P}{Ct} = \left(\frac{P'n}{P} + \frac{C'n}{C} + \frac{t'n}{t}\right)$$

all elements being essentially positive.

11 Such a general study can be made with respect to economy if more than one influential element is involved by means of simul-

taneous equations for each. The graphic method offers, however, an easier means of solving the problem. To determine the *economic limit* reached by a continuous increase or decrease of the influential elements is not an easy problem, but unless it is solved we are in the dark not only as to *what economy can be obtained*, but also *what changes in conditions and methods are essential.* The analysis of the effect of the variations of these elements involved in determining the *maximum limit of economy* can be compared to a determination of the height of the apex of a hill by taking altitude readings on both slopes in one direction and then repeating observations in the crosswise direction.

STANDARD COSTS

12 It is relatively unimportant whether the maximum limit of economy is determined empirically by rigorous observations, tests and analyses of all influential elements, or calculated from the principal data already available. *It is imperative that such study be made and the economy limit established, as this is the only criterion for judging the actual performance.* Carrying out the analysis of the economy limit to its logical conclusion, the standard cost of the product is arrived at, and evidently during this investigation not only are the itemized costs of the individual partial processes found out, but the conditions and methods whereby the standard cost can be attained are established. In other words, *unless the standard costs are established there is no measure of the existing losses or exact knowledge as to how to eliminate them.*

13 Manifestly, in the course of determining the standard operating cost, such factors as the inherent efficiency of the equipment, its efficiency under different loads, the prices of fuel and supplies, the necessary and sufficient number of attendants and their compensation, etc., are already taken into consideration for a given plant. Any difference observed between the actual operating cost and this standard cost indicates that some of the necessary conditions were not lived up to, and, if standardization has been carried out in sufficient detail, it leads directly to the allocation of the loss to operating methods. On the other hand, any change in the basic data used in determining the standard cost being known, an adjustment of the standard cost can easily be made before the blame is put at the door of the operators. Since the efficiency of the thermodynamic process so largely determines the operating cost in a power plant, it should be made the subject of a most thorough investigation to ascertain first the maximum efficiency limit of each partial process, and then the

result of their interplay. When this is accomplished, the entire process should be restudied for the purpose of standardizing the methods and adjusting for such a balance of the efficiencies of the partial processes as will secure the maximum profit or economy for the time, energy and money expended. In this it is sometimes found that the most economical thermal efficiency is somewhat below the maximum obtainable, as the slight additional gain in efficiency necessary to reach the maximum is not warranted by the expenditure required to attain it. When these limiting conditions are studied and determined, a method can be defined for each member of the working force, prescribing his duties and the conditions he must maintain to secure the *most profitable* degree of efficiency.

14 Upon the conclusion of these studies, the best efficiency of each unit and their combination being known for any load, the standard cost for any output in a given time unit can be conveniently represented in graphical form.

15 The principles of determining the standard cost of maintenance and upkeep of the plant and equipment are substantially the same; the method of study, however, is somewhat different. It involves a study of the design and construction of all the elements of the equipment; minute records of their service and cost of maintenance may lead to a modification of design, use of cheaper renewable parts, etc. Next, the standardization of supplies, beginning in the laboratory and followed by actual service tests, helps to determine not the lowest purchase price but the lowest service cost. Finally, time studies embracing schedules for inspection, routes for maintenance men, standardization of tools, motions, methods, etc., conclude the investigation. The criterion is, of course, not the wages of the employees, but freedom from accidents, breakdowns and the lowest attainable cost of upkeep per unit of the plant's output. It is evident that there may not be any theoretically certain standard cost of maintenance, but an empirical standard thus developed is generally but a fraction of the best actual records of the past.

CURVES OF STANDARD COSTS

16 Upon concluding this double analysis of the maximum economy obtainable, the graphs of the standard cost of the power production may be drawn. Curves may be conveniently arranged with the coördinates of cost C and product per unit of time $\frac{P}{t}$ (output). It will then be noticed that the time element is one of the most

influential factors in power economy. Whereas in some cases, where the number of generating units is large, the coal rate per unit of output remains fairly flat and the other items of cost reduce rapidly with increased production per unit of time, in other cases the standard cost of fuel also decreases as the time during which a certain output is produced is reduced. Figs. 1 and 2 represent curves of standard operating costs. It is evident that any number of curves may be plotted following the above method, each curve representing an itemized standard cost according to the adopted classification. Fig. 1

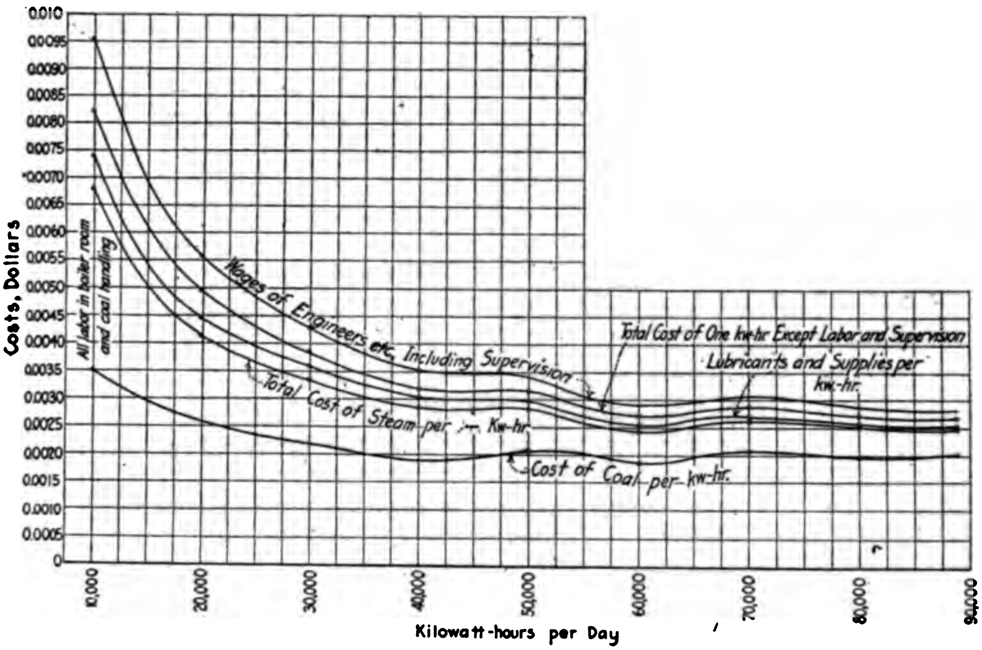

FIG. 1 CURVES SHOWING VARIATION IN STANDARD COSTS OF OPERATION IN A PUBLIC-UTILITY CENTRAL STATION

is thus prepared for a medium-size public-utility central station It shows the variations of the standard costs of coal, boiler-room labor, water, supplies, overhead charges, engineers and supervision per kilowatt-hour at various monthly outputs. The cost scale does not show, however, the actual standard. This plant comprises four 600-h.p. boilers and three turbo-generators, one of 2500 kva. and two of 500 kva. rating; it operates 24 hr. per day, 7 days a week.

17 From this diagram it appears that the cost of coal per kilo-

watt-hour is lowest when the output of the plant is approximately 1,900,000 kw-hr. per month. Further increase of output coincides with the increased cost for fuel required, due to the characteristic of boilers and turbines that they lose in their efficiency at higher rates of driving. Again, costs of labor, supplies, prorated overhead charges, etc., per kilowatt-hour, which drop more rapidly than the cost of fuel rises, offset the difference and render a greater monthly output more desirable economically. Even there, however, we meet a limit when at the rate of 2,200,000 kw-hr. per month the unit cost

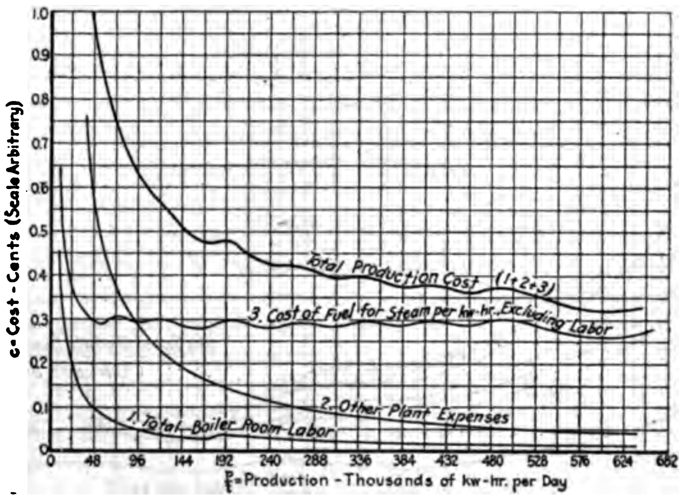

FIG. 2 CURVES SHOWING VARIATION OF STANDARD COSTS OF POWER PRODUCTION AT AN ELECTRIFIED RAILWAY CENTRAL PLANT

becomes higher than it was at the lower output of 1,900,000 kw-hr. per month.

18 Fig. 2 illustrates a few characteristic curves of the standard cost per kilowatt-hour for various rates of output of a large central station feeding the lines of an electrified trunk railroad. The plurality of waves on the fuel-cost curve, etc., is explained elsewhere in this paper, and the tendency of the total standard cost to go steadily downward with increase of the output rate is due to the water rate of generating units, as well as to distribution of constant expenses over a large output.

USE OF STANDARD COST CURVES

19 The practical use of such predetermined standard costs can be made extremely simple by employing these graphs. For busy execu-

tives or owners, the entire cost record visualized by graphical representations of the items of account is found very convenient. An example of such a graph is seen in Fig. 3, wherein the actual unit cost and the standard unit cost are plotted to the same scale, the deviation of one from the other suggesting at a glance the degree of perfection of the performance. The total-expense curve and the cumulative-expense curve may be shown on the same graph to a suitable scale; the latter curve is found very serviceable for comparing these items with the appropriation made. A cost system kept on a card file in this manner will represent clearly in any desired detail for any length of time and at any period:

a How much was spent
b How much each unit of output cost
c How much it should have cost
d What the fluctuations of expenses and unit cost are
e What the fluctuations of efficiency are
f How close the actual amount spent in any time is to the appropriation.

The accuracy of such graphic records is sufficient for most practical uses and references, as it allows the interpolation of unit costs to 0.001 of a cent. If the exact total of expenditure is wanted, it can be had at any time from the book records, whereas the use of books and figures exclusively lacks the comprehensiveness and visual instructive value of graphs.

20 Any comparison of the production costs of various plants may now be made in a different light. By comparing standard costs of one plant with those of another, one gains the knowledge of how much cheaper the power can be produced in one plant than in another, due to its various physical advantages, corrections for load and output all being automatic. Again, by noting how near the actual cost of one plant is to its standard cost, one has at once a measure of the quality of the methods and management. Thus, referring to our Figs. 1 and 2, the actual cost per kilowatt-hour of 35 cents in one plant means a worse operating economy than 48 cents in another plant under another load condition. Yet without such accurately predetermined standard costs that are individual for each plant and condition of load, no correct comparison is possible, and conclusions from a mere study of accountants' figures are apt to be grossly in error.

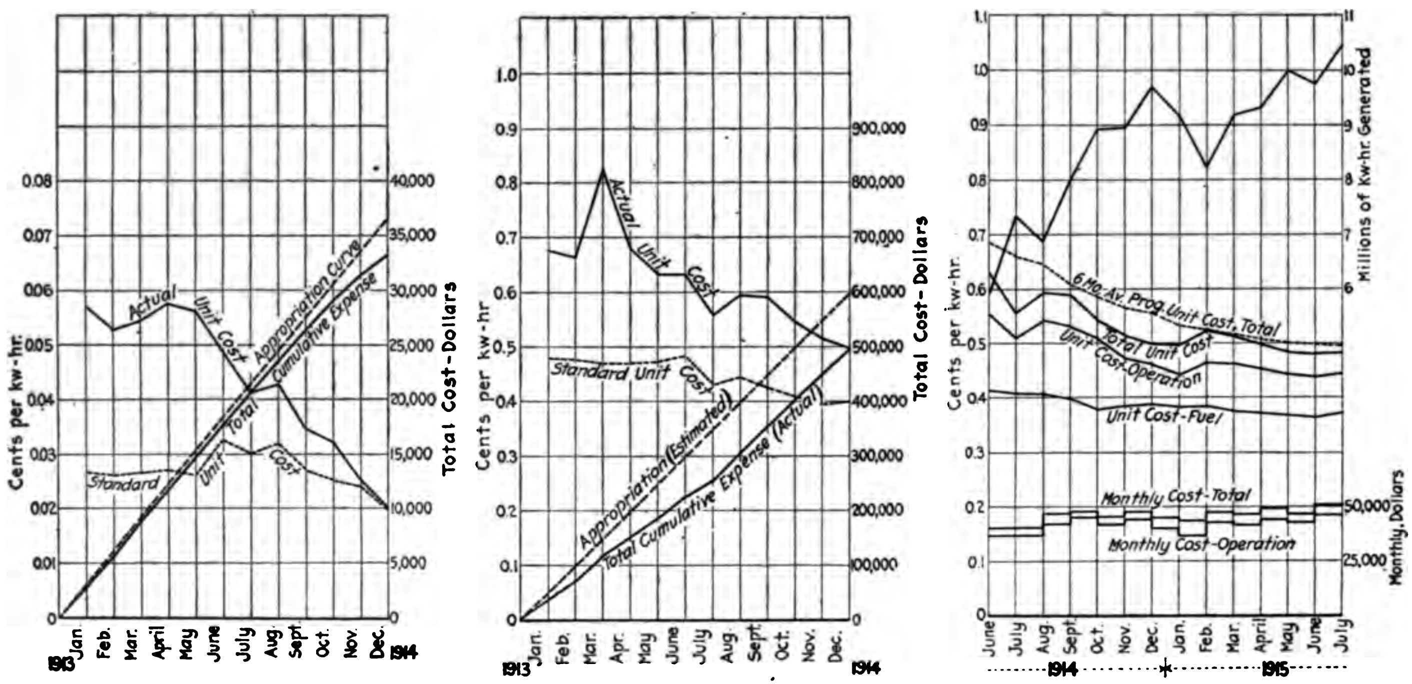

Fig. 3 Visualized Records of the Entire Cost of Power-Plant Operation, in which the Actual Unit Cost and the Standard Unit Cost are Plotted to Same Scale

STANDARDIZATION OF OVERHEAD CHARGES

21 In the above discussion of standard costs, consideration of the fixed charges borne by the plant was so far omitted, as they are not purely operating charges according to the fancy of accountants. Their importance, however, is emphasized by the very common practice of installing in power plants considerable spare equipment, partly as a protection against possible breakdown and partly to take care of high peak loads. In both cases it would not be logical to adulterate the production costs by adding all overhead charges incident to this idle equipment. If all charges due to over-equipment, new business anticipation and similar allied items are charged where they belong, in business profit and loss account — not added to the cost of product — the cost of current will appear constant unless prices of fuel, supplies, wages, etc., are changed or methods of management are not uniform.

22 Inasmuch as the spare equipment has a function identical to a stand-by plant at some distant point of the transmission line or a breakdown emergency connection with a power company nearby, it seems entirely proper to segregate this part of the charges and treat it like the bills of an emergency contract, *i.e.*, to carry the cost in a separate business account. The parallel between a breakdown-connection contract and a plant's own breakdown reserve is apparent, yet all too often the same concern that charges the bills on such a contract against the general business or protection account, insists that the power department shall absorb in its operating account all expenses accrued in the reserve plant or any part thereof.

23 The degree of safety and protection from interruptions with which it may be desired to maintain the service is dictated by the general policy of the concern; similarly, the decision to keep a stand-by station in readiness to go into service at a moment's notice or the policy of keeping within the central station a certain number of boilers banked, turbines warmed up and generators phased, should not obscure the cost and efficiency data pertaining to the generation of power. The expenses incurred in connection with such a *protective service* should not be charged to the operating or production account, but might be arbitrarily standardized, reduced or increased as the business policy might require and be *treated as an independent account*.

24 If, in a plant built and equipped to produce twice the output it normally yields, fixed charges are dumped together with other costs, it reflects unfairly on the ability of the superintendent and his

operating force; whereas if they were properly charged to a separate account, the excessive charge against the stand-by equipment, both for fixed charges and running expense, would produce a stimulus for the new-business, or commercial, or transportation departments to secure additional load. It is quite common for these departments to be not fully aware of the reserve capacity of the plant or plants, or, on the contrary, to secure more load than it is safe to carry permanently or until an additional generating equipment is provided. A *standard* cost of the protective policy would act in all such cases as a gage on a tank of certain capacity with outlet and inlet valves subject to continuous adjustments. In other words, *the new-business departments should at all times know the ratio between the ideal protective costs and the actual cost of carrying the unutilized part of the power-generating equipment.*

COMPARISON OF COSTS AND EFFICIENCIES

25 A cost system developed along the lines discussed will not only afford a means for clearly understanding the operating and managerial problems, but offer a basis for cost comparisons of different plants. The essentials of the knowledge gained through comparing actual with standard costs are:

a Relative supremacy of plants proper
b Relative advantages of managerial methods
c Relative extent of preventable losses
d Relative advantages of prices of materials, etc.

These cannot be found unless there is a comparable basis or scale for comparison, which is offered by standard costs determined for each plant individually, as at least ten main variables must be accounted for, as follows:

a Nature of load
b Character of service
c Conditions imposed by location
d Inherent efficiency of equipment
e Arrangement of equipment, floor plan, etc.
f Cost efficiency of fuel and supplies
g Legal requirements
h Methods of operation
i Labor conditions
j Methods of compensation for service.

Each of these variables being in its turn a product of a plurality of factors, it is manifestly impossible to state, without the aid of carefully-worked-out standard costs, that the economy of one plant or another is satisfactory, or where and how it can be bettered.

26 Inasmuch as the standard cost cannot be determined without first finding out how the maximum economy can be secured, the process of standardizing costs is also a process of devising the best way for operating and managing. Once both methods and results are positively established, costs are but the form of expressing the final result. It is true that the standard cost is influenced by the prices of commodities used in connection with the generation of power, as well as by some conditions beyond the control of the management and the operating engineers, but the adjustment of the standard costs to every change of these factors can be made as simple as the use of a slide rule. Furthermore, a separate account should be kept for such charges as are due to business policy, so that *a division of responsibility between those managing the production and those directing the business could be drawn.*

27 So long as costs for producing power come as an unexpected surprise and arouse the curiosity to an extent of comparing them with the preceding month, year, or some other plant's data, the management of such power plants is evidently very haphazard, lacking an aim at any definite goal. Without predetermined standards, superintendents and managers will continue to believe that they control the production, and owners will remain in happy ignorance as to how much of their money goes to waste and why. Only after the establishment of standards and ideal costs of production by means of the most rigorous analysis will cost accounting be of help to engineers, and only then can it be said that the generation of power is directed by the management and controlled by the engineers.

DISCUSSION

This paper was originally presented at a meeting of the New York Section on January 11, 1916. The complete discussion at this meeting was published in The Journal, April, 1916. Excerpts from this discussion bearing directly on statements made by the author are given below.

William D. Ennis wrote that it is perfectly true that accountants' records in themselves are inadequate, yet they are the official and final figures, and owners are interested only in the final result. They will not take kindly to the elimination of overhead charges on idle equipment from the operating account. Furthermore, relative necessities for reserve equipment and breakdown service strongly influence the decision as to what is the best type of equipment for a given set of conditions. Certainly breakdown service is not provided for the sake of anticipated "new business."

The author regards the proper tabulation and distribution of costs as less important than their analysis. But proper tabulation is the essential prerequisite for any sort of analysis, and in this respect our power-plant management is defective or at least unstandardized. Table 1 gives operating and maintenance costs. It would be a matter of some complication for any power operator to line up these figures with his own. Different power companies classify their costs in different ways. There is not even a clear distinction drawn between the items "operation" and "maintenance." It would be a decidedly useful thing if we had a standard classification in power-station accounting, such a standard classification as those used by all industrial corporations. Central stations would no more than at present be expected to publish their costs, but when costs were furnished, the reader would know without explanation the meaning of each item.

Henry G. Stott. Contrary to the author's statement, predetermination of results of power production is purely an engineering question, so far as the power operation is concerned. I have never heard of an accountant being asked to furnish an estimate of how much it costs to furnish power.

The author states the expense incurred in connection with keeping a standby unit should not be charged to the operation or production account, but should be treated as an independent account. That, of course, is absolutely impossible, because it is just as much a part of the cost of producing power to carry the reserve power as it is to carry the main unit in operation, and it is just as proper to charge a reserve unit into the cost of power as the operating unit.

W. F. Schaller.[1] As to the proposition of figuring stand-by costs and charging them to separate accounts, if a plant has such a load that a number of extra boilers have to be kept on the line, and the plant is to be compared with another which has a steadier load, so

[1] 366 Fifth Ave., New York, N. Y.

that the smaller number of boilers need be kept hot in reserve, it is unfair to compare directly the coal per kilowatt.

JOHN W. LIEB. The author, in addition to his analysis from the viewpoint of the operating engineer, has indicated certain factors which are not properly qualified operating costs, but elements of central-station economics having to do with finance and management. Now production costs in central-station economics have come to mean a definite thing. They exclude all elements of interest, insurance, taxes, general expense, return on capital and depreciation of various kinds; they include merely the items that go into what is known as switchboard cost, which is a very specific and definite thing in the central-station industry. Production costs are quite different from the cost of power.

Some work has still to be done in the branch of standardization from year to year, — as to what output shall serve as the unit measurement of the production costs. Also, when considering the load factors as the basis of comparison between one station and another, there is not a general agreement as to the time interval over which the so-called maximum demand on the plants shall be made.

SELBY HAAR. If a certain ideal condition is established, as suggested by the author, or one of a similar character, how long will it remain standard? Every hour even some improvement is brought forward and the bottom knocked out of all of our preconceived and carefully standardized ideas.

THE AUTHOR. Mr. Lieb overlooks the division line drawn in the paper between production cost and overhead charges, and it is just the thing that the paper exposes as a harmful fallacy that Messrs. Percy and Ennis approve of. Again, Mr. Haar's criticism is applicable only to a hypothetical plant in which "every hour" a new improved equipment is installed, but the author's central and only idea in this monograph was silently accepted.

The cost of power is the result of method as well as of other factors. Prices of commodity and labor used are known; efficiency of equipment used is known; nature of service is known, and their influence on cost of product is easily found from a comparatively simple study. In a given plant these factors remain constant until change is made by purchase. For different plants factors of merit could be established equalizing for the unequal condition. Now if costs vary it is due to variation of the methods of management of the plant. If these methods are standardized and the costs still vary, other conditions being constant, it means that standard methods are not lived up to; that is, the management fails to manage.

The paper thus offers a method of measuring the efficiency of the plant management. If the principles and methods are right, the results will necessarily be the best obtainable, and consequently the only best method or "standard method" finds its expression in the lowest possible cost or "standard cost."

Comparing such standard and actual results, with due allowance for uncontrollable factors, we get the measure, established on a scientific basis. Under such régime the plant management ceases to be a haphazard undertaking and most preventable losses are eliminated. Separate and aside from the operating and production costs are other elements of overhead charges, and a separate plea is made to analyze them independently so that their reason and purpose be clearly indicated, and if expense is incurred for no useful purpose or on poor reason the remedy could be found.

No. 1550

THE UTILIZATION OF WASTE HEAT FOR STEAM-GENERATING PURPOSES

By Arthur D. Pratt, New York, N. Y.
Member of the Society

The utilization of waste heat from various industrial processes for the generation of steam is not new. The advance within the last few years, however, in methods of utilizing such gases and in the results secured from their utilization has been so remarkable as to make of interest a comparison of former with present-day methods and results.

2 The design of waste-heat boilers has progressed to a point where it is today possible to generate steam successfully from gases whose temperatures are as low as 950 or 1000 deg. fahr.[1] It is but a few years since it was considered absolutely impracticable from a commercial standpoint to attempt to produce power from such gases, and it is in its ability to satisfactorily utilize these gases that the development of the modern waste-heat boiler has its most far-reaching effect.

3 The sole theory on which early waste-heat boiler installations were made had as its basis the non-interference in the operation of the primary furnace. This meant, in practically all cases, that the draft at the exit of the primary furnace should in no way be impeded, and resulted in the installation of a given amount of heating surface arranged in such manner that the frictional resistance to the gases in their passage through the boiler should be a minimum. Presumably the object of such an arrangement was to enable a natural-draft stack of a practicable height to be used. As the result of minimizing this draft loss, such boilers as were installed were ordinarily entirely without baffles and the gases were given a straight passage through the boiler, though partially baffled boilers were used occasionally. High exit

[1] All temperatures herein given are in deg. fahr.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

gas temperatures were considered rather desirable than otherwise in order to assist the stack in giving the required drafts.

4 Boilers installed in this way were considered practicable only with gases whose temperatures approached those of coal-fired practice. The users accepted the steam generated as "something for nothing," and no particular endeavor was apparently made toward increased capacities.

5 Present-day waste-heat practice has come about with a more thorough understanding of the laws governing heat transfer and an appreciation of the function of gas velocity as affecting transfer rates. Without going into this aspect, it may be broadly stated that the rate of heat transfer is dependent upon gas velocity and temperature difference between the gas and the absorbing surface. Experiments have shown that at even the highest velocities now used in waste-heat work the effect of increased temperature difference is small as compared with that of increased gas velocity.

6 In coal-fired-boiler practice the average temperature difference between gases and boiler surfaces is approximately 1150 deg. The heat-transfer rate corresponding to a boiler's rated capacity is about 3 B.t.u. per hr. per sq. ft. of surface per deg. difference, or a heat absorption of 3450 B.t.u. per sq. ft. of surface per hour. In waste-heat work the temperature difference varies widely with the class of waste heat. With a gas temperature, say, of 1250 deg. entering a boiler, the average temperature difference between boiler surface and gas for a working pressure of 170 lb. per sq. in. will be about 500 deg. For such a temperature difference the transfer rate, to give an absorption per square foot of surface equal to that of the coal-fired boiler at rating, would have to be 6.9 B.t.u. per hr. per sq. ft. per deg. difference. This rate is slightly higher than the rates corresponding to velocities that are as yet ordinarily used, but with somewhat higher entering gas temperatures the absorption per square foot of surface is such as to enable a boiler's rated capacity to be developed without difficulty. With gas temperatures entering the boiler of 1800 to 2000 deg., which approach coal-fired practice, high overloads are being developed.

7 The gas velocities necessary to give what is now considered a desirable transfer rate lead to a frictional resistance through waste-heat boilers which makes the use of a natural-draft stack impracticable. Let us consider what is probably the extreme case in so far as draft conditions are concerned, namely, the open-hearth steel furnace. Common practice in this class of work is to use stacks 160 ft. high, which

give a draft at their base of approximately 1.4 to 1.6 in., depending upon the gas temperatures. For the present purpose, assume that the draft loss through a modern waste-heat boiler installed with an open-hearth furnace is 2.0 in., a figure which approximately represents the practice of today. With a natural-draft stack, the draft at the checkers corresponding to the 1.4 to 1.6 in. given above is approximately 1.3 to 1.5 in., and this amount is necessary for the proper operation of the furnace. With a waste-heat boiler installed, then, the draft necessary at the exit of the boiler must be sufficient to overcome the resistance through the boiler, 2 in., that necessary to overcome resistance through the flues, say 0.75 in., and 1.5 in. necessary at the checkers, or a total of 4.25 in. It is to be remembered that with a waste-heat boiler installed, the temperature of the gases entering the stack, instead of being 1000 or 1200 deg., will be 450 or 500 deg., under which conditions a stack of 160 ft. instead of giving a draft of 1.4 to 1.6 in. at its base would give approximately 0.9 in., and to give the necessary 4.25 in. the stack height would have to be somewhat over 700 ft.

8 While, as stated, this is perhaps the extreme case, the same reasoning applies in practically all waste-heat work, and an induced-draft unit is now almost universally used with the modern design of waste-heat boiler. In certain classes of waste-heat work, such a fan not only furnishes the required draft suction, but is of a decided advantage in the operation of the primary furnace. This feature will be discussed in connection with the individual classes of waste heat later considered.

9 As may be inferred from the foregoing, the successful utilization of waste gases becomes more difficult with decreasing gas temperatures. It was in connection with regenerative furnaces and low-temperature gases that the principles of high gas velocity were first applied and the modern waste-heat boiler was developed. The success of the early installations in this particular class of work led to the application of this theory to all classes of waste-heat practice.

10 Waste-heat boilers of the modern design are in successful operation to-day with copper-refining furnaces, cement kilns, open-hearth steel furnaces, beehive coke ovens, zinc-refining furnaces and heating furnaces of various types, both regenerative and non-regenerative.

11 As stated, the waste-heat boiler of today was developed with low-temperature gases and its largest field, until the present, has been

with open-hearth steel furnaces. For this reason this class of waste-heat work will be considered first.

OPEN-HEARTH STEEL FURNACES

12 This particular phase of waste-heat work may be considered new. In the open-hearth steel furnace we have the best and by far the most numerous examples of the regenerative furnace. From the very nature of the operation of such furnaces, gas temperatures passing to the stack are low, and the ability of the modern waste-heat boiler to utilize successfully these gases for the generation of steam is, without question, the best proof of the progress in the development of this particular class of boiler.

13 An experimental waste-heat boiler for this work, embodying the principles but possibly not the design of the modern waste-heat boiler, was installed at the South Chicago plant of the Illinois Steel Company, in 1910. The results from this installation, while by no means comparable with those secured today with the more highly developed design, so clearly indicated possible savings in the practically untried field of open-hearth practice that two boilers, the design of which may be considered well past the experimental stage, were purchased for this same plant in 1911.

14 C. J. Bacon, Mem.Am.Soc.M.E., steam engineer of the Illinois Steel Company, presented before the American Iron and Steel Institute in May, 1915, a paper, Waste Heat Boilers for Open Hearth Furnaces, in which he describes the experimental boiler referred to, and traces the development in this class of work through the first three installations made.

15 The first of these, at the plant of the Illinois Steel Company, in South Chicago, consisted of two Stirling boilers equipped with special cross baffles to give the desired gas velocities over the heating surfaces.

16 The second installation, made at the Gary plant of the Indiana Steel Company, consisted of twenty-eight special 6-drum Rust boilers equipped as in the case of the Stirling boilers with special cross baffles.

17 The third installation described is that made at the Pencoyd Works of the American Bridge Co. and consists of a cross-drum Babcock and Wilcox boiler, the width of boiler and the tube length being such as to give the gas-passage areas necessary for the required gas velocity. For low-temperature waste-heat work, the type of boiler

of which this installation is an example is in general best suited. Fig. 1 shows a typical layout of a boiler of this last design.

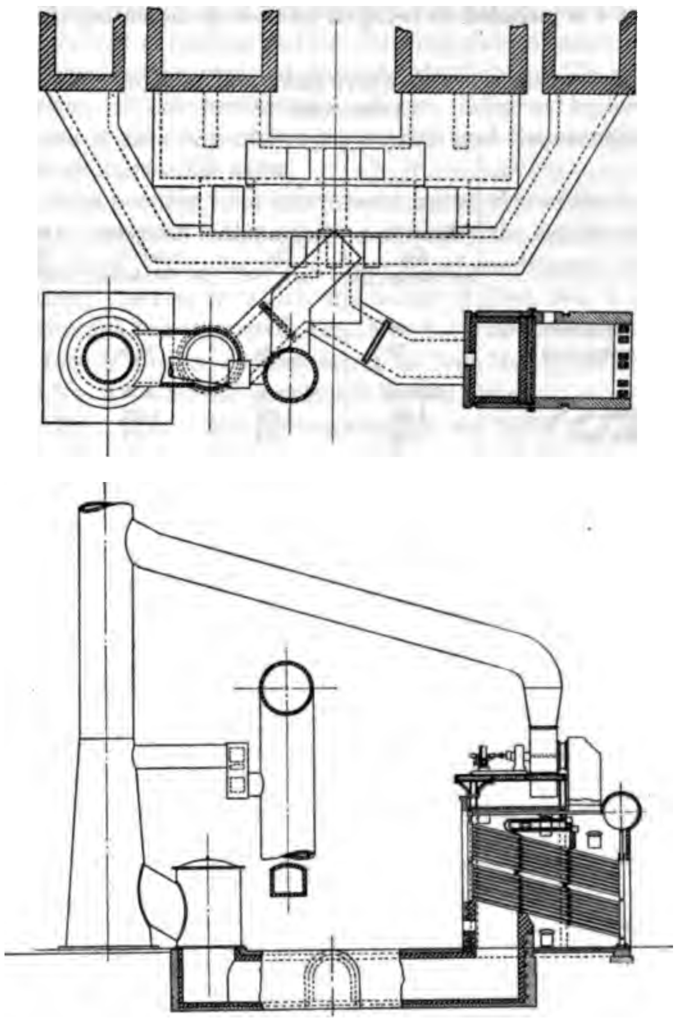

FIG. 1 WASTE-HEAT BOILER FOR OPEN-HEARTH FURNACE

18 Table 1 gives the results secured from the Stirling and Rust boilers described and from Babcock and Wilcox boilers similar in design to that installed at Pencoyd, though set with considerably larger open-hearth furnaces. That the three installations described by

Bacon are truly progressive, at least from the aspect of heat absorption, may be seen from the results given in this table.

19 Tests 1, 2 and 3 are as given in Bacon's paper and its discussion. Test 4 is included as being of interest in indicating the results

TABLE 1 RESULTS OF TESTS OF WASTE-HEAT BOILERS FOR OPEN-HEARTH FURNACES

Test Number	1[1]	2	3	4
Plant	Illinois Steel Co.	Indiana Steel Co.	Bethlehem Steel Co.	Lackawanna Steel Co.
Location	So. Chicago, Ill.	Gary, Ind.	So. Bethlehem, Pa.	Buffalo, N. Y.
Rated capacity of furnace, tons	65	75	80	[5]
Actual production, tons	72[2]	85	82.6	....
Boiler	Stirling	Rust	B. & W.	B. & W.
Heating surface, sq. ft.	4,000	4,880	5,232	5,407
Superheat, deg. fahr.	128	176	121	97
Gas weight, lb. per hr.	73,000	83,434	75,271	78,947
Gas per hr. per sq. ft. of h. s., lb.	18.3	17.1	14.4	14.6
Temperatures:				
Gas entering boiler, deg. fahr.	1,227	1,155	1,362	986
Gas leaving boiler, deg. fahr.	621	530	493	468
Drop in temp., deg. fahr.	606	625	869	518
Draft at boiler inlet, in.	....	1.47	1.55	1.76
Draft at boiler damper, in.	....	3.95	3.29	3.63
Draft loss, in.	1.78	2.48	1.74	1.87
Gross h.p. developed	334.5	393	425.8	306
Per cent of rated capacity [6]	83.6	80.6	81.4	56.7
Boiler h.p. to fan	[3]	60[4]	24.3[3]	....
Net horsepower	....	386	401.5	....
Approximate transfer rate (R)	5.08	6.92	4.77	5.12

[1] Average of ten tests.
[2] Approximate.
[3] Motor-driven fan.
[4] 53 h.p. returned in feedwater heater.
[5] Tilting furnace.
[6] All ratings are on the basis of 10 sq. ft per h.p.

that may be secured with gases at a temperature considerably below that ordinarily found in open-hearth work. At this particular plant, the gas temperatures leaving the checkers were not appreciably lower than in other tilting furnaces, but the necessary flue arrangement was such that there was an excessive radiation loss between the checkers and the boiler from flues and reversing valves.

20 In Table 1, as in the following tables, the proper basis for the comparison of results is the rate of heat transfer, but in order that such a comparison may be intelligent, it is necessary in connection with the transfer rates to give proper consideration to the gas weights per square foot of heating surface, entering and exit gas temperatures, and the percentage of rated capacity developed. To give a direct comparison of the performance of two different boilers, the gas weight per square foot of heating surface and the entering gas temperature should be the same.

21 It is possible from our present knowledge of the laws of heat transfer to compute with surprising accuracy the results that may be expected from any boiler for a given set of conditions. Let us consider, then, the results which the boiler of Test No. 4 would give under the gas conditions of Test No. 1. For a weight of gas per square foot of surface corresponding to Test No. 1, the total weight of Test No. 4 would become 98,950 lb. per hour.

22 With this weight passing through the boiler of Test No. 4, at the temperature actually existing in No. 4, the boiler would develop approximately 363 h.p., or 67 per cent of its rated capacity. With an entering temperature equal to that in Test No. 1, the boiler would develop approximately 500 h.p., or 92 per cent of its rated capacity. These figures are included simply to show the importance of giving all factors proper consideration where a comparison of results is to be made.

23 In connection with waste-heat boilers there are certain features of installation and operation, some of which refer specifically to open-hearth work and others to waste-heat work in general, that are of interest. While these features are discussed here, they will apply in certain of the other classes of waste-heat work considered later.

LOCATION OF BOILERS

24 The early boiler installations were made with open-hearth furnaces already in operation. With these furnaces it was necessary to connect the boiler to the flue between the reversing valves and the stack, the boiler thus being on a branch flue while the stack was on the more direct connection. It was and is common practice, in laying out open-hearth furnaces, to place the stack central with the furnace, and wherever such an arrangement is followed the boiler would of necessity be on the branch flue.

25 It is recommended, and today this recommendation is being more or less followed, that the boiler be placed central with the fur-

nace and the by-pass stack on the branch connection. Such an arrangement will give a straight gas passage to the boiler, minimizing frictional resistance in the connecting flues due to the absence of turns, and will have a tendency to give an equal distribution of the gases across the width of the boiler.

26 A further advantage of such an arrangement is that better protection is given to the by-pass stack damper. These dampers are ordinarily of cast iron, and where they close a direct passage to the stack, they are subjected to temperatures considerably higher than if the stack were on the branch connection and this damper installed on such connection.

CONNECTING FLUES

27 The flues from the reversing valves to the boiler should be as short as possible to minimize radiation losses. The location of such flues, with the same object in view, is also of importance. Ordinarily these are placed underground, and their depth beneath the surface should be such as to furnish a sufficiently thick insulating layer of earth on top. The best practice is to make this layer from three to four feet thick. In one installation where, because of certain unavoidable conditions, the thickness of the earth covering on the top of the flue was something less than one foot, the temperature of the surface of this earth was over 400 deg. The loss from radiation from such a flue is obvious.

28 Ample means should be supplied for keeping these connecting flues clean, and the area should be such as to allow a certain amount of accumulation of dust when the furnace is down without impeding the draft. The average life of an open-hearth furnace is probably around 300 heats, though occasionally 350 are obtained. When the amount of dust carried in the gases is considered, it is readily conceivable how there can be an accumulation between furnace lay-off periods that will seriously affect the draft. In one instance where, because of press of work, the connecting flue was not cleaned out during two periods when the furnace was down, the dust had filled the flue to more than a third of its cross-sectional area.

AIR LEAKAGE

29 Too much importance cannot be attached to keeping flue connections and boiler settings tight in order to minimize air leakage, the effect of which is to reduce gas temperatures, decrease boiler capacity, and place an added burden on the fan unit.

30 The effects of air leakage and the possible improvement in results through its reduction may be shown by an example. One of the early boilers installed was with a small furnace which had been in operation for some time. The boiler was first put into service with no particular attention given to the tightness of existing flues, and a continuous run of 119 hr. was started. For the first day or two the gas temperatures entering the boiler were very low, the temperature drop between the checkers and the boiler entrance being some 650 deg. This excessive cooling of the gases was found to be due to air infiltration through the length of the flue and an excessive amount of leakage at the stack by-pass valve. Nothing could be done toward remedying the latter defect while the furnace was in operation, but flue leaks were stopped as far as possible without stopping the test. The reduction in these leaks increased the gas temperatures entering the boiler at the end of the run to approximately 1000 deg., as against 600 to 650 deg. at the start. The average gas temperature entering the boiler during this run was 855 deg.

31 The furnace was then shut down, the sources of air leakage more thoroughly gone over, and the leaky dampers repaired and sealed. A second run of 120 hr. was then started, during which the average gas temperature entering the boiler, with the furnace conditions as nearly as possible like those in the first run, was 1153 deg. The loss between checkers and boiler was reduced to approximately 300 deg., most of which was due to a faulty reversing valve that could not be repaired without replacing.

32 The results of the two runs, showing the effect of reducing leakage, are as follows:

	Run No. 1	Run No. 2
Gas entering boiler CO_2	8.08	10.49
Gas entering boiler O	11.46	8.29
Gas temperature entering boiler	855	1153
Gas temperature leaving boiler	426	479
Draft boiler damper	3.99	3.77
Draft boiler inlet	1.85	1.85
Draft drop	2.14	1.92
Steam to fan engine (in terms of boiler horsepower)	35.7	28.1
Horsepower developed	132.4	200.2
Per cent of rated capacity	53.8	81.4

33 It is of interest to note from these figures that by minimizing the air leakage the draft loss through the boiler was reduced 0.22 in., which made possible a lower fan speed with a considerably higher economical rate for the fan turbine. A saving was effected of 7.5

b.h.p. for the fan drive, and there was an increase of 67 gross horsepower. Although this may be an extreme case, the enormous gains secured by reducing the air leakage to a minimum are obvious.

34 Air leakage through the flues can be minimized through proper design and location. Leakage through the boiler setting is reduced by the use of a compact design, well erected and bricked in, with precautions to prevent leakage about all cleaning and dusting door frames. Much can be done by painting the settings with asphaltum paint.

CLEANING

35 As has been stated, the waste gases from open-hearth furnaces are very dirty; and because of the low temperatures, it is, perhaps, of more importance in this class of work than in coal-fired practice that the heating surfaces be kept clean. In the early installations, fears were expressed that dirt would have a tendency to stick to the tubes and be difficult, if not impossible, to remove. Experience has shown, however, that ordinary methods of dusting give perfectly satisfactory results.

36 In practically all plants now operating waste-heat boilers in connection with open-hearth furnaces, the boilers are dusted once in 24 hr. In one plant which operates 11 such boilers and in which the results are checked perhaps more closely than in the average steel plant, the engineers have concluded as the result of actual experience that it pays to dust every 8-hr. shift.

37 Some figures showing the effect of dusting on exit-gas temperatures are of interest. In a certain plant a series of tests was run with cleaning intervals of 20, 10 and 6 hours. When the boiler was allowed to go 20 hr. without cleaning, the reduction in gas temperatures after cleaning was 60 deg.; where the cleaning intervals were 10 hr., 50 deg., and where 6-hour intervals elapsed, the reduction varied from 10 to 30 deg. In a second plant, with larger boilers, after an undusted period of 24 hr., for an unchanged entering temperature the exit temperature was reduced 35 deg. by a thorough cleaning. In this instance, with the gas weight passing through the boiler, this 35-deg. reduction was equivalent to an increase of 22 h.p. This increase in capacity would not, of course, hold over a period of 24 hr., but it would appear conservative to state that the 24-hr. cleanings for such conditions would correspond to at least a net saving of 10 h.p. as against 48-hr. cleanings. At $40.00 a year per horsepower,

— a rough estimate of the yearly value — this would mean an annual saving of $400.00 per boiler.

38 The draft loss through a clean boiler will be less than through one not dusted; and while the decrease in load on the fan due to cleaning may not be great, it will at least be appreciable.

DAMPERS

39 Customary practice has been to use a vertical cast-iron sliding damper for the stack by-pass. Considerable difficulty has been experienced in keeping such dampers tight, and where the stack connection, rather than that to the boiler, is the direct connection, difficulty through warping has sometimes occurred. While as far as is known it has never been tried, it would appear that a damper similar in design to reversing valves warranted a trial, in order to reduce leakage at this point. Such a damper would automatically close the passage to the stack when that to the boiler was open, and vice versa, at the same time stopping all air leakage at a point where this might be excessive. It is possible that such a design of by-pass damper would require a small amount of cooling water, but the loss in heat to this water would, it is believed, be more than offset by the gain through reduced air leakage.

EXPLOSIONS

40 In the operation of open-hearth furnaces certain characteristic explosions are liable to occur during the reversal of gas valves. These vary in intensity from slight puffs to rather heavy explosions, and are caused by a mixture of fresh air with carbon monoxide which must pass off during the operation of reversal. Such explosions, often unnoticed with furnaces directly connected to a stack, become very evident where boilers are installed, and every effort is made to keep flues and settings tight. It was feared, and rightly, that with the installation of waste-heat boilers the heavier explosions might have a destructive effect on the boiler settings and connecting flues. For this reason particular attention should be given to proper buckstay construction, and an ample number of explosion doors should be furnished to relieve the pressure within the setting, should explosions take place.

41 A thorough investigation of the trouble from explosions showed that by a proper system of reversing-valve operation the difficulty would be practically overcome; certainly to an extent where sufficient and properly designed explosion doors would obviate the possibility of wrecking or injuring the settings.

42 A method of reversal that experience has shown gives satisfactory results is as follows: Fig. 2 shows in diagrammatic form a typical layout of checkers, valves, flues, stack and boiler inlet, the valves being indicated by *A* to *H*, inclusive, and the checkers *I* to *L*, inclusive.

43 Assuming the gases from the open hearth are passing through the gas checkers *J* and the air checkers *I*, the order of operation is:

1 Turn off steam to producers
2 Close air stack valve *F*
3 Close gas inlet valve *A*

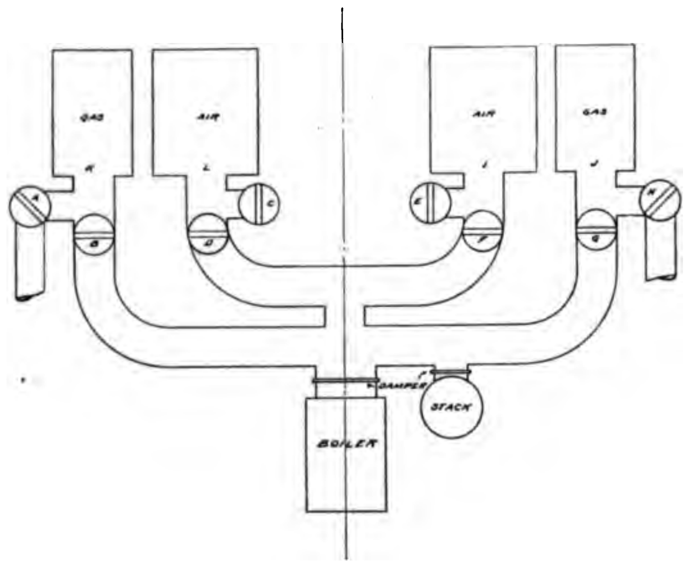

FIG. 2 LAYOUT OF CHECKERS, VALVES, FLUES, STACK AND BOILER INLET

4 Close air inlet valve *C*
5 Open gas stack valve *B* (producer gas held in gas checkers *K* now passed to boiler)
6 Close stack valve *G*
7 Open air inlet valve *E* (air now admitted to the furnace to give proper combustion upon the opening of the gas valve)
8 Open gas inlet valve *H*
9 Open air stack valve *D*
10 Turn on steam to producers.

44 The essential feature in such a sequence of operation is, of course, to avoid opening the stack valves from gas and air checkers simultaneously, or nearly enough together to allow the hot producer gas and air from the checkers to mingle and ignite within the flues or boiler settings. In this system of reversal the air stack valve *D* is opened a sufficient interval after the opening of the gas stack valve *B* to allow the gas pocketed in the checkers *K* to pass away and mingle with the existing gases before the hot air from the checkers *L* is admitted to the flues through opening the valve *D*. In reversing the operation, similar precautions should be taken respecting gas stack valve *G* and air stack valve *F*.

45 In a plant where this system was followed, explosions were reduced from approximately 40 per day (about one each two reversals), varying in intensity, to 4 in 411 consecutive reversals covering a period of five days, and of these 4 but 2 could be called heavy and were directly traceable to a leak in a hydraulic valve which, through the operation of the ram, caused the gas valve to unseat.

46 This system may be modified for different valves and flue arrangements, but the time between opening the stack valves and gas and air checkers is the important point.

47 Another system used is one of interconnected valves, sets of valves being operated simultaneously. Perhaps the best method would be a combination of the two; that is, valves interconnected to assist in manipulation but so arranged that the first order will be followed. The connections should be designed so that sufficient time will elapse between opening the stack valves and gas and air valves to make impossible the mingling of hot producer gas and air.

SAVINGS

48 Great savings have been made possible by the installation of waste-heat boilers with open-hearth furnaces, and some approximate figures are of interest.

49 There have been installed, or ordered, over 90,000 rated boiler horsepower for this class of waste-heat work alone. These boilers are set with over 190 open-hearth furnaces, the annual total production of which is considerably in excess of 9,200,000 tons of steel. Experience has shown that through the use of waste-heat boilers the net cost of production of this material is reduced from \$0.20 to \$0.25 per ton, which would mean a net annual saving for the production weight given above of \$1,840,000, using the lower figure.

50 From another aspect: A conservative estimate of the power delivered by the boilers in this class of work through a year's operation would be 60 per cent of their rated capacity, or 54,000 horsepower. The value of a boiler horsepower in a steel mill varies widely with any number of factors, and ranges from $35 to $50 per year. On the basis of the lower figure, which is certainly conservative, the value of the 54,000 h.p. produced would represent a saving of $1,890,000.

51 Waste-heat boilers, because of their special design, are more expensive than coal-fired boilers, and the expense is increased by the cost of fans, connecting flues, and structural material. The latter expense, of course, is the greatest where boilers are installed with existing furnaces. Regardless of this increased cost, however, the return on the investment is such as beyond question to warrant the expenditure. Mr. Bacon, in the paper to which reference has been made, states that this return on the investment is 60 per cent or over, and recent experience would seem to indicate that this figure is conservative.

52 In the early days of this work, open-hearth steel-furnace operators were fearful that the installation of waste-heat boilers with their fan units would seriously affect the operation of the furnace, and that the production would be cut down, the cause of such reduction being in the necessity of longer heats. By actual experience just the contrary was found to be true, and furnaces equipped with boilers showed a decrease in time of heats.

53 In one instance, a 35-ton furnace equipped with a waste-heat boiler and fan was set in a line with five other furnaces not so equipped. The operation of all furnaces was alike and the size of the heats the same. Over a period of six days the average time of heat for the furnace equipped with the waste-heat boiler was 9.17 hr., while the average for the other five furnaces was 11.17 hr.

54 Eight different sets of furnaces in all were investigated in connection with the saving in heat length. These furnaces varied in rated capacity from 35 to 75 tons, the actual production ranging from 40 to 87 tons per heat. The normal length of heat in the furnaces connected direct to the stack, from records secured previous to the installation of the waste-heat boilers, averaged, for the 8 furnaces, 12.1 hr. The average length of heat for the same furnaces, after the installation of the waste-heat boilers, was 10.8 hr., the saving in time varying in individual furnaces from but a few minutes to 2.9 hr. The total tonnage per heat of the eight furnaces in question was 507 tons. The increased tonnage of the 8 furnaces due to the decrease in heat

length would amount to 42,000 tons per year, or approximately a 12 per cent increase in the total tonnage of these furnaces.

55 There is a further saving due to the use of the induced-draft fan with which these waste-heat boilers are equipped. This is in the increased life of the furnace and results from the fact that a fan makes it possible to run economically a heat from a furnace which under ordinary natural-draft conditions would take so long that it would not pay.

CEMENT KILNS

56 Strictly speaking, the use of waste-heat boilers in the cement industry is not new. The number of installations, however, has been extremely small, and there appear to have been several reasons for the non-development of such boilers in this field.

57 It was feared that the installation of a boiler would seriously affect the successful operation of the kilns, though possibly this was not as important a factor in cement practice as in some other work; and in common with other waste-heat installations there was opposition to the installation of mechanical draft because of the power required. There was a decided feeling that a satisfactory arrangement of kiln and boiler could not be worked out and that, because of the amount of dust carried by the gases, boiler difficulties would develop and maintenance costs be high. Further, with the improvement in kiln design (the first commercially successful rotary kilns were used in this country in 1893) and the tendency toward increased kiln length, gas temperatures were lowered to a point where, as in open-hearth practice, it was not considered practicable, if at all possible, to generate steam economically.

58 Several attempts were made to utilize the heat in these gases through economizers, but such a small proportion of the total heat was reclaimed in this way, and so much difficulty developed through the choking up of the economizers with dust, that such attempts were soon abandoned.

59 Prof. R. C. Carpenter, Mem.Am.Soc.M.E., in the Sibley Journal of Engineering, March, 1904, describes what is believed to be the first installation of a waste-heat boiler with a rotary cement kiln. This was made in 1902 at the plant of the Cayuga Lake Cement Co., a Wickes boiler of 3065 sq. ft. of heating surface being connected to two kilns 7 ft. 6 in. and 6 ft. 6 in. in diameter by 60 ft. long; each kiln having a capacity of approximately 150 bbl. per day. The waste gases from these kilns were passed through the boiler and economizer

and, in addition, coal was ordinarily fired on auxiliary grates. The approximate results obtained from this boiler are given in Table 2. This boiler was removed several years ago because of the necessity for using the space it occupied for other purposes.

60 In a paper, Pulverized Coal Burning in the Cement Industry,[1] Professor Carpenter stated that, as far as he knew, the only successful installation of waste-heat boilers in the cement industry was at the Kosmosdale plant of the Kosmos Portland Cement Co.

TABLE 2 RESULTS OF TESTS OF WASTE-HEAT BOILERS FOR CEMENT KILNS

Test Number	1	2[1]	3[1]	4
Plant	Cayuga Lake Cem. Co.	Sandusky Port Cem. Co.	Burt Port. Cem. Co.	Louisville Cem. Co.
Location	Ithaca, N. Y.	Dixon, Ill.	Bellevue, Mich	Speeds, Ind.
Size of kilns, ft.	....	8 × 100	7 × 60	10 × 150
No. of kilns connected to boiler	2	1	2	2
Boiler	Wickes	Stirling	Stirling	B. & W.
Heating surface, sq. ft.	3,065	3,200	3,200	15,330
Gas weight, lb. per hr.	37,749	52,300	53,000	183,700
Gas per hr. per sq. ft. of h.s., lb.	12.6	16.3	16.6	12.0
Temperatures:				
Gas entering boiler, deg. fahr.	1,800[2]	1,300	1,400	1,400
Gas leaving boiler, deg. fahr.	660	600	700	423
Drop in temp., deg. fahr.	1,140	700	700	977
Draft at boiler damper, in.	1.2[3]	0.45	0.30	6.63
Draft at boiler inlet, in.	....	0.06	0.03	0.81
Draft loss, in.	....	0.39	0.27	5.82
Gross h.p. developed	264	275	278	1,346
Per cent of rated capacity	88	85.9	86	88
Approximate transfer rate (R)	3.19	4.85	4.19	6.01

[1] Results very approximate. [2] Optical pyrometer. [3] Draft at economizer outlet.

A rather thorough investigation of this subject showed that in the early part of 1915 there were at least two other plants utilizing waste kiln gases for steam generation, and securing results that were considered satisfactory. These plants are the Burt Portland Cement Co., Bellevue, Michigan, and the Sandusky Portland Cement Co., at Dixon, Illinois.

61 The installation at the Kosmos plant consists of four Wickes boilers rated at 250 h.p. each, each connected to a kiln 7 ft. in diam-

[1] Trans.Am.Soc.M.E., vol. 36, p. 85.

eter by 80 ft. long with a capacity of 340 bbl. per day. The gases leaving the kilns are given a considerable horizontal travel, then a downward path and finally an upward travel, this direction of flow being with the idea of precipitating dust in the downward pass. These boilers are fired with coal on flat stationary grates, and the waste gases are brought into the side of the boiler setting at a point somewhat above the furnace arch.

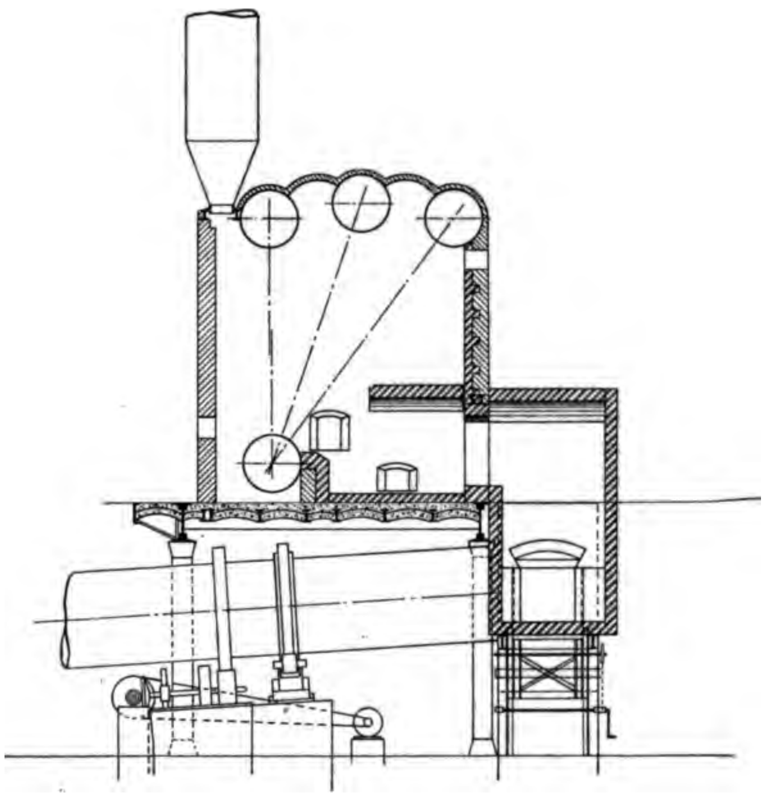

FIG. 3 WASTE-HEAT BOILER FOR CEMENT KILN

62 The installation at the Burt Portland Cement Co. plant consists of four Stirling boilers rated at 320 h.p. each. One boiler is set in connection with two kilns 7 ft. in diameter by 60 ft. long, the capacity of each kiln being approximately 225 bbl. per day. These boilers utilize waste gases alone. This installation is more or less typical of the four described, and the general arrangement of boiler, flue and kilns is shown in Fig. 3.

63 The Sandusky installation consists of seven Stirling boilers, each of 320 h.p., each boiler being connected to a single kiln 8 ft. in diameter by 100 ft. long with a rated output of 540 bbl. per day. These boilers were originally designed for coal in addition to waste heat. Because of low draft available it was found impossible to burn coal economically, and for the past few years firing and ash-pit doors have been bricked up and waste heat used alone.

64 The four installations described were alike in that low gas velocities were used. In the Cayuga Lake and Kosmos plants it is true that induced-draft fans were installed, but this was because economizers were included in the units and not because of any high draft loss through the boiler. In the Burt and Sandusky installations, where the gases were given three passes over the heating surfaces as against two at Cayuga and Kosmos, the baffle openings were considerably larger than standard to minimize the draft loss. As far as can be found, the maximum loss through any of the boilers described was 0.35 in., and it is interesting to compare this loss with that through the modern boiler in this practice.

65 The exit-gas temperatures from all four of the installations were high, and in the case of the Burt and Sandusky plants enabled a reasonable height of stack to give ample draft for proper kiln operation. The gases leaving the kilns were at temperatures varying from 1500 to 1800 deg. and from 1300 to 1650 where entering the boilers. The generation of sufficient steam to warrant the investment in waste-heat boilers was not particularly difficult with these temperatures. With an increase in kiln length, however, and perhaps with the improvement in burning coal in the kilns themselves, these exit-gas temperatures were reduced very appreciably, and gas temperatures at a point corresponding to a boiler inlet were lowered to around 1150 or 1200 deg., a fact that unquestionably delayed the development of waste-heat boilers in this work.

66 Success with the low-temperature gases from open-hearth furnaces indicated possibilities in the cement field that aroused considerable interest in the industry and led, in 1915, to the first installation of the modern design of waste-heat boiler with rotary cement kilns. This was made at the plant of the Louisville Cement Co., Speeds, Indiana. It consisted of a Babcock and Wilcox boiler containing 15,300 sq. ft. of heating surface and 1382 sq. ft. of superheating surface. This boiler is connected to two kilns 10 ft. in diameter by 150 ft. long. The nominal capacity of each of these kilns was considered at the time of the boiler installation approximately

875 bbl. per day, but with improved kiln operation, and because of the ability to remove the products of combustion from the kilns through the installation of the induced-draft fan, this capacity has reached an actual output of 1300 bbl. of cement per kiln per day. The boiler is equipped with a turbine-driven fan capable of handling the gases from the two kilns when operated at their maximum capacity, and to furnish a suction sufficient to overcome the resistance through the boiler, through the settling chambers and connecting flues, and still leave ample at the kiln outlet for proper operation. The approximate arrangement of this boiler with its connecting flues and kilns is shown in Fig. 4.

67 The results obtained from some of the early boilers and those now being obtained from the Louisville installation are given in Table 2. Unfortunately, the data from which the second and third sets of figures are computed are not as complete as could be wished. This is due to the fact that in a number of the tests available coal was burned in addition to the waste gas passing through the boiler, and an attempt made to divide the power developed between the coal and waste gas. A study of different tests, however, some of which were on waste gas alone, enabled us to approximate these results closely, and the figures given are probably correct within reasonable limits. For purposes of comparison with the results secured from the modern waste-heat boiler, they are sufficiently accurate to indicate the enormous gains through the use of such a design.

68 The gas temperature entering the boiler in Test No. 4 is perhaps higher than ordinarily could be expected with kilns of this size, and until the period during which this particular test was run this temperature probably averaged only slightly over 1200 deg. However, the output of the two kilns during this test was at a point which has since been maintained or bettered, and the temperature in Test No. 4 is representative of present and future practice for these kilns. It would appear that the limiting factor in kiln output, and hence, as affecting boiler capacity, in entering-gas temperatures, is governed by the maintenance cost of the kiln lining.

69 That the fears which have retarded the development of waste-heat boilers in the cement industry, as outlined at the beginning of this section, have been more or less groundless, is shown by the results secured at the Burt and Sandusky plants over a period of eight years, and to a more marked degree by the very great success, over a period of eighteen months, of the modern design of waste-heat boiler at the Louisville plant.

DRAFT

70 The installation of waste-heat boilers has not affected kiln operation. At the Burt and Sandusky plants, stacks 135 and 150 ft. high, respectively, give sufficient draft to overcome the frictional resistance through the boilers and allow sufficient for good kiln operation. The baffle openings in these boilers, that is, the gas-passage area, are, as stated, considerably greater than in coal-fired practice, and the draft loss for the weight of gas passing through the boilers is far less than the loss which would occur with standard areas of gas passage. Further, the gas temperature leaving the boilers is so much higher than is found for any reasonable rating for coal-fired boilers, that the draft suction for a given height of stack is appreciably increased.

71 At Cayuga Lake and Kosmosdale, induced-draft fans were installed. The boilers at these plants were of a design giving a low draft loss, and the fan installations were due to the economizers. These increased the total draft requirements and the gases were sufficiently cooled by the economizers to decrease the draft for the given stack height well below that obtained at the Burt and Sandusky plants. The draft at the fan inlet, however, at neither of these plants exceeded 1½ in.

72 The draft loss through the boilers at the Louisville plant is high. The fan installation, however, running well below its maximum capacity, satisfactorily handles the maximum amount of gas from the two kilns and gives ample draft at the kiln throat for a production greatly increased over that secured before the boiler and fan were put into service. While the increase in production is due primarily to changes in kiln operation and improvements in the methods of burning coal in the kilns, the added draft available through the use of the fan and the ability to handle greatly increased gas weights have unquestionably been of assistance in making this greater production possible.

73 When the first installations of waste-heat boilers utilizing the principles of high gas velocity and necessitating mechanical draft were under consideration, the operators, as stated, had a strong feeling against the use of fans because of the power required. Experience has shown, however, that with a turbine-driven fan unit the exhaust is just about sufficient to heat the feed of the boiler it serves. It is to be remembered, too, that with all the kilns in a plant equipped with waste-heat boilers, such boilers will furnish practically the entire power required, and the exhaust from the fan turbines will give the only means of feedwater heating.

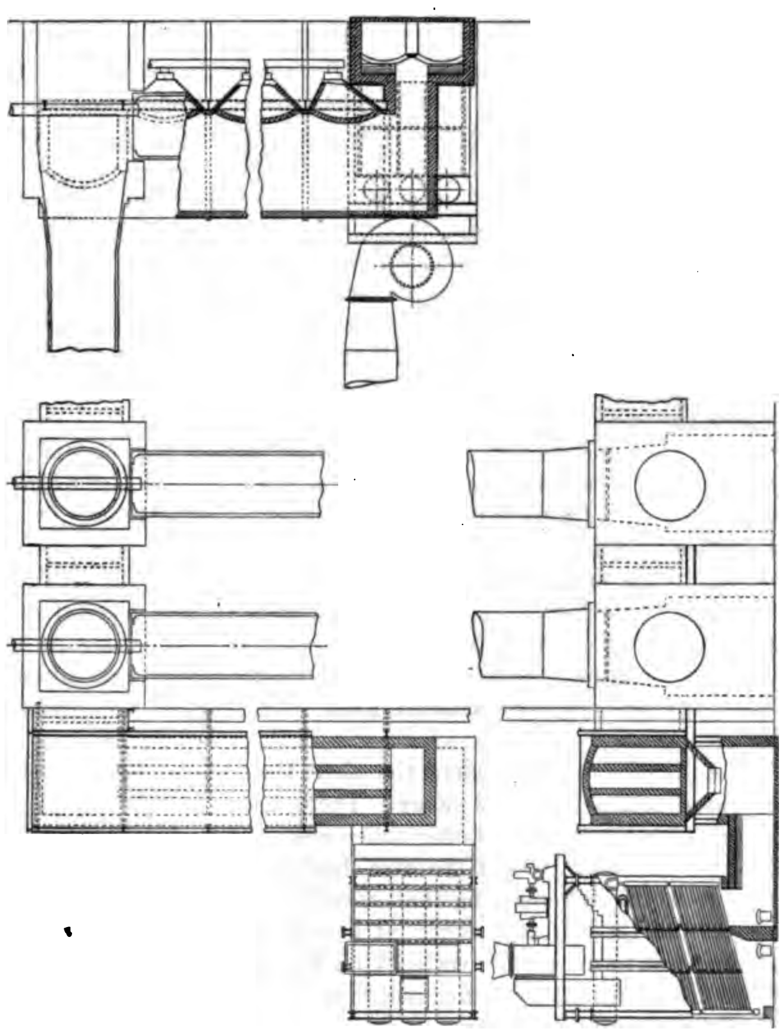

Fig. 4 Waste-Heat Boiler, Cement Kilns and Connecting Flues

CLEANING

74 No difficulty has been experienced in keeping the boilers in this service externally clean. In all but one of the plants investigated the dust has been found to be very light. In the absence of water there is no particular tendency for this dust to stick to the tubes and it may be readily blown off with an ordinary cleaning lance, using either air or steam. Where steam is used, it is perhaps well that this be superheated to minimize the possibility of wetting this dust.

75 At the Burt plant the practice is to dust with air every 12 hours. During dusting periods the by-pass damper to the stack is partially opened to reduce the draft at the boiler damper with the object of settling as much dust as possible within the setting and connecting flue. At this plant the greater part of the dust was deposited on the level floor in the furnace under the arch, Fig. 3, in the pocket behind the mud drum and on the baffle shelves, there being but little dust that collects on the tubes.

76 At the Sandusky plant the boilers are dusted with air every six hours. Here the boiler is not by-passed when dusting and a considerable amount of dust is blown up the stack at such times. One of the boilers at Sandusky has been equipped with stationary blowers, all dusting doors except those necessary for cleaning out the setting having been bricked up. As far as known, these blowers have given satisfactory service. It is to be recommended that where steam is the dusting medium, with such stationary blowers every precaution be taken to prevent condensation from remaining in the piping system.

77 At the Louisville Cement Co. plant the boiler is dusted with superheated steam every 12 hours. There has been no difficulty experienced in keeping the boiler tubes clean, and while there has been some tendency toward the dust bridging over on the superheater tubes, this is something that could be readily remedied by a change in the tube arrangement. It is interesting to note that in this boiler dust forms as an inverted V on the tops of the boiler tubes within half an hour after dusting, and that the increased deposit after this half-hour period until the next dusting interval is almost imperceptible. This is unquestionably due to the high draft suction within the setting; that is, the high gas velocity actually has a scouring action on the tubes.

LEAKAGE

78 As in open-hearth work, it is important in cement waste-heat practice to reduce air leakage to a minimum with a view to increasing temperatures and boiler capacities and reducing the load on the fans. The main source of leakage in this class of work is at the end of the kilns where the rotary enters the dust chamber that ordinarily serves as the stack base. Experiment has shown that leakage at this point can be reduced to a negligible quantity by the use of a properly designed seal ring. The effect of the air leakage on the weight of gas that must be handled by the fan is indicated by the gas analysis. What can be done by way of reducing leakage and increasing gas temperatures is shown in Table 3, which gives results secured in the

TABLE 3 RESULTS OBTAINED BY REDUCING AIR LEAKAGE

	Plant No. 1.		Plant No. 2.		Plant No. 3.	
	Conditions as found	Leakage reduced	Conditions as found	Leakage reduced	Conditions as found	Leakage reduced
Gas Analysis at Stack Base: CO_2, per cent.	11.60	14.44	13.9	21.3	15.43	19.55
O_2, per cent.	11.85	8.86	10.4	4.3	7.99	2.87
CO, per cent.	0.00	0.00	0.0	0.0	0.00	0.00
Gas Temperature at stack base, deg. fahr.	952	1242	1100	1480	1234	1360

investigations of three different plants, the first figures showing conditions of operation as found and the second those after an intelligent effort has been made to better these conditions from the standpoint of a waste-heat boiler.

SAVINGS

79 As in most classes of waste-heat work, the greatest saving is in the value of power produced by the steam generated.

80 The measure of the efficiency of the cement plant is ordinarily the pounds of coal burned per barrel of cement, though presumably this should be corrected to a B.t.u. basis. The amount of coal per barrel depends upon the class of coal, the efficiency of the steam generators and prime movers, and the class of operation in the plant. The coal per barrel is divided into that actually burned in the kilns, which figure usually includes the amount required for drying the coal previous to pulverizing, and that burned in the power plant.

In most plants an amount of coal, approximately 7½ lb. per bbl., is used in drying the rock, and this should be included in the total. The installation of a waste-heat boiler does not of necessity reduce the coal per barrel of cement burned in the kiln, but it is obvious that any steam generated through the use of the waste gases from the kiln results in a reduction of the coal that must be burned in the power plant.

81 Possible savings through the utilization of the available heat in the gases leaving the kiln are indicated by the results secured at the Burt and Sandusky plants where boilers are not of the most highly developed waste-heat design, and to a much greater extent by the results obtained at the Louisville plant, where the boiler is more truly representative of such design.

82 Unfortunately, no figures are available from the Burt or Sandusky installations as to the coal burned per barrel of cement in the power plants previous to the installation of waste-heat boilers, but the savings are seen in the proportion of the total steam generated by the waste-heat boilers.

83 At the Burt plant approximately 60 per cent of the total steam requirements is furnished by the four waste-heat boilers. The coal burned under the direct-fired boilers amounts to approximately 40 lb. per bbl. of cement. On this basis, if the waste-heat boilers were not installed it would be necessary to burn 100 lb. of coal per bbl. of cement in the power plant, a figure which is perhaps slightly high, as the 40 lb. per bbl. now burned includes sufficient coal for banking two 300-h.p. direct-fired boilers, in addition to the one actually operated. At Sandusky the waste-heat boilers develop approximately 65 per cent of the total power required. The coal burned in the direct-fired boilers is approximately 30 lb. per bbl. of cement, on which basis it would be necessary to burn approximately 87 lb. per bbl. in the power plant were no waste-heat boilers installed.

84 As stated, the possibilities of saving in this work through the use of waste-heat boilers are most clearly shown by the results secured at Louisville. Here the boiler is of the most modern design, and actual coal figures are available before and after the boiler installation. In addition to the two 150-ft. kilns attached to the waste-heat boiler, there are two kilns 8 ft. by 125 ft. There are also two kilns 7 ft. by 100 ft., although these are not at present operated. The coal burned per barrel of cement is some 15 lb. greater in the 125-ft. kilns than in the 150-ft. kilns. The plant is operated about nine months in the year.

85 In 1914, before the installation of the waste-heat boiler, the average coal consumption per barrel of cement for the total plant output was 191.2 lb., of which 99.5 lb. were burned in the kilns and 91.7 lb. in the power house.

86 The waste-heat boiler was installed in the early part of 1915 and was put into service May 12th, although the plant was started in this year in March. The total coal consumption for the 1915 operation was 162.3 lb. per bbl. of cement, of which 104.7 lb. were burned in the kilns and 57.6 lb. in the power house. The increase in the coal burned in the kilns per barrel over 1914 was due partly to a greater kiln output and partly to a larger proportion of the total production being from the smaller kilns. The decrease in coal burned in the power house is not representative of what could be secured for the reason that the figures are for the entire year's operation, while the boiler was not put into service for some two and one-half months after the plant was started.

87 For the months of March and April, 1916, the total coal consumption was 150.9 lb. per bbl., 104.05 lb. being burned in the kilns and 46.85 lb. in the power house. The decrease in the power-house consumption here below the 1915 figures is due to the fact that the waste-heat boiler was in operation continuously and because the large kilns were operated at a rate to give a greater capacity from the boiler than was secured in 1915.

88 For one week in May, 1916, conditions approached those that will be secured at this plant when the additional boiler, a duplicate of the present installation, is installed with the two 125-ft. kilns, and possibly with one 100-ft. kiln also connected. During this week the total coal consumption per barrel of cement was 123.5 lb., 94.7 lb. being burned in the kilns and 28.8 lb. in the power house. The reduction in the coal per barrel in the kilns was due to the fact that only the 150-ft. kilns were in operation. As in the case of the 1915 figures, the coal per barrel burned in the power house does not represent what may be expected after the installation of the second waste-heat boiler, for, while the kilns were operating at a rate of 1145 bbl. each per day, or 2290 bbl. for the two kilns, due to the amount of clinker on hand, the grinding machinery was being operated at the rate of 3500 bbl. per day. With the second boiler in operation it would appear conservative to believe that the coal consumption in the power house for all kilns operating will be less than 15 lb.

89 For the purpose of figuring savings, assume that the figure 28.8 lb. per bbl. will be burned in the power house. This would rep-

resent a saving of 62.9 lb. of coal per bbl. of cement over the 1914 figures. For a production of 3500 bbl. per day, a conservative estimate for the Louisville plant, this would represent, for nine months' operation, some 29,000 tons per year. A boiler horsepower is valued at this plant at approximately $25 a year, a figure which, in view of the operating year and the cost of coal, is conservative. On the basis of this value, the steam generated as shown in Table 2 is worth $33,850 a year.

90 In addition to the saving represented by the power developed, there is an appreciable saving in the amount of dust reclaimed from the flue connecting the kiln or kilns to the boiler and from the boiler setting proper. This reclamation is ordinarily fed over again to the kilns with the raw material. That such a saving exists is apparent upon a visit to a cement plant equipped with waste-heat boilers, where it is seen that the deposit on surrounding property is appreciably less than in plants where no boilers are installed.

91 At the Burt plant, where the boilers are dusted once in 24 hr., approximately 2000 lb. of dust are reclaimed from each boiler setting per day and about 8000 lb. from the flues connecting two kilns to one boiler. This represents about 3 per cent of the total output of the kiln.

92 At the Sandusky plant, where the flue arrangement is not such as to give as good a settling action as at the Burt, about 4000 lb. are reclaimed from the boiler setting and connecting brickwork per day, or 1¼ per cent of the total output. At Sandusky, as stated, the boiler is not by-passed in any way during dusting intervals and an appreciable amount is blown up the stack.

93 At Louisville the flue connecting the two kilns to the boiler is long and is especially designed with the object of settling out as much dust as possible before the boiler is reached. This flue is equipped with hoppers at intervals along its length which are emptied each day and the material returned by conveyors to the kilns. The first boiler installed had settling pockets beneath the first pass and beneath the second and third passes, which are cleaned out by hand every few days. The second boiler installed is raised sufficiently to have hoppers beneath the various passes, the pockets to these hoppers being emptied as are the flue hoppers.

94 The amount of reclamation at the Louisville plant is such as to have reduced the raw material fed to the kilns per barrel of cement from 610 to 586 lb., a saving due to the dust collected of 3.9 per cent.

95 The above savings do not represent an increase over the amount reclaimed in plants where no boilers are installed, for at such plants there is a certain amount of material taken out of the collecting chambers which serve as stack bases. The figures given, however, probably represent a saving of at least 100 per cent more in the case of the kilns equipped with boilers over kilns not so equipped.

COPPER FURNACES

96 The copper furnaces with which waste-heat boilers have been installed may be classed under two general heads: Smelting furnaces (matte furnaces) and refining furnaces. In the United States the smelting furnaces are in localities more or less accessible to the copper mines, while the refining, with the exception of plants at Great Falls, Mont., and Tacoma, Wash., is done entirely on the Atlantic seaboard.

97 Smelting furnaces and refining furnaces are fundamentally the same in design, though for metallurgical reasons dimensions of the two classes and individual furnaces in the two classes are varied considerably.

98 *Copper-Smelting Furnaces.* Because of the high temperature of the gases leaving copper-matte furnaces, it was but natural that the attention of engineers was early drawn to the utilization of such heat for steam generation.

99 At the time of the first installation such few waste-heat boilers as were in service in any class of work were single-pass boilers utilizing high-temperature gases and operating in such a way as to interfere in no manner with the working of the primary furnace. It is probably because of this fact that the effect of the boiler installation on furnace operation was apparently given no particular attention at the time of the purchase of the first boilers for this work.

100 The Anaconda Copper Mining Company purchased, in 1903, eight Stirling boilers, rated at the time at 375 h.p. each, for installation with reverberatory smelting furnaces. The boilers were not equipped with firing fronts or grates, but were erected with the standard baffle arrangement for this type of boiler. As soon as the boilers were put into service it was found that the resistance to the gases through the boiler was so great as to affect appreciably the operation of the furnace, and the matter proved so serious that at one time the removal of the boilers was seriously contemplated. It was found, however, that by removing all baffles, the gases passed

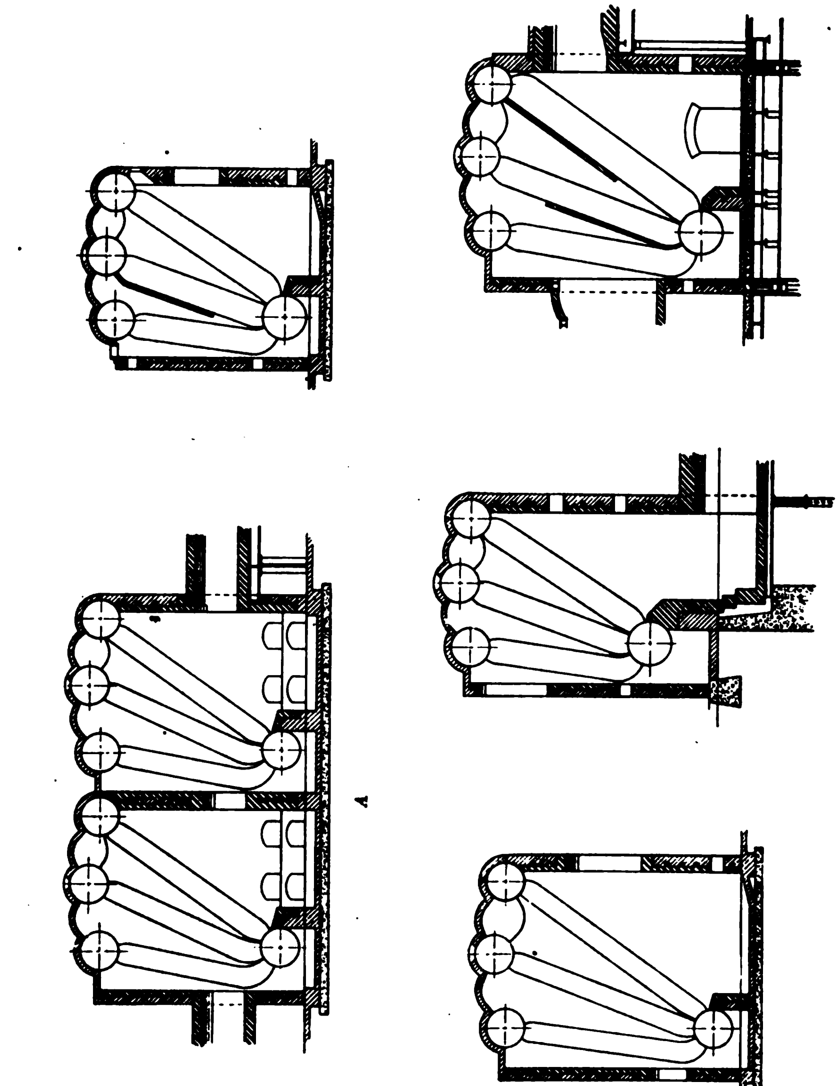

Fig. 5 Typical Waste-Heat Boilers for Copper Furnaces

through the tubes without undue restriction, and that there was no interference with the operation of the smelting furnace.

101 The temperature of the gases entering the boiler averaged approximately 2200 deg. After the removal of the baffles, it was found that the temperature leaving the boiler was considerably in excess of 1000 deg., and in order to diminish this temperature and utilize a greater proportion of the heat, a second boiler, rated at 300 h.p., and also without baffles, was placed in the rear of the first boiler, that is, the two were set in tandem. The approximate arrangement is shown in Fig. 5-A. Under these conditions, for an average gas temperature of 2200 deg. at the entrance of the first boiler, the temperature when leaving the second boiler was found to vary from 620 to 720, the average being approximately 680 deg. It was determined by test that the first boiler developed 470 h.p. and the second 100 h.p., a total of 570 h.p., or, on the basis of 10 sq. ft. per h.p., 86 per cent of the rated capacity of the combined boilers. It is interesting to compare the results obtained from the second boiler, where the entering temperature was somewhat over 1000 deg., with the results secured from the modern waste-heat boiler in open-hearth steel practice, where the entering temperature is in certain installations the same. As against 33 per cent of nominal rating in the case of the copper-furnace waste-heat boiler, in steel plants, for a corresponding entering temperature, some 60 per cent of nominal rated capacity is developed with the modern boiler.

102 The flues in the Anaconda plant were so arranged that one pair of boilers could be cut out and all of the gases from two furnaces be passed through the next set. Under such conditions the temperature of the gases leaving the second boiler varied from 800 to 950 deg., the average being approximately 900 deg. The first boiler, with two furnaces connected, was found to develop 776 h.p. and the second 194 h.p., a total of 970 h.p., or 143 per cent of the combined boilers. While this is considerably over rating for the amount of surface installed, it is interesting to compare the capacity and the exit-gas temperature with the results secured in utilizing waste gases from coke ovens (Table 5), where the temperatures of the gases entering the boiler are considerably lower than in copper-furnace work and the weight of gases approximately the same.

103 The high boiler-exit temperature with two furnaces connected to a pair of boilers, probably explains why double the amount of gas could be handled through the boilers without interfering with furnace operation.

104 The engineers of the Anaconda plant were more or less directly connected with a number of copper-smelting plants, and naturally influenced largely the selection of the type of boiler to be used in this class of waste-heat work. The theory of such selection, based on the Anaconda results, apparently was the furnishing of ample gas-passage area. The operation of smelting in copper furnaces requires large quantities of coal. Because of the nature of the furnaces, air leakage was excessive. These factors led to enormous gas weights that had to be handled, and in the absence of mechanical means of draft, the furnishing of large gas-passage areas through the boilers was the sole means of keeping the draft loss within limits in which satisfactory furnace operation could be secured.

105 Acting on this theory, a great number of installations was made of boilers set in tandem, as at Anaconda, or single boilers of large cross-sectional area. Some of these single boilers were partially baffled, while others were entirely without baffles. In some few cases where single boilers were used, superheaters were placed in the outlet connection from the boiler to reduce exit temperatures and increase the combined capacity. Some typical settings of such boilers are shown in Fig. 5.

106 In 1905 or 1906 an installation was made at the Cananea Copper Company, which differed in general arrangement from what was the common practice at the time, as described above, and was a step in the direction of the more modern design of waste-heat boilers. Here, instead of using single boilers or boilers in tandem, three Stirling boilers rated at 314 h.p. each were set in parallel, the gases from a single reverberatory being divided and passed simultaneously through all of the boilers. With this arrangement, each boiler was handling what might be considered a normal amount of gas at an entering temperature approaching that of coal-fired practice, and under such conditions baffle openings could approximate those standard in coal-fired work without excessive draft loss or interference with furnace operation.

107 The practice at the Cananea plant is described in a paper entitled experiments in Reverberatory Practice at Cananea, Mexico, presented in 1909 before The Institute of Mining and Metallurgy, London, by Dr. L. D. Ricketts.

108 The success of this first installation at Cananea was such that two additional Stirling boilers were purchased in 1906. Three Aultman and Taylor boilers which were available were also reset for waste-heat work at this time, the seven boilers being connected to a

common header flue receiving the gases from two reverberatory furnaces. The general arrangement of furnaces, flues, boilers, economisers and stack is shown in Fig. 6.

109 The operators and designers of this plant feel that because of the greater tendency of dust to lodge on horizontal than on the more nearly vertical tubes, the latter design of boiler is better suited to this particular work. This feeling is hardly borne out by experience in the cement industry, where the waste gases carry as much if not more dust than in copper-furnace practice. It seems probable that if the modern theory of high gas velocity were used in copper-furnace waste-heat boilers, there would be the same tendency for this velocity to keep the tubes "scoured" as was noted in the cement-kiln waste-heat boilers at Speeds, Ind.

110 Difficulties were experienced at Cananea through the clogging of the economizers with dust. This interfered seriously with the draft and hence with the furnace operation. Since mechanical draft was not used in this branch of waste-heat work, the designers of the plant omitted economizers in later installations.

111 The baffle openings in the Stirling boilers at the Cananea plant are approximately what are installed for standard coal-fired practice, and the baffling in the Aultman and Taylor boilers is standard. In some of the later installations where boilers were set in parallel the boilers were but partially baffled.

112 From 1906 to the present time there has been no radical change in the design of the boiler above described for utilizing the waste gases from copper-smelting furnaces. In the large number of installations made during this time all three arrangements — the single unit, the two units in series, and the two or more units in parallel — have been used, though perhaps the single large unit per furnace has been the most popular. The amount of baffling, that is, the gas-passage areas in the different installations, apparently has been governed largely by the individual ideas of the engineer in charge of the installation.

113 In the early installations it was common practice to install 12 sq. ft. of heating surface per horsepower to be developed. The tendency in later installations has been to increase rather than decrease this ratio, and many copper-plant engineers are advocating the installation of 20 sq. ft. per h.p. required from the class of boilers in general use for this work. A comparison of this allowance with that amount necessary in modern design of waste-heat boilers is of interest.

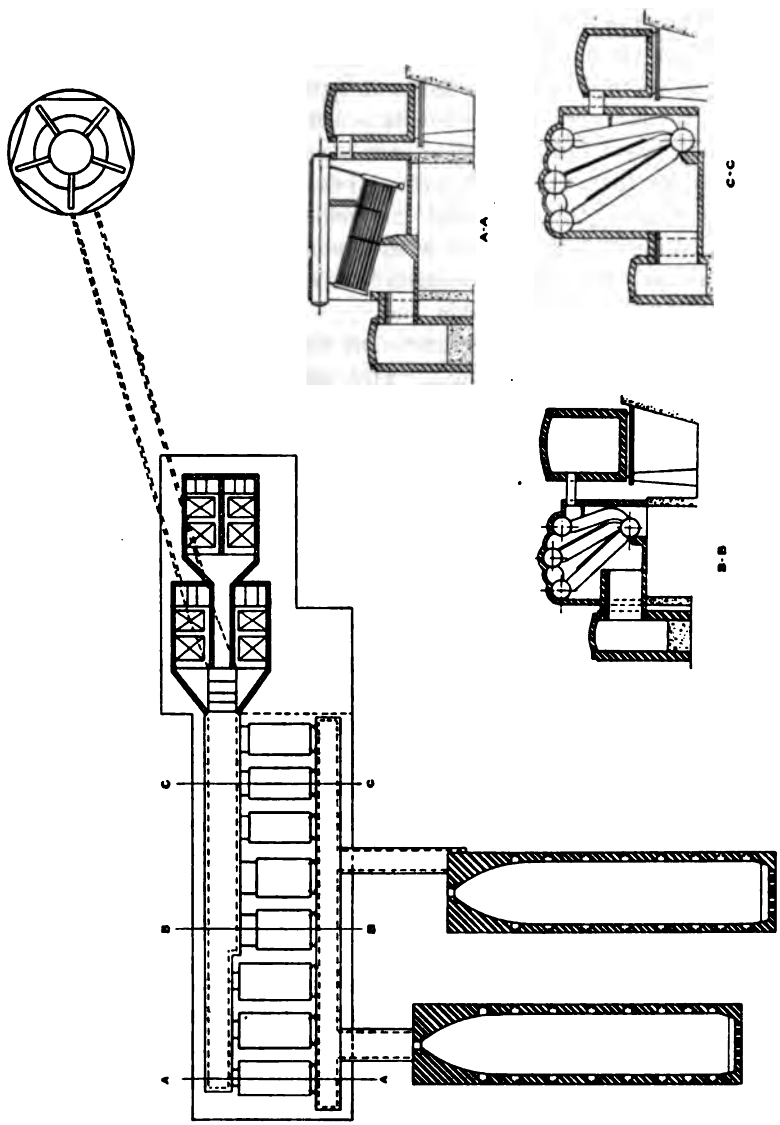

FIG. 6 GENERAL ARRANGEMENT OF FURNACES, FLUES, BOILERS, ECONOMIZERS, AND STACK FOR COPPER-FURNACE INSTALLATION

114 In the early installations in this work, serious fears were expressed that because of the great quantities of sulphur in the gases from smelting furnaces the life of the boiler would be short. These installations were carefully watched on this point over a considerable period, and as far as is known no trouble has arisen from this source; and there has been no action of the sulphur in the gases on the tubes except in such instances where, because of leaks or other causes, moisture has been present to form sulphuric acid. To minimize the possibility of moisture finding its way into the setting, it has been the general practice to use air for dusting tubes rather than steam.

115 It was noticed, however, in the early installations that there was a tendency for small lumps or nodules of solids to form on the surface of the tubes, particularly where the gases first impinged. These nodules, while they apparently had no action on the metal of the tubes, were very difficult to remove with an air jet, and for that reason provision was always made to give access to the heating surface so that this deposit could be removed with a wire brush. This last factor unquestionably had a considerable bearing on the omission of baffles for this work.

116 With smelting furnaces the only saving due to the utilization of waste gases is the steam generated. The amount of this saving may be roughly computed from the power developed as given in the figures in Table 4. That the copper industry considers that this saving warrants an investment in waste-heat boilers is perhaps best shown by the fact that there are today over 43,000 h.p. of boilers set with copper-matte or smelting furnaces alone.

117 *Copper-Refining Furnaces.* As opposed to the continuous operation of a copper-smelting furnace, refining furnaces have a distinct cycle of operation, the gas weights available and gas temperatures varying widely at different periods in this cycle. While at times the temperatures from refining furnaces are as high, or perhaps higher, than those from smelting furnaces, the average over a complete cycle of operation is considerably below the average for an equal period in the smelting furnace.

118 The temperatures from a matte furnace average approximately 2000 to 2200 deg. In the case of the refining furnace these temperatures, over a complete cycle, will vary from about 400 deg. during charging to something over 2000 deg. while melting, and the average for such a cycle, some 30 hr., will be approximately 1400 deg. The length of the cycle and the temperature of the gas available for different periods will vary with the size and operation of the furnace.

As an indication of the variation in results to be expected at different periods of the cycle, some figures on boiler capacities from a 400,000-lb. furnace may be of interest. These figures represent 30 hr. of a 33½-hr. complete cycle. The average horsepower developed during this period was approximately 320.

Period	Duration, Hr.	Average H.P.
Charging	6	36
Melting	16	450
Poling	2	237
Between poling and tapping	1	277
Pouring	5.5	286

119 While the writer is not sure of the exact date of installation of the first waste-heat boiler with copper-refining furnaces, he believes that the Baltimore Smelting and Rolling Company (American Smelting and Refining Company) were the pioneers in this field. This company installed two Stirling boilers rated at 335 h.p. each in 1907. These boilers, as in the case of the first installations with matte furnaces, were furnished with standard baffles. Because of interference with furnace operation, it was found necessary to increase the baffle openings to more than twice the standard. This company has installed over 3500 h.p. for additional furnaces.

120 Other refining companies followed the Baltimore Smelting and Rolling Company in the utilization of this class of waste heat, and the writer is familiar with over 11,000 h.p. of boilers in this class of work.

121 As far as is known, there has been but one attempt made to utilize the theory of high gas velocity in the production of steam from copper-furnace waste gases. This was done in 1914 at the plant of the Raritan Copper Company, Perth Amboy, N. J. This Company purchased three 450-h.p. Babcock and Wilcox waste-heat boilers of the general design described in connection with open-hearth-steel furnace practice, a design necessitating the use of an induced-draft fan for the proper operation of the primary furnace. The Raritan Copper Company has also 2600 h.p. of Stirling boilers in which the principle of high gas velocity is not used, and results from the two designs are available. These results are given in Table 4.

122 In Test 3, the temperature of the gases leaving the furnace was 2190 deg., in the main flue, 1928, and entering the boiler, 1621 deg. This loss was due to leakage and radiation.

123 It will be noted from a comparison of the results of Tests 4 and 5 that the gas weight passing through the Babcock and Wilcox

boiler is 42 per cent less than the weight passing through the Stirling boiler. While the entering-gas temperature is some 120 deg. higher in the case of the Babcock and Wilcox boiler, the exit temperature is approximately 100 deg. lower. In other words, the gas-temperature drop through the Babcock and Wilcox boiler is approximately

TABLE 4 RESULTS OF TESTS OF WASTE-HEAT BOILERS WITH COPPER FURNACES

Test Number	1	2	3	4	5
Plant	Anaconda Copper Mining Co.		Inspiration Copper Co.	Raritan Copper Co.	
Location	Anaconda, Mont.		Miami, Ariz.	Perth Amboy, N. J.	
Furnace	Matte	Matte	Matte	Refining	Refining
Boiler	Stirling	Stirling	Stirling	Stirling	B. & W.
Heating surface, sq. ft.	7,500[1]	7,500[1]	14,060[8]	3,940	4,490
Gas Weight, lb. per hr.	49,950[2]	98,400[5]	160,010	70,100	50,400
Gas per hr. per sq. ft. of h.s., lb.	6.66	13.13	11.38	17.79	11.23
Temperatures:					
Gas entering boiler, deg. fahr.	2,200	2,200	1,621	1,429	1,547
Gas leaving boiler, deg. fahr.	680[3]	890[6]	696	670	584
Drop in temp., deg. fahr.	1,520	1,310	925	759	963
Draft at boiler damper, in.	....	....	0.80	1.15	2.28
Draft at boiler inlet, in.	....	....	0.44	0.56	0.60
Draft loss, in.	....	....	0.36	0.59	1.68
Horsepower developed	570[4]	970[7]	1,111	401	365
Per cent of rated capacity	76	129	78.8	103	81.4
Approximate transfer rate (R)	2.3	3.6	3.4	4.9	4.0

[1] Two boilers in tandem, approximately as *A*, Fig. 5.
[2] Gas from one furnace passing through two boilers.
[3] Range, 620 to 720 deg.
[4] Front boiler, 470 h.p.; rear boiler, 100 h.p.
[5] Gas from two furnaces passing through two boilers.
[6] Range, 800 to 950 deg.
[7] Front boiler, 776 h.p.; rear boiler, 194 h.p.
[8] Two single boilers, approximately as *B*, Fig. 5. Boilers equipped with superheaters in exit flues. Horsepower includes superheat; exit temperatures are after leaving superheaters.

200 deg. greater than through the Stirling boiler for a gas weight of 20,000 lb. less.

124 As was shown in the discussion of some of the results secured with open-hearth furnaces (Table 1), it would be necessary, before a direct comparison would be possible, that the gas weight per square foot of heating surface and the entering-gas temperature should be

the same. On this basis, the total gas weight passing through the boiler of Test 5, for comparable results, would be approximately 80,000 lb. per hr., and with an entering-gas temperature as in Test 4, the boiler would develop 495 h.p., or 11 per cent above its rated capacity. This would indicate a gain of 11 per cent in gross power developed, in favor of the more modern waste-heat boiler.

125 The results on such a basis are, as stated, in favor of the Babcock and Wilcox boiler, though they are not as much better as we would ordinarily expect. This is due to the fact that the fan installation is not entirely suited to the conditions which must be met. Furthermore, as a commercial proposition the fans in the present installation are motor-driven, and the power required for such drive is directly chargeable to the boiler; whereas, if a turbine drive were employed, much of the heat in the steam used for such drive could be reclaimed in feedwater heaters.

126 As in all waste-heat work, the power developed from these waste gases represents a very considerable saving in coal that would otherwise have to be burned under direct-fired boilers. Unlike copper-matte furnaces, however, the value of this power does not represent the greatest saving through the use of waste-heat boilers. The boilers connected to copper-refining furnaces make ideal "dust catchers," and the value of the copper so reclaimed is a highly important item. The writer is not at liberty to give actual figures of savings from this source, but can state authoritatively that in one plant the value of the dust reclaimed from the boiler setting is over one and one-half times the value to the owners of the steam generated. The values of this steam may be approximated from the boiler capacities given in Table 4.

BEEHIVE COKE OVENS

127 While it is true that beehive coke ovens are gradually being replaced with by-product ovens, there are still a very large number of the former in operation. Since there are few, if any, coals from which good coke can be made in beehive ovens that cannot be satisfactorily coked in by-product ovens if properly handled, it is questionable whether more beehive ovens will ever be built. On the other hand, it will be a number of years before all of the existing beehive ovens can be replaced by the more modern type.

128 The total coke production of the country in 1913 was 46,300,000 net tons, and of this quantity the production of by-

product ovens was 12,715,000 tons, or 27.5 per cent of the total. The total estimated production in 1916 is 55,000,000 net tons, of which approximately 19,000,000 tons, or 34.4 per cent, will be produced in by-product ovens. By 1918 it is estimated that the production from by-product ovens will be somewhat over 50 per cent of the total production of the country. These figures offer a basis for approximating the length of time before beehive ovens can be entirely replaced.

129 It is not in the province of the present paper to discuss the merits of the two systems. This class of waste heat is dealt with here because of the very remarkable results that are being secured from modern waste-heat boilers utilizing gases from beehive ovens, and because of the possibilities of saving, due to this utilization, until such time as beehive ovens are replaced. Furthermore, there are available figures from boiler installations made a number of years ago for this class of work, and from a comparison with the results secured today the great advance in the design of waste-heat boilers may be readily seen.

130 From the nature of the beehive-oven coking process, the quantity of gas available from each oven is large. These weights will vary with the analysis of the coal coked and the class of coke being made. The analysis of the gases escaping from the individual ovens will vary widely during different periods of the coking operation. Where a large number of ovens, however, are connected by a single flue to a boiler, or boilers, the average analysis at the boiler entrance will not show a particularly wide variation due to the fact that different ovens in the range or block are at different stages of coke making. At certain periods during operation large quantities of carbon monoxide pass off from the ovens and burn in the flues; in no plant investigated has there been noticed any secondary combustion within the boiler. For estimating purposes, the gas weight available may be taken as from 6 to 7 lb. per lb. of coal coked per hour.

131 The temperature of the gases leaving the individual ovens is high, probably averaging between 2000 and 2200 deg. The temperature of these gases as they enter any boiler that is installed will depend upon the length, design and location of the connecting flues. Some typical entering temperatures are given in Table 5.

132 As far as is known, the first water-tube boilers installed in this country for this class of waste-heat work were placed in operation in 1900 at Greensburg, Pa. Due to improper design of connecting

flues, these boilers were not successful in the utilization of the waste gases, and shortly after their installation were changed to coal-fired boilers.

133 Rather complete figures are available from three different plants in which boilers are installed for this class of work, the first two representing early waste-heat practice and the third representing the modern waste-heat boiler.

TABLE 5 RESULTS OF TESTS OF WASTE-HEAT BOILERS FOR BEEHIVE COKE OVENS

Test Number	1	2	3	4
Plant	Priestman Col. Ltd.	Frick Coal & Coke Co.	....	....
Location	Newcastle-on-Tyne	York Run, Pa.	....	....
Boiler	Stirling	Stirling[1]	B. & W.	B. & W.
Heating surface, sq. ft.	1,610	10,800	10,200	10,200
Gas weight, lb. per hr.	23,200	83,650	125,500	155,100
Gas per hr. per sq. ft. of h.s., lb.	14.4	7.7	12.2	15.2
Temperatures:				
Gas entering boiler, deg. fahr.	1,720[2]	1,804	2,329	2,158
Gas leaving boiler, deg. fahr.	650	490	463	477
Drop in temp., deg. fahr.	1,070	1,314	1,866	1,681
Draft at boiler damper, in.	0.56	0.56	4.0	4.4
Draft at boiler inlet, in.	0.24	0.30	1.9	2.0
Draft loss, in.	0.32	0.26	2.1	2.4
Horsepower developed	187	824	1,756	1,956
Per cent of rated capacity	116	76	172	192
Approximate transfer rate (*R*)	4.7	3.2	5.6	6.8

[1] Three boilers, each of 3600 sq. ft. heating surface. [2] Temperature leaving ovens 1970 deg. fahr.

134 The first plant is that of the Priestman Collieries, Limited, near Newcastle-on-Tyne, England. Here a Stirling boiler, rated at 161 h.p., was connected to 22 beehive ovens producing coke from an average of 3800 lb. of coal per hr. for the 22 ovens, or 173 lb. per hr. per oven.

135 The second is at the York Run, Pa., plant of the H. C. Frick Coke Company. In this installation there are three Stirling boilers having approximately 3600 sq. ft. of heating surface each, though the boilers were rated at the time of purchase as 300 h.p. each. These three boilers were connected to a range of 50 ovens, and during the

week devoted to boiler testing, there were on an average 44 ovens in service which were coking coal at the rate of 13,800 lb. per hr., or 314 lb. per oven per hour.

136 It has not been possible to secure permission to name the owners of the third and distinctly modern installation, although they have kindly consented to the publishing of some of the results secured from the first unit put into service. This installation consisted of a Babcock and Wilcox boiler 18 sections wide, each section being made up of twenty-six 20-ft. tubes and containing 10,200 sq. ft. of heating surface. The boiler is equipped with a superheater and furnished with a turbine-driven induced-draft fan.

137 During the time the tests were run on the Babcock and Wilcox boiler the ovens were being operated to meet plant demands for coke, some ovens making 24-hr., some 36-hr., and others 48-hr. coke. The speed of the fan connected to the Babcock and Wilcox boiler was regulated to handle an approximately constant weight of gas, and as ovens were cut in or cut out, other boilers would be put into service and taken off again when the gas weights were reduced. Because of this fact it was practically impossible to determine the actual amount of coal coked corresponding to the capacity developed by the Babcock and Wilcox boiler. It is possible, however, to check pretty definitely the gas weight passing through the boiler, and from this weight approximate the coking rate corresponding to the horsepower developed.

138 The performance of the first boiler unit has been so satisfactory that the owners have purchased, since its installation, six additional units for use with their ovens. These are similar to the one described, except that they are 27 sections wide instead of 18, each unit containing 15,300 sq. ft. of heating surface, or a total of 9180 h.p. for the six units.

139 The results of two of a series of tests on this Babcock and Wilcox boiler, together with the results secured at the other two plants described, are given in Table 5. In this table the gas weights, while approximate, are certainly within 10 per cent of the actual weights.

140 The baffle arrangement in the boiler at the Priestman Collieries is not know. It is probable, however, that the gas-passage areas were somewhat larger than standard, as the draft loss through the boiler is considerably less than would be expected for an equal weight of gas in coal-fired practice. The exit temperature is high but, considering the design of boiler and the weight of gas, cannot be

considered excessive. In the report covering the series of tests from which this was taken, a statement is made to the effect that the capacity obtained did not represent the maximum, as, due to a leaky damper, all of the gases from the ovens were not passed through the boiler.

141 In this plant the boiler was at the end of a range of 22 ovens. The flue was thus rather short, and because of the very low draft requirements of the ovens and natural-draft stack, provided sufficient draft for proper oven operation.

142 The test at the Frick plant extended over a full week's run of 168 hr. As stated, 44 of the ovens were in operation during the test, the ovens making 48-hr. coke and half the number being charged each day with the object of maintaining even gas conditions. Gas weights and temperatures did vary considerably throughout the week. The lowest temperature noted was approximately 1500 deg., while the highest was 2075. The gas weight here corresponds to about the weight that would be passed through the boilers if they were being operated at about rating, coal-fired. The baffles in these boilers were standard, and the draft loss checks the statement made as to rating.

143 Under such conditions, the exit temperatures, which varied during the test from approximately 475 to 550 deg. and averaged, as given, 490 deg., are no lower than might be expected.

144 The boilers at the Frick plant were located at the middle of a range of 50 ovens, so that the length of flue on either side was not excessive, and the draft requirements could be readily met by a stack 150 ft. high.

145 In Tests 3 and 4, the interesting features are the low exit-gas temperatures and the high capacities. The exit temperatures of 463 and 477 deg., for ratings of 172 and 192 per cent respectively, are lower by a considerable amount than would be obtained with coal-fired boilers at equal ratings. The design of the boiler to give the high heat-transfer rates noted offers sufficient explanation for these temperatures. In other tests of the Babcock and Wilcox boiler in this series, where the gas weights were such as to give approximately 96 and 128 per cent of rating with entering-gas temperatures of 2013 and 1811 deg., the exit temperatures were 437 and 428 deg., respectively.

146 At this plant the boilers were at the end of the blocks of 100 ovens and the connecting flues were of considerable length. This explains the necessity for the greater draft suction found at the boiler

inlet in this boiler as compared with that in the plants previously described. The draft loss through the boilers is high, but is no more than would be expected when the weight of gas and the area of gas passage are considered. The fan, as stated, was turbine-driven, and a large proportion of the power required for this drive was returned through the exhaust to the feedwater.

147 In this class of work the saving due to the utilization of waste heat is solely in the value of the steam generated. The amount of such saving may be approximated from the boiler capacities given in Table 5. As stated, the gas weight available will be approximately 6.5 lb. per lb. of coal coked per hour. On this basis it will be noted from Tests 3 and 4 that 1 h.p. was developed for 11.0 and 12.2 lb. of coal coked per hr., respectively, and it would seem conservative to state that with the modern design of waste-heat boiler a horsepower may be developed on the waste gases from 15 lb. of coal coked per hour. At such a return, even granting that beehive ovens will ultimately be replaced with by-product ovens, it would seem that, in numerous plants, an installation of waste-heat boilers would pay for itself many times before such a change could be made. It would be entirely possible, too, in these installations, to design the boilers in such a manner that at the time of replacing the beehive ovens the boilers could be dismantled and reset either for burning coal or coke breeze, or to be fired with by-product coke-oven gas.

HEATING FURNACES

148 The first boilers for any class of waste-heat work were unquestionably installed with heating furnaces of different descriptions, and the early history of boilers with heating furnaces is in reality the early history of the utilization of waste heat in general. While the writer does not know the actual date of the first installation, it is certain that it was previous to 1872. Before that date such boilers as were used in connection with heating or puddling furnaces were of the cylinder or two-flue design.

149 An article in the Iron Age of April 6, 1893, describes the early installations of water-tube boilers for this class of work and gives some more or less complete results of the performance of a number of these boilers. The first of these installations was made at the plant of the McCullough Iron Works, Wilmington, Del., in 1874, and consisted of two boilers of the Babcock and Wilcox design made up of seven sections of six 16-ft. tubes. Each boiler contained 860 sq. ft.

of heating surface and in accordance with the practice of the day were rated on the basis of 11½ sq. ft. per h.p., or 75 h.p. To quote the journal mentioned: "These boilers, erected seventeen years ago, have been in constant use ever since and have given entire satisfaction."

TABLE 6 RESULTS OF TESTS OF WASTE-HEAT BOILERS WITH STEEL FURNACES

Test Number	1	2	3[3]	4[3]	5[3]
Plant	Penn. Bolt & Nut Co.	Cambria Steel Co.	Bethlehem Steel Co.		
Location	Lebanon, Pa.	Johnstown, Pa.	So. Bethlehem, Pa.		
Furnace	Puddling	Heating	Structural Mill Reheating Furnace		
Boiler	3 Pass B. & W.	Single Pass B. & W.	B. & W.	B. & W.	B. & W.
Heating surface, sq. ft.	1196	2,998	5,840	5,840	5,840
Gas weight, lb. per hr.	....	16,150	87,571	63,932	80,757
Gas per hour per sq. ft. of h.s., lb.	....	5.4	15	11	13.8
Temperatures:					
Gas entering boiler, deg. fahr.	....	1,990[1]	1,745	1,071	1,445
Gas leaving boiler, deg. fahr.	542	729[2]	436	401	418
Drop in temp., deg. fahr.	....	1,261	1,309	670	1,027
Draft at boiler damper, in.	0.32	0.23	1.87	1.15	1.50
Draft at boiler inlet, in.	....	0.19	0.68	0.39	0.54
Draft loss, in.	....	0.04	1.19	0.76	0.96
Horsepower developed	73.2	152.8	784.1	326	542
Per cent rated capacity	61	50.9	134	56	93
Approximate transfer rate (R)	....	1.7	6.1	5.0	5.4

[1] Optical pyrometer.
[2] Varied at different periods of furnace operation from 604 deg. to 920 deg.
[3] Tests 3, 4 and 5 on same boiler. No. 3 represents 12-hr. period during which the structural mill was in operation, No. 4 a 12-hr. period with the mill not operating, and No. 5 a continuous run of 104 hr. during which the mill operated 56 hr. and was not in service 48 hr.

150 A number of boilers similar to these were installed after 1874 in the different iron and steel plants throughout the country. Test 1, Table 6, is representative of the performance of this design of boiler. This test was conducted by J. de Kinder at the plant of the Pennsylvania Bolt and Nut Company, Lebanon, Pa. At this same plant a test on a cylinder boiler in the same service showed exit temperatures considerably in excess of 1000 deg. as compared with 542 deg. for the water-tube boiler, and this temperature difference probably

represents the difference in efficiency of the two types of boiler in this class of work.

151 It is of interest to note that the early waste-heat water-tube boilers with heating furnaces were of the three-pass design. There were, however, a number of factors in heating-furnace work which enabled satisfactory results to be obtained from a design that later was considered more or less impracticable for other classes of waste-heat work. The chief of these was the fact that the draft required at the outlet of the ordinary heating furnace was very low. The boilers were ordinarily set directly over the furnace, and the draft loss between the furnace and boiler was negligible. Furnaces were not driven at the time of the early installations at a rate that is at all comparable with present-day capacity, and the gas weights to be handled were low. Further, the boilers were wide and long compared to the height, so that the gas-passage areas were large in comparison to the heating surface. The high exit temperatures ordinarily found enabled a given height of stack to give a greater draft than would the same height under coal-fired conditions.

152 In 1892 a type of single-pass waste-heat boiler was introduced by The Babcock and Wilcox Company. It is not quite clear why such a design was developed, but the presumption is that with increased furnace capacities the draft loss through a sufficient amount of heating surface in a three-pass design was great enough to interfere with the operation of the furnace. These boilers were of standard sectional Babcock and Wilcox design, but with tubes of 8, 9 and 10 ft. in length. The heating surface required to cool the gases sufficiently was obtained in the height of the boilers, and these were made from 18 to 27 tubes high.

153 The first boilers of this description were installed in the rolling mill of N. E. Ayre and Company, Portland, Ore., in 1892; one boiler being set over a box-scrap-heating furnace and the second in connection with a small billet-heating furnace. The Iron Age article, to which reference has been made, gives some figures on the performance of these boilers, but inasmuch as no temperature or draft measurements are reported, the figures are not included in Table 6. There are included in this table, however, results secured from a boiler of similar design installed in 1901 at the plant of the Cambria Steel Company, Johnstown, Pa. This boiler was of the single-pass type, made up of 10 sections of 27 tubes 10 ft. long and rated at 300 horsepower.

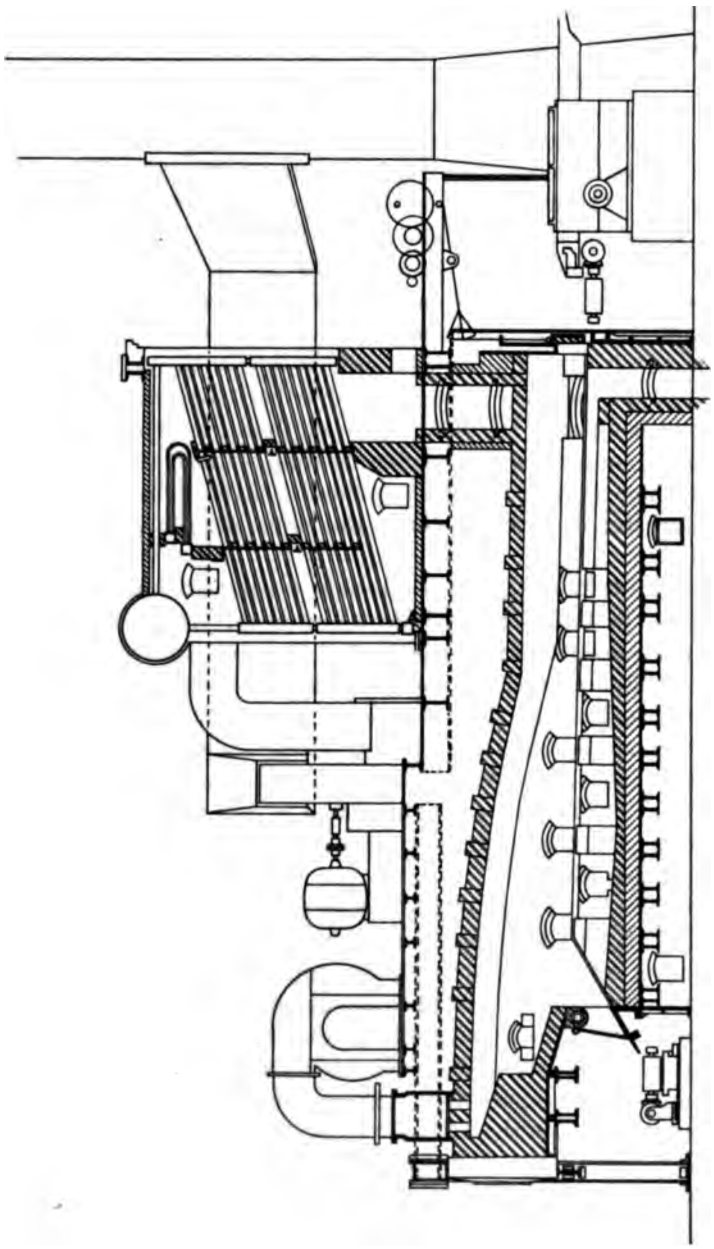

FIG. 7 REHEATING FURNACE AND WASTE-HEAT BOILER ARRANGEMENT

154 The results as compared with the earlier three-pass boilers are considerably in favor of that type. Some 7500 h.p. of the single-pass design were installed, and in general the results secured were considered commercially satisfactory. Occasional installations of this design of boiler are still made. Because of the construction the boiler was expensive, and probably for this reason was never as popular as the next design of single-pass waste-heat boiler.

155 This was the Cahall waste-heat boiler, introduced in 1894 by the Aultman and Taylor Machinery Company. This was a single-pass vertical boiler in which the gases were introduced through an annular ring in the mud drum. Baffles were placed in the cone between the tubes with the object of throwing the gases into the tubes, but in view of the temperatures frequently encountered, it is questionable how long such baffles remained in place.

156 The results secured with this type of boiler were about in line with those of the previous single-pass type, though for a given set of conditions the exit temperatures were probably slightly higher. The Cahall boilers were popular, however, in the iron and steel industry, and over 20,000 h.p. were installed. A number of these boilers were fired with blast-furnace gas and were not, therefore, waste-heat boilers.

157 Other types of boilers were used in this waste-heat work, but the three designs described were by far in the most general use. All of the boilers used were such as to give a minimum draft loss for the amount of gas to be handled, and in this way interfered in no manner with the operation of the primary furnaces.

158 In 1914 the first waste-heat boilers of strictly modern design and utilizing the theory of high gas velocity were installed for this class of work by the Bethlehem Steel Company. These boilers were of the general design described in connection with open-hearth-steel practice, and were made up of 19 sections of 17 tubes 16 ft. long. They were rated at 584 h.p. each, were equipped with superheaters designed to give approximately 75 deg. of superheat, and furnished with motor-driven induced-draft fans. They were set in connection with two 28-in. structural-mill reheating furnaces, the approximate arrangement of furnace, boiler and fan being shown in Fig. 7. An extensive series of tests was run on one of these boilers in the early part of 1915. During these tests the structural mill was operated during certain shifts of 12 hr., while during other shifts the mill was not in service, though the furnace was kept hot. Under these conditions, since the temperatures were considerably lower while the mill

was in operation, tests covering a wide range of gas temperatures are available, and the boiler performance under such conditions may be noted. Table 6 gives three of the tests on this boiler. Test 3 of this table represents a 12-hr. period during which the mill was in operation, Test 4 a 12-hr. period during which the mill was out of service, and Test 5 a continuous run of 104 hr., during which the mill was being operated for 56 hr. and was out of service 48 hr.

159 As in all previous tests, the interesting feature brought out by a comparison of early and modern results is the increase in capacity and decrease in exit temperatures in favor of the modern design. These lower exit temperatures further are secured with gas weights per square foot of heating surface greatly in excess of early practice.

160 The draft loss through the modern boiler, as indicated in the last three tests of Table 6, is reasonably high, though not nearly as great as found in open-hearth and cement-plant waste-heat practice. Since the fan on this boiler was motor-driven, none of the power required for such drive could be reclaimed in the boiler feed. The amount of power required however was small, the maximum for the whole series of tests being about 2 per cent, while the average was 1.55 per cent of the total power generated by the boiler.

161 In this class of work the only saving through the use of waste-heat boilers is the value of the steam generated. This, however, is certainly a considerable item, and it would appear that there are numerous industrial furnaces that are similar to heating and puddling furnaces and where the temperatures are comparable, with which the installation of waste-heat boilers can show a handsome profit on any investment made.

MISCELLANEOUS

162 While by far the greater number of waste-heat boilers in service are in the industries outlined above, there are numerous installations in plants of different character. In this miscellaneous class the number of installations in any single industry is small, but it appears advisable to make reference to certain of these, if only to give an idea of the wide and varied field for development in the use of waste heat.

163 *Zinc Furnaces.* The writer has knowledge of over 11,000 h.p. of waste-heat boilers set in connection with and utilizing waste gases from zinc furnaces. The temperatures in this class of work are high, being ordinarily in excess of 2000 deg. with certain designs of

furnaces and probably averaging well over 1700 deg. with all classes. The draft requirements for the zinc furnace itself are low and, in fact, a number of furnaces are operated with a slight pressure at the exit. Because of these facts, the larger part of the installations with such furnaces has consisted of boilers that are practically standard in design. With furnaces in which the ore treated is in contact with the gases, fears of the building-up on the tubes of an objectionable deposit led to a modification of boiler design to give more than the customary access to the heating surfaces, in order that all possible might be reached and brushed down should occasion demand. In refining furnaces, where the gases do not come in contact with the metal treated, there is, of course, no danger of such deposit, and standard boiler designs work out satisfactorily.

164 The results secured from these standard boilers have been entirely satisfactory, and from a commercial standpoint the return in all cases has shown that the expense of the installation has been more than warranted.

165 One of the first concerns using waste-heat boilers in this class of work for years installed units that essentially were of standard design. As the modern waste-heat boiler was developed, however, this company was persuaded to install such a unit, though the gas velocities were not quite as high as could be desired for the best results. No complete figures of the performance of the modern design as compared to the standard design are available, but the exit-gas temperatures from the two designs are indicative of the higher efficiency of the former. One large installation of the modern type of waste-heat boiler has been purchased. These units are not as yet in service, however, and no results are available, but from our knowledge of waste-heat boilers in general there appears to be no reason why the performance should not be in line with known performances in other branches of the work.

166 *Nickel-Refining Furnaces.* The International Nickel Company (Orford Copper Works) has in service over 3000 h.p. of waste-heat boilers set in connection with nickel-refining furnaces. As in the case of zinc furnaces the temperature of the waste gases is high. While it varies during different portions of the operation, the average is probably in the neighborhood of 1700 deg. when leaving the furnace. The boilers at this plant are of practically standard design, such as used for coal fuel, insofar as baffle arrangement is concerned. No complete test figures are available, but from data at hand the boilers are developing over 90 per cent of their rated capacity with exit-gas

temperatures of approximately 600 deg. With such a temperature leaving the boiler and a draft resistance through the boiler corresponding to about rating in coal-fired practice, the low draft requirement at the furnace exit is readily met by a 100-ft. stack.

167 These boilers, as stated, are of practically standard design. As compared with the exit temperature of 600 deg. actually being obtained it would be entirely possible, with the modern type of waste-heat boiler, to reduce this to 450 deg., corresponding to an increase of over 13 per cent in capacity. On the other hand, the design of boiler in use is developing approximately its rated capacity with gases that previous to 1911, the date of the first installation, were wasted. The success of the first units installed was such as to cause this company to duplicate them in later installations, and since these waste-heat boilers have made it possible to shut down a large portion of the coal-fired boiler plant, there was no apparent reason for changing the first design.

168 In this class of work, as with copper-refining furnaces, the saving due to the reclamation of dust within the setting is an item that compares favorably with the saving due to the steam generated from the waste gases.

169 *Gas Benches.* Several installations of the modern type of waste-heat boiler in small units have been made in connection with illuminating-gas benches. The gas temperatures leaving the benches vary somewhat at different periods of the cycle, but where a number of benches are connected to a single boiler — and this is necessary to get a practicable weight of gas — the temperatures entering the boiler are reasonably constant. Complete figures are available of a test run on a modern design of waste-heat boiler in this class of work. The boiler in question has 1330 sq. ft. of heating surface, is equipped with a superheater, and furnished with an induced-draft fan. The boiler is connected to five gas benches, using an average of 211 lb. of coke per hr. per bench. The average temperature of the gases leaving the benches in the test was 1484 deg., while the average temperature entering the boiler was 1225 deg. The boiler test was run at the time the gas benches were first put in operation, and the temperatures, both leaving the benches and entering the boiler, are in all probability somewhat higher than would be the case after normal operating conditions were established.

170 Under the above conditions, the boiler cooled the gases to approximately 450 deg. (though this figure was undoubtedly affected somewhat by leakage at the damper) and developed 106 h.p., or 79.3

per cent of its rated capacity. The draft loss through the boiler was slightly less than 1 in. While this is not high as compared with other waste-heat work, when it is remembered that the boiler must be at some distance from the furthest bench and hence the connecting flue be of considerable length, and also that approximately 0.75 in. or 1 in. of draft is necessary at the bench outlets, the impracticability of a natural-draft stack, where a boiler is installed in this class of work, is obvious.

171 In the plant in question the waste-heat boiler generated not only sufficient steam for the requirements of the coal-gas section of the plant, which are low, but a surplus that was almost sufficient for the operation of the water-gas department of the plant.

172 *Oil Stills.* At least two installations of the modern type of waste-heat boilers have been made in connection with oil stills, though, aside from the statement to the effect that these boilers are giving satisfactory service, nothing definite on their performance is available.

173 In this class of work it would appear that the amount of gas available is dependent, to a certain extent, upon the operation of the still, though from reports at hand it would seem safe to figure on approximately 22 to 24 lb. of gas per lb. of oil burned under the stills. The temperature of such gas at a point corresponding to the boiler inlet is approximately 1000 deg., and because a number of stills are connected to a single boiler, this temperature is practically constant.

174 With the modern design of boiler, as in other waste-heat work, the draft loss is high, and as a draft of approximately 1 in. is required at the stills for proper operation, induced-draft fans are necessary.

DISCUSSION

Alexander G. Christie. In connection with the cement-mill work, the noteworthy point brought out in the paper is that there is not only a large reduction of fuel consumption by the utilization of this waste heat, but also a better dust recovery, and in some sections this latter is a question of considerable importance.

The performance of the waste-heat boilers described seems to depend largely on high gas velocity being maintained by an induced draft, and that, naturally, leads one to question whether the same principle could not be applied to the standard boiler.

The late Professor Nicolson experimented with high gas velocities in boilers a number of years ago in England, and his results indicated that considerable gains were possible. At the time, however, it was felt that too much power had to be expended to obtain these high velocities. With forced-draft stokers and with economizers it may be possible to modify our present boiler baffling to secure better results on our standard boilers.

L. D. Ricketts[1] (written). The Cananea plant is the oldest one installed with the header type of flues for the distribution of the waste-heat gases to the boilers. In more modern smelters in the Southwest, Stirling boilers or ones of similar type but of much larger capacity are used. They are equipped with superheaters. Economizers have not been installed recently on account of their tending to become foul and to cut down the draft in the furnaces.

The new plant of the International Smelting Co., near Globe, Arizona, has three reverberatory furnaces, each 21 ft. wide by 120 ft. long, and seven waste-heat boilers of a capacity of 713 boiler h.p. each, and they are supplied with superheaters which furnish 50 deg. of superheat to the steam, which is generated at a pressure of 195 lb.

We find it advantageous, however, to use three boilers to a furnace, and now that we have to increase the size of the plant and add an additional furnace, we contemplate installing three more boilers of this size so that we can do this in operating three furnaces and have one boiler as a spare.

It may be of interest in this connection to give some idea of the amount of power recovered at a plant like the one in question.

Each of the furnaces treat about 500 tons of solid charge per day, and with two furnaces running continuously the plant smelts about 30,000 tons of solid charge per month. The evaporation from and at 212 deg. in the oil-fired boilers in the power house is 16.46 lb. of water per pound of oil. The evaporation in the reverberatory furnaces is 7.32 lb. of water per lb. of oil. In other words, a pound of oil burned for smelting purposes in the reverberatory furnaces yielded on an average (for the first ten months of 1916 at the International Company's plant) 44.77 per cent of the power such oil would yield if burned under its boilers. The gross oil consumed in smelting was 0.856 barrel per ton of solid charge, and of this 0.475 barrel was charged to smelting and the balance to steam generated.

Warren B. Lewis (written). The extraction of waste heat from furnace gases has not always been successful, and the unfortunate

[1] 42 Broadway, New York.

examples have been heralded fully as much as the successful ones. Boiler economizers are the most widely known waste-heat extractors; and their success has been due, in no small measure, to a thorough understanding of the characteristics of the gases to be handled. The conditions surrounding steel furnaces have not been so thoroughly understood, nor the requirements of the furnace so well appreciated, so that the apparatus for recovering waste heat has not been standardized.

The following description of a plant using waste-heat boilers in which the transfer rate is low, is cited to show what has been accomplished from a different reasoning point to that employed in the paper.

Admitting at the start that mechanical draft is practically a necessity, the regulation of draft takes place in the flue between the furnace and the boiler, or, to put it another way, a certain definite draft must be maintained at the end of the checkers. What happens beyond that point is of comparatively small importance.

The furnaces were fired with producer gas made from bituminous coal. The amount of coal consumed in a year was 8400 tons. The gases between the producers and the furnaces were analyzed, as were also the gases between the checkers and the stack, and a determination made of the weight of gas issuing from the stack. The temperature of the gases averaged 1050 deg. fahr., and the boilers were designed to abstract 550 deg. fahr.

The boilers chosen were of the Manning type, with tubes 20 ft. long; and draft was produced by means of a motor-driven steel-plate fan mounted on a platform at the top of the boilers. A rotatable steam-jet tube blower was installed, by means of which the tubes could be blown out as often as desired with a minimum of labor. The ground space occupied was small, and the protection necessary from the weather inexpensive. The boilers were equipped with automatic feedwater regulators.

The early calculations in connection with this installation showed that there should be an average output of 212 boiler h.p. One year after the investigation was made the boilers were in operation, and the actual boiler horsepower developed was 227; the temperature of the flue gases was 418 deg. fahr. The total square feet of heating surface was 6600, and the transfer rate less than 2. The actual power required to drive the fan was about 2½ per cent of the net return from the boilers.

The tests showed a comparatively high percentage of recovery,

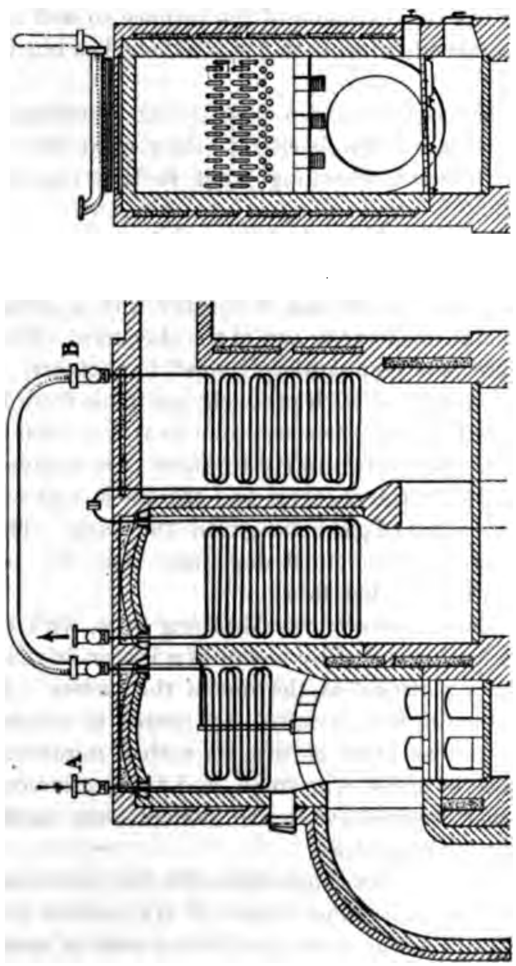

Fig. 8 Independent Superheater Utilizing Heat of Waste Gases.
Capacity, 40,000 Lb. Steam per Hour

as indicated by the low temperature of the flue gases issuing from the fan.

In order to recover a high percentage of heat with a low transfer rate, a large amount of surface must be used; and it is simply a question of what that additional surface costs as compared with the cost of a high-pressure drop.

B. N. Broido[1] (written). Waste heat is also very often used in Germany to superheat steam. In cases where for some reason superheaters cannot be installed in boilers, or in which circumstances require the superheater to be near the engine, independent superheaters are recommended. In such cases superheaters heated by waste gases are the ideal installation.

The writer has designed and installed a number of superheaters both large and small for waste heat. Fig. 8 shows such a superheater for 40,000 lb. of steam per hour, installed at the plant of the Coal Mining Co., Gelsenkirchener Bergwerks Gesellschaft. The gases had an average temperature of about 1382 deg. fahr.

The moist steam enters the superheater at *A* in order to prevent rapid burning of the tubes at the point where the gases first come in contact with them. The steam flows through this portion of the superheater in the same direction as the gases, passes over to *B* and flows in the opposite direction, taking full advantage of the counterflow principle.

With a velocity of the flue gases of about 900 ft. per min., and a velocity of steam inside of the pipes of about 5000 ft. per min., the average heat transfer was 4.3 B.t.u. per sq. ft. per deg. fahr. temperature difference. The steam for this superheater was supplied by five waste-heat water-tube boilers, 300 h.p. each, four of which were always in operation.

On account of its smaller heating surface the cost of this independent superheater was considerably smaller than it would be with each boiler provided with its own superheater.

Most of the superheaters have been installed for waste heat from coke ovens and open-hearth steel furnaces. Also copper furnaces and cement kilns often furnished waste heat for superheaters.

The Author. Mr. Christie raises the question as to whether the principles of high gas velocity could not be successfully applied in direct-fired work. This has been tried a number of times, and the

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absence of unqualified success has come rather from improper boiler design than from faulty principle.

It is of interest to note that there have recently been sold a number of direct-fired boilers in which the principle of high gas velocity along the lines of modern waste-heat practice is followed. These units are not as yet in operation, but in view of the knowledge gained in waste-heat work there is no reason why the installation should not be wholly successful. That this view is warranted is best indicated by the boilers described in the section of the paper dealing with beehive-coke-oven gases. Here with entering gas temperatures of 2100 to 2300 deg. fahr., temperatures which at least closely approach the direct-fired practice at ratings of 200 per cent and upward, exit temperatures of about 475 deg. are secured.

The operating figures of the International Smelting Co. given by Dr. Ricketts are of interest. With one exception, all of the boilers furnished with copper reverberatory furnaces have been in accordance with early rather than modern waste-heat practice; that is, gas velocities comparable to those now used in open-hearth, cement-kiln and beehive-coke-oven work have not been used. In view of the success of the application of this principle in these industries, and from a comparison of exit-gas temperatures from boilers set with smelting furnaces and temperatures from the modern design of waste-heat boiler, it would certainly appear that a trial installation at least of the modern design is warranted on the part of one of the copper companies.

Mr. Lewis in his discussion gives some interesting figures on the performance of a boiler with low-temperature gases where, based on the transfer rate, the velocity must have been very low as compared to modern waste-heat-boiler velocity. He points out that for such low transfer rates the amount of surface to be furnished for a given capacity must be high. The boiler in question, presumably of 660 nominal rated horsepower, cooled approximately 48,000 lb. of gas per hour from 1050 to 418 deg. and developed 227 h.p., or some 34 per cent of its normal rating. It is interesting to compare the heating surface of a modern waste-heat boiler to develop the same capacity with that described by Mr. Lewis. With such a boiler it is entirely possible to cool this weight of gas from 1050 to 418 deg. and thus develop 227 h.p. (It is to be remembered that the temperature to which it is possible to cool a gas is to an extent governed by the pressure carried in the boiler.) The amount of heating surface necessary for such capacity, however, would be approximately 3500 sq. ft. as compared with 6600, and this heating surface would be operating at some 65 per cent of its normal rated capacity.

Mr. Lewis states that the power required to drive the fan was approximately 2½ per cent of the net capacity of the boiler. With the gas velocity through the boiler corresponding to the transfer rate obtained, the draft loss through the boiler proper must have been very low, and it is possible that the duty of the fan consisted largely in furnishing draft at the checkers. If the ordinary draft of 1.4 in. was required at the checkers, with the furnace directly connected to a natural-draft stack, the height necessary for a gas temperature of 1050 deg. would be approximately 150 ft. With the boiler installed and the gases cooled to 418 deg., this height of stack will give at the boiler outlet about 0.75 in., which would not be sufficient for proper furnace operation.

As compared with 2½ per cent of net power required for the fan with the boiler described, a motor-driven fan for the modern design of waste-heat boiler would require approximately 4.4 per cent of the gross output. A turbine-driven fan for the modern unit would require approximately 6.4 per cent of the gross output, but if the exhaust from the turbine could be used in a heater, a large part of this power could be returned to the system.

While, as Mr. Lewis states, the question involved is the cost of additional surface as compared with the cost of a high-pressure drop (for a net capacity), the foregoing comparison seems to be decidedly in favor of the modern design.

Mr. Broido's discussion, in which he refers to the utilizing of waste heat for superheating steam, brings out a point that was perhaps not sufficiently emphasized in the paper, namely, that by far the greater portion of modern waste-heat boilers installed have been equipped with integral superheaters. The high gas velocity has the same effect in increasing transfer rates in superheaters as in boilers, and the amount of superheat being obtained even with low-temperature gases is comparable to what would be secured from the same amount of superheating surface in direct-fired boilers.

Several installations have also been made of superheaters outside the boiler proper after the manner of separately fired superheaters. In one plant with which the writer is familiar such superheaters were set in the flue connecting open-hearth furnaces to waste-heat boilers. In these superheaters a transfer rate of approximately 6 B.t.u. was obtained.

In one or two other installations superheaters have been placed outside of the setting in the exit boiler flue. Such a location, however, would only be practicable with boilers where the gas velocities were such as would result in high exit-gas temperatures.

No. 1551

REPORT UPON EFFICIENCY TESTS OF A 30,000-KW. CROSS-COMPOUND STEAM TURBINE

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and

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Members of the Society

The determining factors in the selection of power-generating equipment, either for an original installation or replacement purposes, present about as many variables as there are installations. Engineering experience has practically standardized the electrical end. The steam engine, on the other hand, does not seem to have been capable of such standardization, as engineers are not agreed as to the most desirable superheat, pressure, etc., which must be considered both from the standpoint of reliability and boiler and engine economy. This thought is more than ever pertinent at this time, as it appears that we are on the eve of the employment of higher pressures and perhaps even new systems of power generation.

2 Further, the capacity of the machine to be selected and the relation of point of best steam consumption to maximum continuous load and provision for heavy overloads for limited periods are all dependent on load factors, diversity factors, etc., which present a new problem for every installation.

3 Without detailing at length the considerations which led to the selection of three 30,000-kw. units for the 74th Street Station of the Interborough Rapid Transit Company, a brief summation of the main questions involved will possibly be of interest and serve to classify the installation properly.

4 The prospective daily plant load, as was subsequently demonstrated, could be best provided for by units of 30,000-kw. maximum continuous capacity each. At that time there was assurance that

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machines of this capacity could be constructed, and furthermore, by the employment of the two-speed compound principle, as described more fully hereinafter, no hazards were involved, nor in fact would there be any departure from well-tried principles of construction. Ample steam supply was available by reason of development in the matter of relative boiler capacity coincident with the increase in efficiency of prime movers since the original construction of the plant in 1902. Economic considerations warranted the withdrawal of the horizontal-vertical double cross-compound engine-generator sets installed at that time, and their replacement by more efficient and larger units.

5 The generally accepted essential requirements of a railroad plant were the next considerations, *viz.*, reliability, efficiency and cost.

6 In the development of turbine design, at the time the installation was being considered, possibly the simplest type of machine in many ways was the single-shaft, single-rotating-element turbine, which was a natural outgrowth of the generally accepted type developed in smaller capacities. Certain structural features inherent with the larger capacity, however, tended to favor the division of the unit into two elements.

7 In a steam turbine, maximum centrifugal stresses are encountered at the exhaust end, where the greatest steam volume requires the greatest blade area. In the high-pressure blading, which uses steam of small specific volume, the best velocity ratio conducive to high economy cannot be met by the rotative speed as determined by the exhaust end. To avoid a compromise the conditions are more readily satisfied by carrying out the expansion in two separate elements and avoiding congestion, due to the high specific volume of the steam at 1 in. absolute exhaust pressure, by the use of the lower speed, longer blades and the double-flow principle.

8 Reliability also seemed better served in the double-element machine by the shorter shaft and reduction of danger from temperature strains. Furthermore, there was the possibility, in an emergency, of operating either element of the unit alone, and that at a fairly high efficiency, the low-pressure element being available for service simply by the use of a by-pass. The use of the high-pressure end alone in the case of a bad breakdown on the low-pressure end was to be obtained by removing the low-pressure rotor and closing the shaft openings with special covers.

9 An extremely important consideration, favoring the divided unit in this particular case, was the matter of the weight of the parts

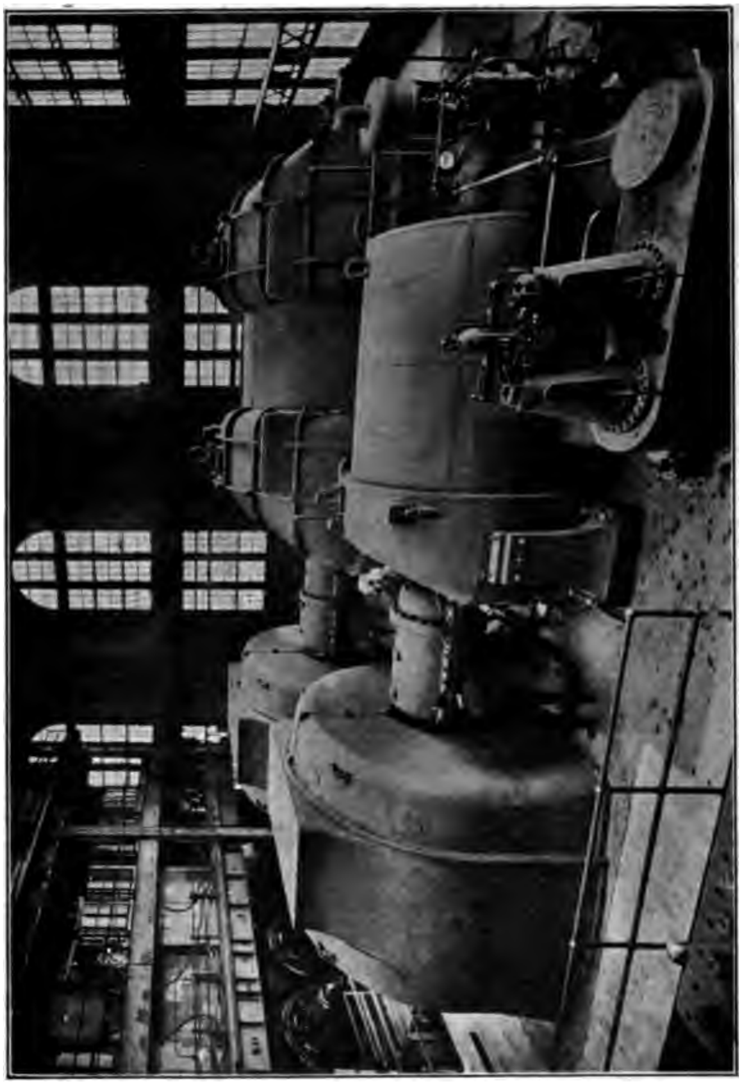

FIG. 1 30,000-KW. CROSS-COMPOUND STEAM TURBINE to be handled by the crane, which, in the case of the single unit, involved provision for additional crane capacity beyond that then available at the plant. To take care of such additional requirements would have necessitated the installation not only of a new crane but reinforcements of the steelwork of a most elaborate nature and far in excess of that required for use in connection with the double unit.

10 In the matter of efficiency the double unit offered still more desirable possibilities, prominent among which was the relative flatness of its water-rate characteristic. Without entering into the

FIG. 2 CASING, LOW-PRESSURE TURBINE

matter of cost in this particular case, it may be said that the advantage was again with the double unit.

11 As a result of such considerations as the foregoing, the double-unit turbines were selected, together with condensing equipment designed to perform at a relatively high rate of efficiency, and including the necessary auxiliary apparatus, all of which is described in the following paragraphs.

12 One of the complete units is shown in Fig. 1, the high-pressure element with its governing apparatus being in the foreground. This element is practically a typical single-cylinder reaction turbine, containing 38 rows of blades. The low-pressure element in the background of Fig. 1 is a turbine of the double-flow type. The casing is

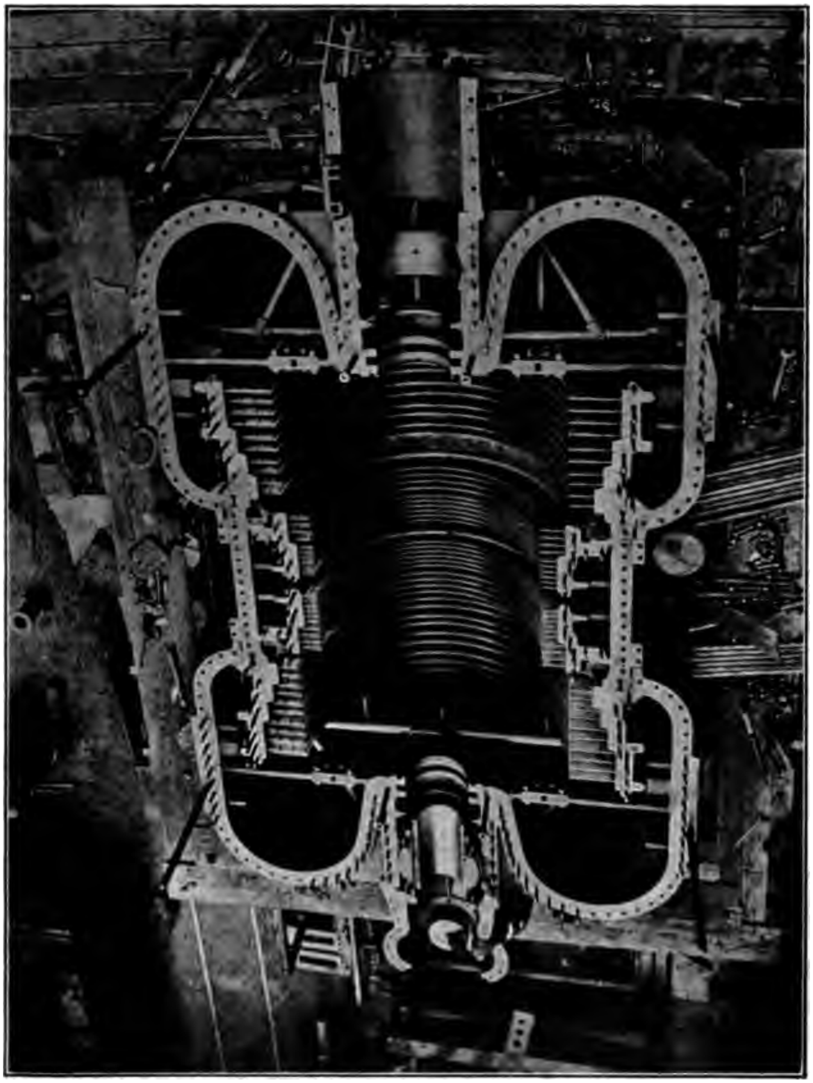

Fig. 3 Interior, Lower Half of Low-Pressure Turbine

possibly unusual in its size and details of reinforcement. The complete casing of a low-pressure element is shown in Fig. 2, and the interior of the lower half in Fig. 3. Fig. 4 shows a detail of the lower casing with the rings removed which gives a very clear idea of the reinforcing. No blading is mounted directly in the casing but through the intermediary of rings secured to it, as shown in Figs. 3 and 10. The central portion of the low-pressure rotor (Fig. 5) consists of a drum to which are bolted two cast-steel shaft ends. These shaft ends have disks mounted upon them, carrying the last rows of blades.

13 Each turbine was erected upon a foundation of steel frame-

Fig. 4 Detail Showing Bracing, Low-Pressure Casing

work set in concrete, the steel structure alone being amply sufficient for support, the concrete being added to eliminate vibration.

14 Inasmuch as the high-pressure rotor was inherently of small diameter, it was possible to design it to operate at the maximum speed possible for 25-cycle service, *i.e.*, 1500 r.p.m. The low-pressure turbine was designed for the next lower synchronous speed, namely, 750 r.p.m., utilizing a four-pole generator.

15 The surface condensers which occupy most of the open space beneath the foundation framework consist each of two shells containing 25,000 sq. ft. of condensing surface apiece. The condenser shells are connected directly to the turbine outlets without an intermediate

Fig. 5 Rotor, Low-Pressure Turbine

expansion joint. The weight of the condensers is carried by means of lugs, cast as a part of the shell, resting upon a number of spring jacks, which are adjusted to carry the load without appreciable strain upon the turbine exhaust nozzles. Practically no restraint has been put upon longitudinal expansion of the low-pressure turbine by the circulating water piping, which is fitted with rubber expansion joints. There is one expansion joint of copper in the steam-equalizing pipe between the shells.

16 The condenser auxiliaries were selected and installed with the prime factor of reliability continually in mind. The turbine-driven tri-rotor circulating pumps of 37,500 gal. per min. capacity each were installed in duplicate, the reliability consideration being augmented by one of efficiency in the matter of the relative requirements for winter and summer use; during the winter months with cold water one pump is sufficient, but during the summer months high water temperatures necessitate two units in service.

17 The hot-well pumps were also installed in duplicate. A single rotary dry-vacuum pump was provided for each unit, with sufficient capacity to handle the dry air from two units, cross-connections being provided for the purpose.

18 The turbines are equipped with water-sealed glands. The gland-water system has a small centrifugal turbine-driven pump for each unit, the piping for all of which is cross-connected through a common header. The water used for the turbine-oil coolers returns the heat which is thus regained by way of the feedwater heaters.

19 The exhaust from the auxiliaries has been carried into the feedwater heaters which operate at low pressure. The shortage or excess of heat in this heater system, as the case may be, is compensated for or utilized by means of heat-balance valves operating between the auxiliary exhaust lines and the receivers between the turbine elements.

20 For test purposes one unit was selected as representative of the installation. Tests were made upon this unit by the use of standard methods, special provisions being taken to secure accuracy, some of which are detailed below.

21 High steam pressures were observed by the use of gages, in duplicate where of importance, such gages being calibrated before and after each test. Temperatures were observed by carefully calibrated thermometers immersed in iron-pipe wells filled with mercury or oil, depending upon conditions. Wells were of ample depth and correction was made for immersion. Low pressures were observed

by the use of mercury manometers. Vacuum readings were made by the use of mercury columns provided with a vernier reading to $\frac{1}{100}$ of an inch. The mercury in these columns was regularly cleaned and its specific gravity determined before and after test series and corrections made accordingly. Temperature corrections on mercury columns were provided for by the use of thermometers set in each column casing, and additional corrections were made for specific gravity, meniscus and barometer reading. The barometer used on the test was calibrated by reference to the U. S. Weather Bureau.

22 The condensate was weighed in tanks mounted upon two carefully calibrated platform scales. Platform scales were also used to measure drips and leakage. The unit was isolated so far as water and steam outlets were concerned. The large atmospheric relief valve necessarily remaining connected to the exhaust system was water-sealed, the seal being kept under continual observation by means of a window in the valve cover and interior illumination by electric light.

23 The unit output was obtained by means of three single-phase rotating standard watthour meters, one connected to each phase. These watthour meters were calibrated before and after the test series, which calibration included the current and potential transformers and showed no variation in excess of 0.2 per cent.

24 Each test was of three hours' duration, with about a half-hour preliminary operation under test conditions. The load was controlled from the switchboard, the turbine operating upon the governor with hand throttle wide open. This subjected the turbine to the full swings of the railroad load, as observed by means of a graphic wattmeter, which furnished an interesting index of such load variations. Table 1 gives a summary of the results of these charts and Fig. 11 shows a typical section of a chart. At light loads these variations were more pronounced, approximating frequently 5000 kw. either side of the average. A variation of 10,000 kw. total in half a minute was not uncommon, especially on loads of 16,000, 18,000 and 20,000 kilowatts.

25 The test results have been shown graphically in Figs. 7, 8 and 9, upon bases of load and water rate, load and thermal efficiency, and load and Rankine-cycle efficiency ratio. In conducting these tests, naturally every effort was made to maintain certain standard conditions under which guarantees had been given, and the test results as tabulated have been corrected to such standards. These standards represent what is probably the average of operating service, and

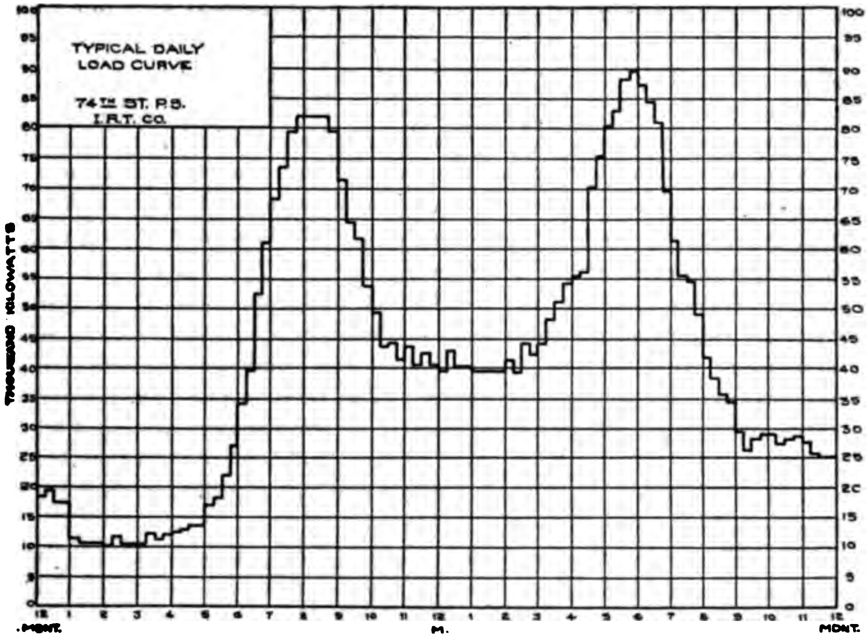

Fig. 6 Performance Curves, 30,000-Kw. Steam Turbine

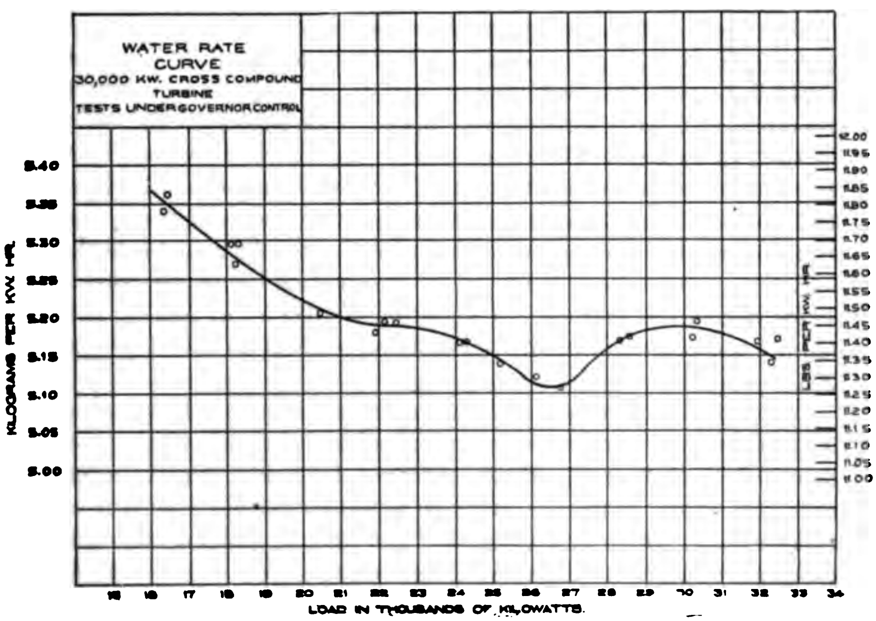

Fig. 7 Performance Curves, 30,000-Kw. Steam Turbine

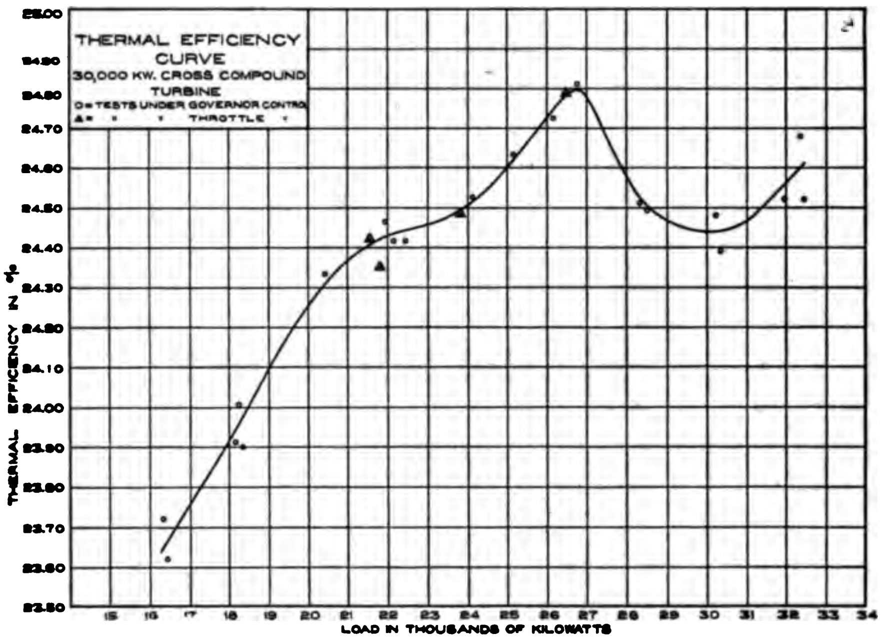

Fig. 8 Performance Curves, 30,000-Kw. Steam Turbine

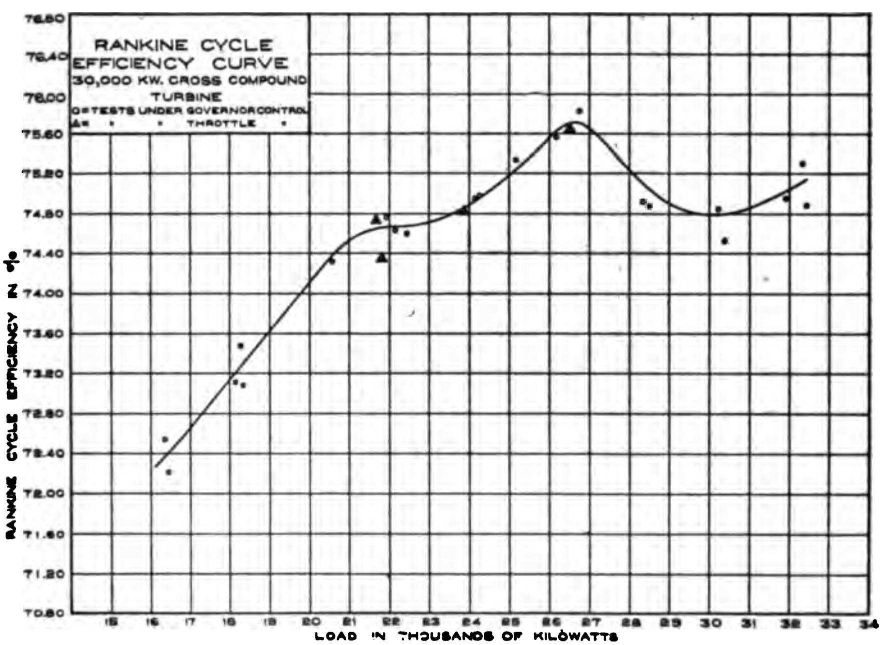

Fig. 9 Performance Curves, 30,000-Kw. Steam Turbine

TABLE 1 SWINGING LOADS DURING TESTS AS RECORDED BY GRAPHIC CHART

Test No.	Duration of Test, Hr.	Average	From Graphic Load Chart									
		Kw. load by watthour meter, A	Minimum load during test, B	Maximum load during test, C	Difference in kw. of max. and min. loads, $C - B = D$	Per cent of avg. load, $\frac{100\,D}{A}$	Avg. of diff. bet. min. and max. load during each 2½-min. period during test, E	Per cent of avg. load, $\frac{100\,E}{A}$	Max. single load swing during test, F	Per cent of avg. load, $\frac{100\,F}{A}$	Length of swing F in sec.	Time of test
1	3	16,342	10,100	20,800	10,700	65.5	6080	37.2	8,700	53.3	21	12.00- 3.00
2	3	16,447	9,800	20,300	10,500	63.8	6200	37.7	8,200	49.8	40	12.00- 3.00
3	3	18,158	12,800	23,000	10,200	56.2	5680	31.3	7,800	43.0	34	1.00- 4.00
4	3	18,273	10,400	23,100	12,700	69.5	5860	58.6	10,600	58.0	41	12.00- 3.00
5	3	18,317	11,600	22,000	10,400	56.8	6360	34.7	9,400	51.3	24	12.00- 3.00
6	3	20,472	15,600	22,600	7,000	34.2	3810	18.6	6,500	31.8	42	2.00- 5.00
7	3	21,925	14,900	26,100	11,200	51.1	3740	17.1	6,650	30.3	26	8.00-11.00
8	3	22,150	16,500	27,400	10,900	49.2	4200	19.0	6,500	29.4	24	8.00-11.00
9	3	22,432	16,900	25,400	8,500	37.9	4580	20.4	7,400	33.0	33	8.00-11.00
10	3	24,137	18,700	27,800	9,100	37.7	3700	15.3	6,900	28.6	36	8.00-11.00
11	3	24,200	20,500	27,600	7,100	29.3	3160	13.1	4,300	17.8	15	8.00-11.00
12	3	25,170	19,700	27,400	7,700	30.6	3260	13.0	5,500	21.9	12	10.30- 1.30
13	3	26,153	20,200	28,900	8,700	33.3	3080	11.8	5,400	20.6	12	8.00-11.00
15	3	28,378	21,000	32,700	11,700	41.2	3700	13.0	5,400	19.0	1	3.30- 6.30
16	3	28,517	23,200	29,700	6,500	22.8	2200	7.7	4,000	14.0	11	3.30- 6.30
17	3	30,232	24,400	32,000	7,600	25.1	3040	10.1	4,500	14.9	12	3.30- 6.30
18	3	30,397	24,600	32,800	8,200	27.0	2780	9.1	5,600	18.4	2	3.30- 6.30
19[1]	3	31,986	29,400	35,200	5,800	18.1	2450	7.7	3,700	11.6	9	4.00- 7.00
20	3	32,348	32,900	28,300	4,600	14.2	2490	7.7	3,100	9.6	5	4.00- 7.00

[1] Tests 19–22 incl. — no chart taken.

TABLE 2 SUMMARY OF RESULTS, TURBINE TESTS

Test No.	Abs. Steam Pressure at Throttle		Steam Temperature at Throttle		Superheat		Abs. Steam Pressure, Primary Inlet		Abs. Steam Pressure, Exhaust H.P.		Abs. Steam Pressure, Inlet L.P.		Exhaust Vacuum		Exh. Pres., Referred to 30-In. Bar, 762 Mm. 58.1 Deg. Fahr. 14.5 Deg. Cent.		Load Average	Actual Water per Hr.		Water[1] per Hr.		Water[1] Rate		Rankine[1] Cycle Efficiency	Thermal[1] Efficiency
	Lb. per sq. in.	Kg. per sq. cm.	Deg. fahr.	Deg. cent.	Deg. fahr.	Deg. cent.	Lb. per sq. in.	Kg. per sq. cm.	Lb. per sq. in.	Kg. per sq. cm.	Lb. per sq. in.	Kg. per sq. cm.	In. Hg.	Mm. Hg.	In. Hg.	Mm. Hg.	Kw.	Lb.	Kg.	Lb.	Kg.	Lb. per kw-hr.	Kg. per kw-hr.	Per cent.	Per cent.
1	224.3	15.78	493.2	256.2	101.6	56.5	148	10.41	10.6	0.745	10.2	0.717	28.411	721.63	1.589	40.37	16,342	206,606	93,716	192,350	87,250	11.770	5.339	72.540	23.72
2	223.3	15.7	486.8	252.7	95.8	53.2	144	10.12	11.8	0.83	10.0	0.703	28.52	724.41	1.48	37.59	16,447	207,422	94,087	194,431	88,194	11.822	5.362	72.222	23.62
3	224.1	15.77	485.7	252.1	94.2	52.3	166	11.68	....	....	11.0	0.773	28.539	724.9	1.461	37.1	18,158	225,668	102,363	212,029	96,176	11.677	5.297	73.118	23.91
4	223.3	15.7	489.6	254.2	98.4	54.7	159	11.18	11.8	0.83	11.5	0.809	28.471	723.15	1.529	38.85	18,273	226,659	102,813	212,330	96,313	11.620	5.271	73.476	24.02
5	222.6	15.65	496.5	258.05	105.7	58.7	163	11.46	11.8	0.83	10.8	0.759	28.588	726.15	1.412	35.85	18,317	224,564	101,862	213,976	97,059	11.682	5.299	73.086	23.90
6	220.4	15.5	510.9	266.1	121.0	67.3	175	12.31	12.7	0.893	12.0	0.844	28.586	726.08	1.414	35.92	20,472	243,365	110,390	234,930	106,564	11.476	5.206	74.400	24.33
7	219.8	15.46	519.3	270.7	129.5	72.0	182	12.8	12.8	0.9	12.5	0.879	28.683	728.55	1.317	33.45	21,925	255,233	115,774	250,407	113,585	11.421	5.181	74.757	24.46
8	221.8	15.6	518.8	270.5	128.0	71.1	185	13.01	13.5	0.949	12.8	0.9	28.791	731.3	1.209	30.7	22,150	256,059	116,148	253,388	114,937	11.439	5.189	74.630	24.41
9	221.1	15.54	501.5	260.8	111.2	61.8	187	13.15	....	....	13.0	0.914	28.744	730.1	1.256	31.9	22,432	263,943	119,725	256,716	116,446	11.444	5.191	74.606	24.41
10	221.3	15.56	513.9	267.7	123.5	68.6	194	13.64	14.2	0.998	14.1	0.992	28.902	734.1	1.098	27.9	24,137	276,139	125,257	274,971	124,727	11.392	5.167	74.947	24.52
11	220.3	15.49	512.9	267.2	123.0	68.4	196	13.78	13.7	0.963	13.6	0.956	28.882	733.6	1.118	28.4	24,200	277,377	125,818	275,593	125,009	11.388	5.166	74.973	24.52
12	220.3	15.49	496.8	258.2	106.8	59.4	197	13.85	14.6	1.027	14.1	0.992	28.896	734.0	1.104	28.0	25,170	290,332	131,695	285,184	129,359	11.330	5.139	75.357	24.63
13	217.9	15.32	524.3	273.5	135.2	75.2	211	14.84	14.5	1.019	14.7	1.033	28.761	730.53	1.239	31.47	26,153	297,644	135,011	295,373	133,981	11.294	5.123	75.597	24.72
14	224.0	15.76	500.0	260	108.5	60.3	215	15.12	....	....	15.0	1.055	28.862	733.1	1.138	28.9	26,740	306,718	139,127	301,035	136,549	11.258	5.107	75.840	24.81
15	221 6	15.58	510.0	265.5	119.5	66.4	210	14.77	16.5	1.16	15.9	1.118	28.862	733.1	1.138	28.9	28,378	326,655	148,171	323,353	146,673	11.395	5.169	74.928	24.51
16	220 0	15.47	515.5	268.6	125.6	69.8	211	14.84	16.6	1.167	16.2	1.138	28.812	731.82	1.188	31.18	28,517	328,279	148,907	325,072	147,453	11.399	5.171	74.901	24.49
17	220 3	15.49	513.0	267.2	123.0	68.4	210	14.77	16.5	1.16	16.8	1.181	28.78	731.01	1.22	30.99	30,232	349,775	158,658	344,787	156,395	11.405	5.173	74.862	24.48
18	217.2	15.28	518.0	270.0	129.1	71.8	206	14.48	17.3	1.216	17.2	1.208	28.801	731.55	1.199	30.40	30,397	351,009	159,218	348,018	157,861	11.449	5.193	74.574	24.39
19	217.1	15.27	515.4	268.6	126.6	70.3	212	14.9	18.5	1.301	18.5	1.301	28.623	727.03	1.377	34.97	31,986	373,532	169,434	364,280	165,237	11.389	5.166	74.968	24.52
20	218.2	15.33	519.5	270.8	130.4	72.5	209	14.7	17.6	1.237	18.2	1.28	28.75	730.25	1.25	31.75	32,348	370,825	168,206	366,572	166,277	11.332	5.140	75.343	24.68
21	216 3	15.21	513.3	267.4	124.9	69.4	209	14.7	18.1	1.272	18.9	1.328	28.765	730.63	1.235	31.37	32,490	375,734	170,433	370,335	167,984	11.398	5.1701	74.908	24.52
22	222.8	15.66	518.0	270.2	127.0	70.5	182	12.78	13.9	0.976	13.0	0.913	28.55	724.70	1.45	36.83	21,610	255,074	115,702	246,661	111,850	11.414	5.179	74.733	24.42
23	224.0	15.76	508.0	264.5	116.5	64.5	186	13.05	14.2	0.998	13.3	0.934	28.53	724.67	1.47	37.23	21,800	260,794	118,296	250,134	113,600	11.474	5.207	74.412	24.35
24	222.9	15.67	512.0	266.7	121.0	67.3	194	13.66	14.8	1.042	13.9	0.976	28.92	734.57	1.08	27.43	23,890	273,535	124,079	272,335	123,600	11.400	5.170	74.824	24.48
25	223.1	15.67	515.4	268.9	124.3	69.0	213	14.98	16.2	1.139	15.1	1.062	28.85	732.8	1.15	33.15	26,505	301,026	136,545	298,810	135,600	11.274	5.114	75.660	24.79

[1] As corrected to 215 lb. per sq. in. (15.1 kg. per sq. cm.) abs. press., 120 deg. fahr. (66.6 deg. cent.) superheat, 29 in. Hg. (736.6 mm.) vacuum.

fortunately it was possible to carry out the tests under conditions but little removed therefrom, making the corrections, therefore, a matter of small consideration. Corrections for variation from standard conditions were based upon curves as shown in Figs. 12, 13 and 14, which corrections in turn were based upon calculations, and the known results of previous practice were agreed upon prior to the commencement of tests.

FIG. 10 RING SECTION OF LOW-PRESSURE BLADES

26 In discussing and analyzing the test results, attention might be called to the following particular features:

First. With due allowance for scale of ordinates the performance curves may be considered unusually flat, naturally conducive to high plant efficiency.

Second. The dip in the curve between 22,000 kw. and 26,000 kw. is a peculiarity which was received at first rather skeptically, but which was later remarkably checked by repetition of tests throughout the range, including a special series under steady load made three months subsequent to the original series and given herein as Nos. 22–26 in the tables showing test summaries. Various theories have been advanced in connection with this dip, and a series of special tests was made to investigate the relative action of the receiver between the two cylinders as a separator and the velocities of the steam passing through it, with the idea that this might have some direct bearing upon the dip. Unfortunately, winter load demands termi-

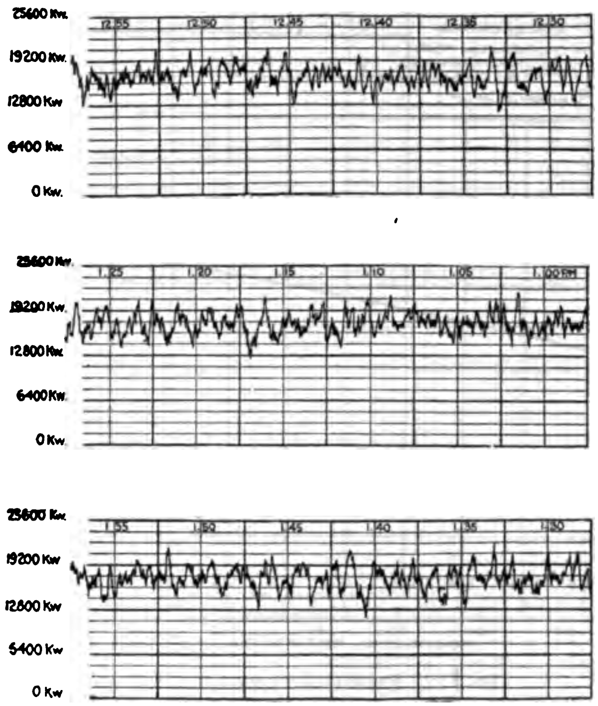

FIG. 11 WATT-METER CHARTS

nated further research work in this direction without any definite results having been obtained.

Third. The turning up of the curve between 30,000 and 32,000 kw. is another peculiarity, accounted for by the turbine designer as follows:

"Concerning the supposed inconsistency in these tests, *i.e.*, the turning up of the efficiency curve between 30,000 and 32,000 kw., consideration and figuring indicate that this is actually not an inconsistency, but a new experience. This turbine was designed for higher hydraulic efficiency than probably any machine heretofore built, thus approaching the crest of the efficiency curve. The overload capacity of the machine is small, or, in other words, the amount the turbine is by-passed when the secondary valve opens is small, and the velocity ratio, therefore, is very little lower when full steam pressure is applied to the secondary inlet than when such pressure is applied to the

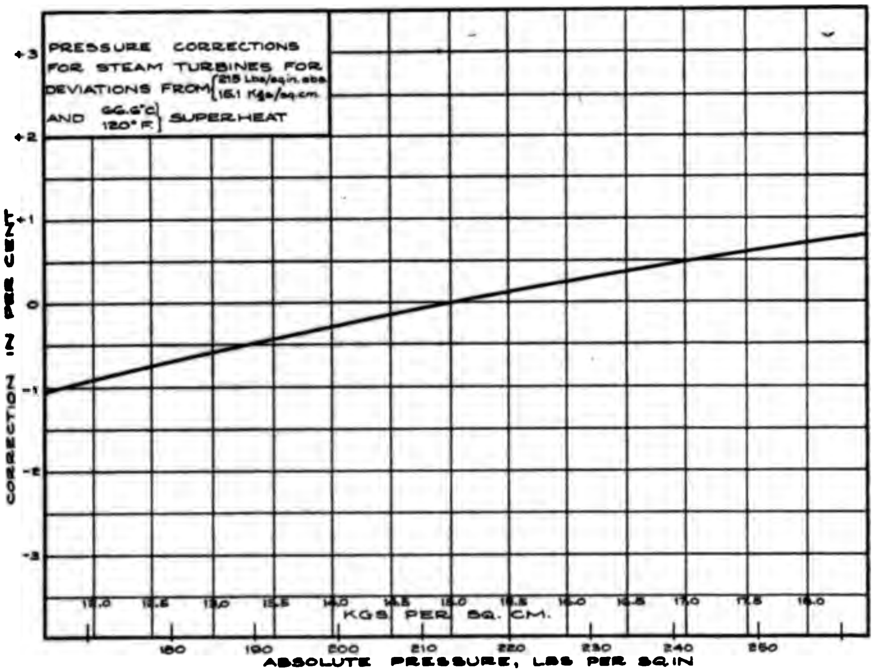

FIG. 12 PRESSURE CORRECTION CHART

primary inlet. Further, the hydraulic efficiency is nearly the same, so that the Rankine cycle at 32,000 kw. should not be more than 1 per cent lower than at the point of best efficiency, *viz.*, 26,000 kw. The efficiency at the intermediate overload, say, 30,000 kw., is somewhat worse than this, for while the blading and hydraulic efficiencies remain as high there is a loss due to a certain portion of the steam expanding through the secondary valve to a lower pressure without doing work."

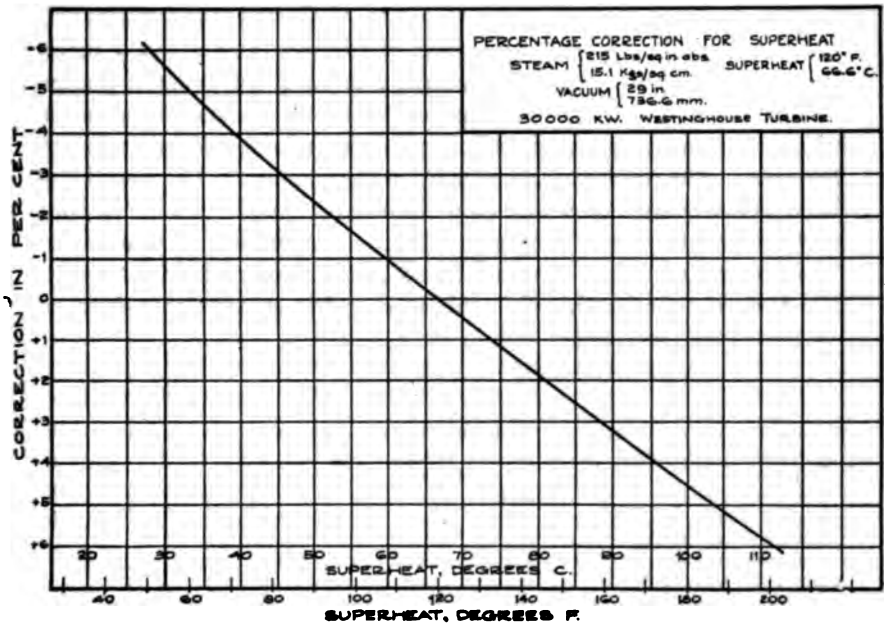

FIG. 13 SUPERHEAT CORRECTION CHART

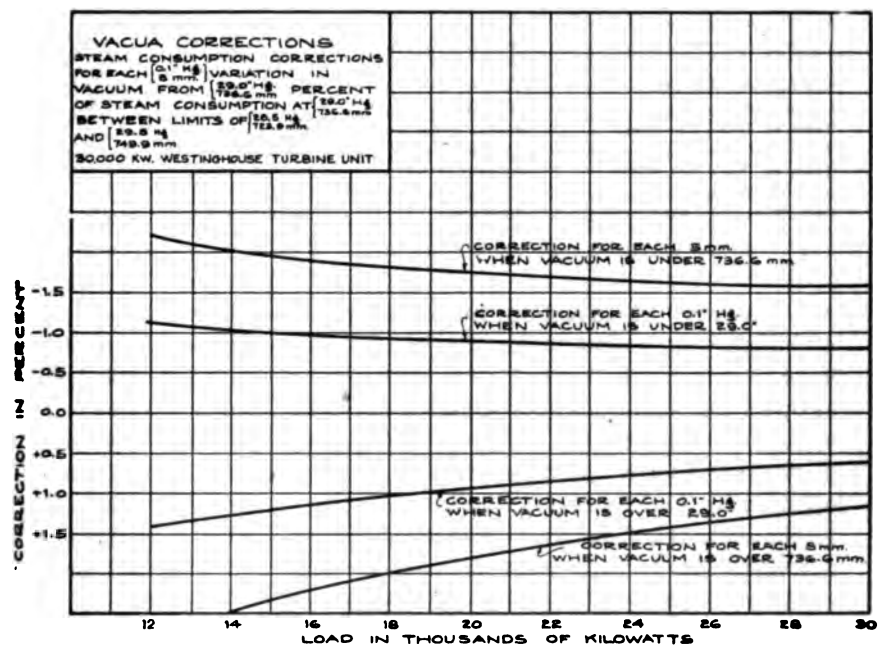

FIG. 14 VACUA CORRECTION CHART

27 A number of tests were made upon the condensers and auxiliaries, separate and individual guarantees having been made upon the condenser proper, circulating pumps, dry-vacuum pumps and hot-well pumps. Publication of the test data upon the pumps would simply show good, efficient performance without unusual features. The results of the tests upon the condenser, however, have been given simply as a matter of information and interest in connection with the installation as a whole. The guarantees were based upon operation under maximum load, *viz.*, about 32,000 kw., and the test was made with the turbine carrying as near that load as

TABLE 3 CONDENSER TEST, SUMMARY OF RESULTS

Pressure at Throttle, Abs.	220 Lb. per Sq. In.	15.466 Kg. per Sq. Cm.
Temperature at Throttle	487 Deg. Fahr.	252.8 Deg. Cent.
Superheat	97 Deg. Fahr.	54 Deg. Cent.
Load, Avg. Kw.	31,233	
Exhaust Vacuum	28.61 In. Hg.	726.69 Mm. Hg.
Exhaust Pressure, Abs.	1.39 In. Hg.	35.31 Mm. Hg.
Corresponding Temp.	89.4 Deg. Fahr.	31.9 Deg. Cent.
Mean Temp. Difference (Log.)	12.9 Deg. Fahr.	7.17 Deg. Cent.
Heat Transferred per Hour	316,000,000 B.t.u.	79,632,000 Large Calories
Heat Transferred per Unit Surface per Hour	6,330 B.t.u. per Sq. Ft.	17,150 Large Calories per Sq. Meter
Heat Transferred per Unit Surface per Hr. per Deg. M.T.D.	490 B.t.u. per Sq. Ft.	2390 Large Calories per Sq. Meter
Air Leakage per Min.	16.88 Cu. Ft.	478 Cu. Dm.
Condensate per Hr	357,060 Lb.	161,962 Kg.
Temp. of Hot-well Water	86 Deg. Fahr.	30 Deg. Cent.
Temp. of Intake Water	70.8 Deg. Fahr.	21.5 Deg. Cent.
Temp. of Discharge Water	80.9 Deg. Fahr.	27.2 Deg. Cent.
Rise in Temperature	10.1 Deg. Fahr.	5.7 Deg. Cent.
Circ. Water per Min.	64,700 Gal.	243,890 Liters

was possible. The duration of this test was three hours, and of the preliminary operation under test conditions, 30 min. The results have been shown in Table 3. No corrections were made in this test, as operating conditions approximated the guaranteed conditions very closely, with the exception that air leakage was high. The tests were made very shortly after the installation of the machine and little opportunity had been given under operating requirements properly to eliminate this leakage, which has since been done. A permanent gasometer has recently been installed in connection with each unit, in order to observe air leakage at regular intervals and aid in its elimination.

28 Summarizing the results of these tests, it may be said that the performance in the case of both turbine and condenser showed higher efficiencies than were guaranteed under contract, and the installation has proved to be thoroughly satisfactory in every particular, having fully realized the considerations which governed its selection.

DISCUSSION

Henry G. Stott, in opening the discussion, stated that the 30,000-kw. turbine unit described in the paper had been installed in the space formerly occupied by a 5000-kw. engine unit erected in 1900, and was served by the same group of eight 520-h.p. boilers. These latter, however, had been recently equipped with underfeed stokers. The cost per kilowatt of the modern unit, including condensers, was a little over $9, as compared with $40 in the first place. The actual water rate with the modern unit, discounting the superheat, was about 12 lb. per kw.-hr. on the average, as compared to about 17.5 lb. with the engine unit.

The steam turbine had now been developed to such a point that its thermal efficiency was as high as that of the best gas engines, and as gas-engine units cost seven or eight times as much as turbine units, they were practically out of the running as regarded large power-plant work. For that matter, so far as the future could be foreseen, the steam turbine would have no rival but water power, and then only in those cases where the load factor was large.

Francis Hodgkinson. In the report of these tests one should not be misled by the large-scale ordinates employed. The casual inspection of Figs. 7 and 9 would lead one to believe there are many inconsistencies. Both exhibit two rather curious reverse bends, one occurring between 22,000 and 26,000, the other between 27,000 and 32,000. The latter lends itself to the explanation quoted in the text of the paper; the former is not as easily explained.

It is unfortunate that Fig. 7 did not have total steam against kilowatts plotted thereon, because up to the point of best efficiency this should be a right line and hence any irregularities in the test results may be best studied by means of such a line.

Plotting this line through the averages of the test points it is found that the greatest deviation therefrom amounts to only 0.053 lb. in test No. 4. It was supposed that the behavior of the receiver separator installed between the high-pressure and low-pressure turbine elements was the reason.

It seemed in advance that there was an advantage in the cross-compounding scheme in that there was an opportunity to separate effectively the precipitated water between stages and relieve the

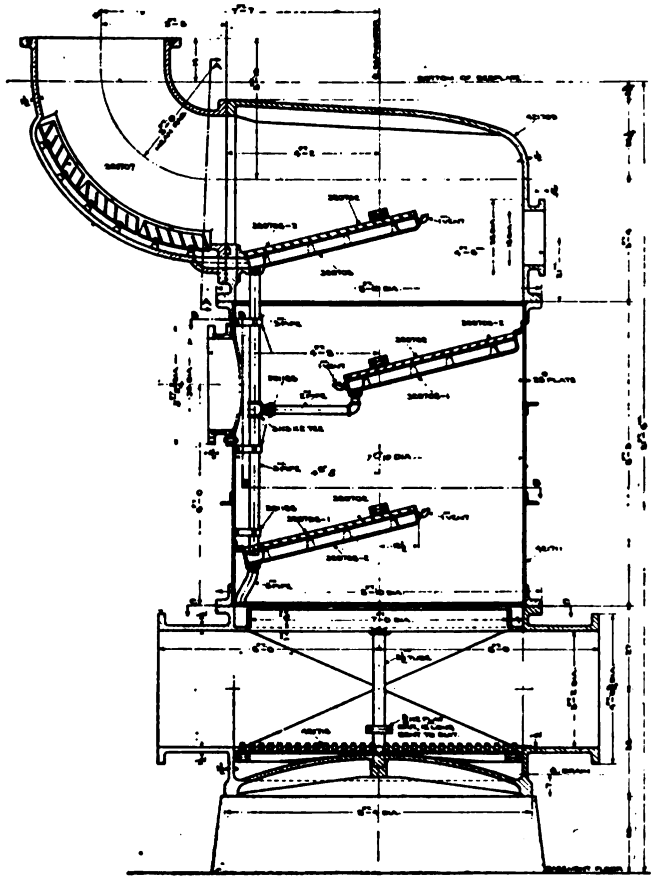

FIG. 15 STEAM SEPARATOR USED IN THE PLANT

low-pressure elements of the friction of this inert fluid, and so an attempt was made to provide a satisfactory steam separator between the two turbine elements. The design finally adopted is shown in Fig. 15, in which the moisture is abstracted by the use of baffles, and to some extent at least by centrifugal force as the steam flows around the curved surfaces at high velocity. The baffles are hollow boxes with perforations and riffles on their upper side, the interior being piped to the base of the separator to preclude the water once separated being picked up again by the flowing steam; the inlet elbow is similarly treated. The general arrangement is shown in Fig. 16.

Contemplation of the separator makes it readily conceivable that with small flow effective separation may be accomplished by virtue of the low velocity; that at large flow the separator may also be effective by reason of centrifugal forces, and that there may be some

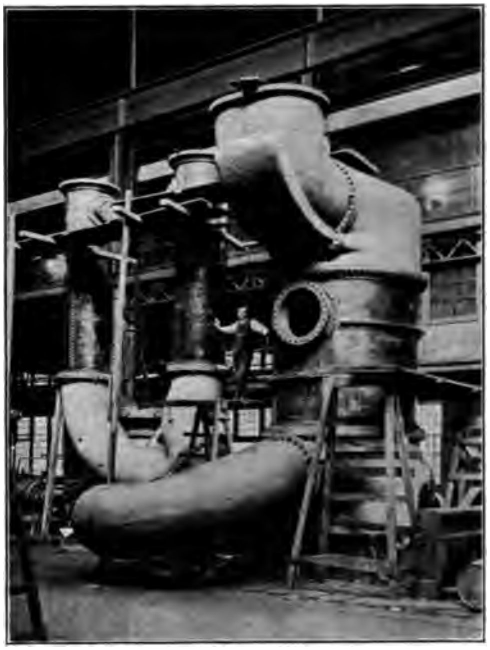

FIG. 16 GENERAL ARRANGEMENT OF SEPARATOR

intermediate and critical point where the water would sweep through without separating.

At the conclusion of the tests the builder's engineers set about an investigation of this separator with a view to eliminating the peculiarity in the test results and generally bettering the performance.

The observations made showed a considerable drop in pressure through the separator, and it seemed that an improvement in performance might be obtained by removing all baffles, even though this might result in some increased moisture delivered to the low-pressure turbine; the riffle plates at the base of the inlet elbow being replaced with a smooth perforated plate.

Fig. 17 has been plotted showing the pressure drop and moisture in per cent plotted against steam flow both with and without the baffles, and it is seen that the removal of baffles not only caused material reduction in the pressure drop through the separator but also a material improvement in separation.

Considering further the matter of the separator, I have had plotted curves, Fig. 18, showing the weight of water going to the low-pressure element; also the calculated quantity of water leaving the

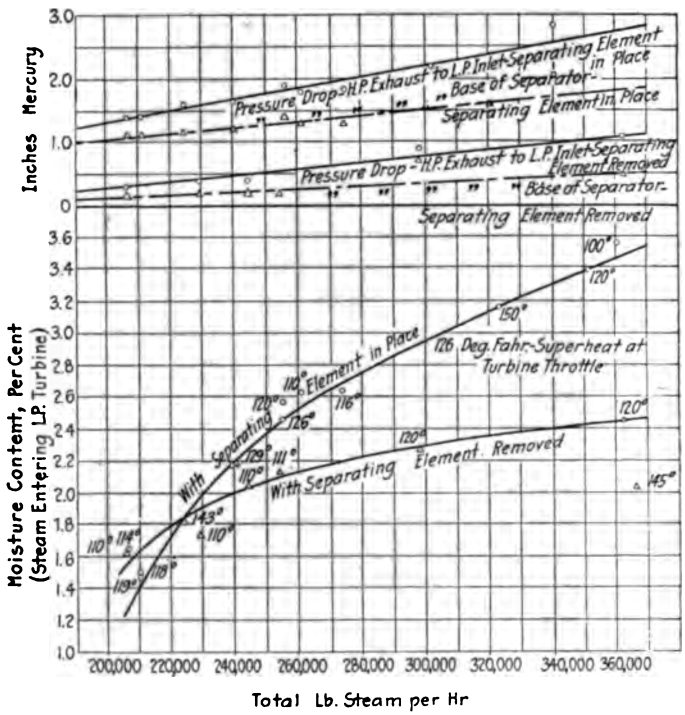

FIG. 17 PRESSURE DROP AND MOISTURE PLOTTED AGAINST STEAM FLOW

high-pressure. This is calculated on the assumption of all turbine elements of the complete unit being equally efficient — which is not quite technically correct — and from the actual steam consumption of the complete unit. The ratio of these two lines is plotted in Fig. 18 and shows the effectiveness or efficiency of the separator.

It is plain, therefore, that were the tests to be now repeated, one to one and a half per cent better results might be expected.

A clause in the contract for these turbines provided that the tests should be carried out with the load variations incidental to the elevated railroad, and indeed the load was anything but steady, as shown in Table 1, there being load variations recorded as high as 69 per cent and single swings as high as 58 per cent — all of which one would suppose would affect the test results quite deleteriously. For while the heat absorbed by the turbine walls during an upward swing is all given back to the steam in a downward swing, the heat is taken from the steam when it is at a higher pressure than when it

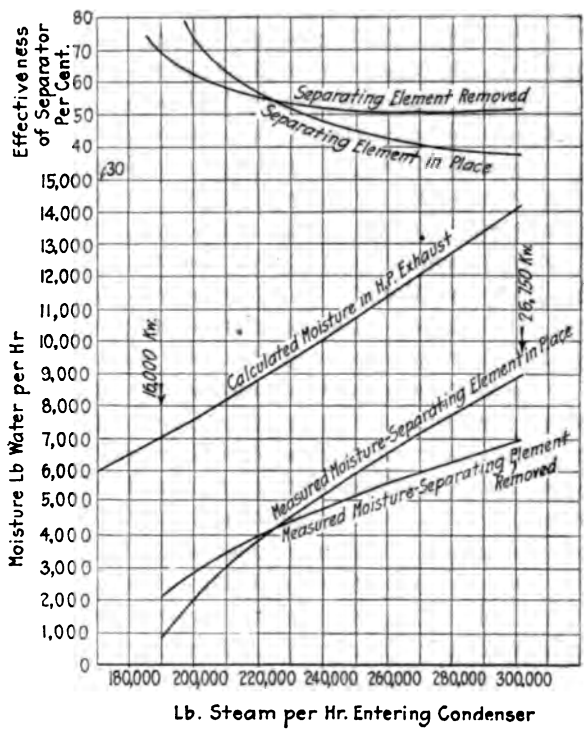

FIG. 18 WATER GOING TO LOW-PRESSURE ELEMENT

is returned to it, which is of course a direct loss. However, later tests with the governor hobbled showed no particular difference.

A point of interest in these tests is that between the loads of 21,000 kw. and 32,000 kw., *i.e.*, a range of 11,000 kw., the variation in steam consumption was from 11.274 to 11.449, a difference of but 0.17 lb.

R. J. S. PIGOTT. Referring to the use of two speeds for high- and low-pressure elements, as noted in the paper, the inherent difficulty with the reaction-type turbine is that the high-pressure blading is always noticeably less efficient than the low-pressure blading. The hydraulic efficiency of each is probably about the same, but the leakage area over the tips of the high-pressure blades is much greater than the leakage over the low-pressure blades because of the relative shortness of the former. The high speed possible with the high-pressure element makes it possible to reduce the diameter of the drum and at the same time maintain satisfactory blade speeds, and in this manner lengthen the blades for a given annular area; thus, of course, reducing the percentage of leakage area. The gain due to this feature alone is approximately 3 per cent of the total water rate of the unit.

Referring to the use of the individual elements separately in emergency, it is to be noted that in case the low-pressure element is used on by-pass, it must be protected against overspeed in the same manner as the whole unit; therefore the by-pass was fitted with a 12-in. automatic valve operated from the overspeed governor of the turbine in the same manner as the main throttle.

Referring to foundations, it is to be noted that in the case of turbine foundations of steel the question of strength is of little importance if the structure has been designed for the allowable deflection, since the foundation is then many times as strong as a normal load-supporting structure. The maximum stresses, disregarding the stiffening effect of the reinforced concrete, amounted to about 7000 lb. per sq. in. The increase of strength due to the concrete envelope is about 40 to 45 per cent additional.

The main reason for employing concrete coating was to destroy the excessive harmonic vibration occurring in plain steel structures. It has been found in those plants where uncovered steel foundations are in use (notably the 201st Street Station), that with a turbine equally well balanced, harmonic vibrations are very much more pronounced. The concrete envelope has also an additional function: of protecting the steel permanently against corrosion. The maximum deflection used in designing the steel structure was 0.020 in.

Referring to the condensers and their supports, the unusual feature of this installation is the large amount of movement between the two exhaust openings. In most installations at the present time a single condenser has been employed, but as the change of length between the centers of the exhaust openings amounts in this case to 0.150 in. between the operating condition and the non-condensing condition, it would be practically impossible to use a single condenser without expansion joints between the turbine outlets and the condenser. In the original design bronze single bellows were used on the piping, there being two 60-in., two 42-in., two 12-in. and a 36-in. It was found, however, that the comparative stiffness of these joints caused the turbine casing to be distorted enough to endanger operation. When it is remembered that one of the 60-in. joints is at a distance of approximately 27 ft. from the center line of the turbine, it is clear that a tremendous leverage is exerted. Some of the joints were tested, and it was found that one of the 42-in. joints took a 32,000-lb. load to compress ⅛ in. It was evident that no metallic joint with a reasonable thickness of shell would be satisfactory. Rubber joints were therefore substituted, using the rubber in much the same manner as the bronze had formerly been employed, the cast-iron parts being so designed as to give the rubber the maximum degree of support against either pressure or vacuum. As these rubber joints are only used in contact with water, there is no especial danger of oxidation and consequent short life.

Referring to the return of all waste heat possible, it is to be noted that the return of heat from the oil coolers amounts to nearly 1 per cent of the total heat in the turbine, and the heat in the air from the turbine generators — which amounts to about 1 per cent — is also returned by carrying a discharge duct to the boiler-room cellar, where the warm air is used by the forced-draft fans. The question of steam- or electric-driven auxiliaries was settled by a graphic method of heat analysis covering all loads; in this plant all auxiliaries are steam-driven. The total quantity of auxiliary steam increases very slightly with the load on the main unit; it is, therefore, evident that at light load there will be excess exhaust steam over that required for the feedwater heater, and with heavy load there will be too little, the balance occurring at approximately 25,000 kw. on the main unit. The heat-balance valve compensates for this feature; at periods of light load it will be noticed that the receiver pressure is less than atmospheric; at such times the excess exhaust steam is free to be admitted into the low-pressure element and there produces a certain amount of power. At high loads the pressure in the receiver is higher than atmospheric, and the conditions are therefore right to deliver steam to the auxiliary exhaust system as noted. Considering the nature of the above operations, the heat-balance valve designed by Mr. Hodgkinson is admirably simple. In the original layout no free atmospheric exhaust was to be provided; ordinary free exhaust being expensive, large-sized piping, used a few times during the life of the turbine. It was thought practicable to dispense with it by providing an additional automatic throttle valve controlled by a mercury tripper operated by the vacuum. This tripper was directly connected to the condenser and was in the form of a mercury gage; if the vacuum fell below a predetermined amount, say 19 to 20 in., a float in the mercury was lifted, tripping a pilot valve which closed the main throttle. It was also so arranged that in starting up the same method of safeguarding could be employed, operating at 1 to 2 lb. above atmospheric pressure, the device automatically resetting itself for the vacuum conditions, as the vacuum rises in the condenser.

However, the auxiliary exhaust system is of such dimensions that it can be used as an atmospheric relief, merely by enlarging a short section of the line from the turbine to the main exhaust. The old 30-in. engine relief valves were remodeled and the connection was made between the high- and low-pressure cylinders. The loading of these valves which, of course, have to stand pressure at the receiver of as much as 25 lb. absolute, is accomplished partly with a hydraulic cylinder connected to the gland sealing water. In addition, the dashpot on the other side of the valve was ring-packed and connected direct to the condenser, so that were the vacuum lost, part of the load would be automatically removed from the valve, assisting it to relieve the system promptly. As the discharge side of the valve was on the auxiliary exhaust system at 15.5 lb. to 16 lb. absolute, it was automatically steam-sealed; a water seal was therefore not absolutely necessary, but was installed as an additional precaution.

Referring to the variation of load during tests, as the Willans line up to best load is straight, and after best load is approximately straight at a different slope, the only point where swinging load would have any effect would be at, or near, the best load, since with a straight Willans line the average water rate for swinging loads is the same as the water rate at a definitely fixed load equal to the average.

Referring to the investigations on the separator, the conditions found were not surprising, considering that practically similar experiences were obtained in the matter of moisture separation from low-pressure steam in connection with the low-pressure turbine at 59th Street, in 1909 and 1910. The first separator installed on No. 1 unit at 59th Street was about 7 ft. by 14 ft., full of baffles; the results were, practically no separation of moisture. The baffles were removed and it was found that the quality immediately increased from 92 per cent to 95 or 96 per cent, using a plain tank.

Separators Nos. 4 and 5 were designed merely as plain tanks, 8 ft. in diameter and 32 ft. long, containing notbing but protection for the water in the bottom of the separator, and accomplished separation to 99 per cent quality and over. From the consideration of the sizes of these separators and the amount of steam passed, it appears evident that similar separation is quite out of the question for the 74th Street turbines. It seemed impossible with any baffle separator to remove anything but the water which is running along the sides or bottom of the pipe. The last 2 per cent of "fog" can be removed by only one method, *i.e.*, low velocity (not over 3000 ft. per min.) and space, allowing time for the small drops to coalesce into larger drops and rain out by gravity.

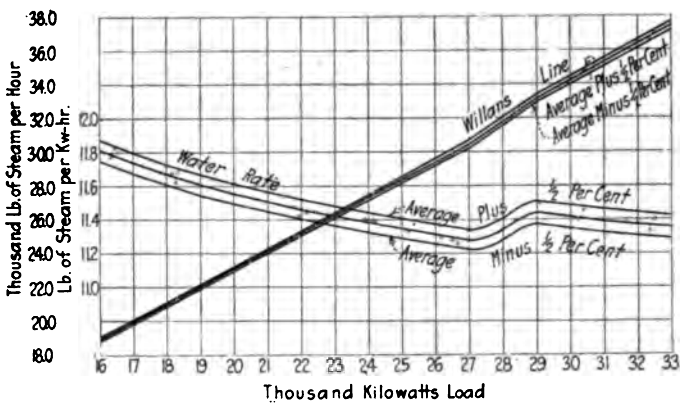

FIG. 19 WILLANS LINE AND WATER RATE FOR 30,000-KW. TURBINE
215 lb. absolute; 120 deg. fahr.; 29 in. vacuum; 30 in. barometer.

Referring to the curve of water rates, Fig. 7, as Mr. Hodgkinson has remarked, some of the curious features of the curve are due to exaggerated scale of the plotting. As he has also pointed out, it is very desirable to make use of the Willans line for fairing the curve. When it is noticed that the total water consumption at any given load may vary 3000 or 4000 lb. between different tests, it seems apparent that the unusual curvature represented, up to 26,000 kw., is not justified, but the water-rate curve drawn from the straight Willans line will pass just as fairly through the points and give results more in accord with ordinary expectations. Fig. 19 drawn with a straight Willans line in accordance with the above shows a maximum variation either side of the average of 0.5 per cent. All of the test points lie within the area of this curve, which is 1 per cent of the total water in width. No commercial tests up to the present time have ever come closer to the mean than this, and it seems unnecessary to look for new laws to fit these curves when the old ones fit very well.

Fig. 20 shows how closely the test results follow the general predictions of the guaranteed results. It is doubtful if there is such improvement in water rates in passing 31,000 as the test curve purports to show. Mr. Hodgkinson's point that the hydraulic efficiency of this machine is near the crest of the curve is perfectly true, but it must be remembered that when operating at maximum load on the secondary valve there are only 32 stages in operation instead of 38, the first eight being entirely blanketed at this time. Therefore, although the hydraulic efficiency of the stages may be as good as at best load, the efficiency could not be quite as good, especially as

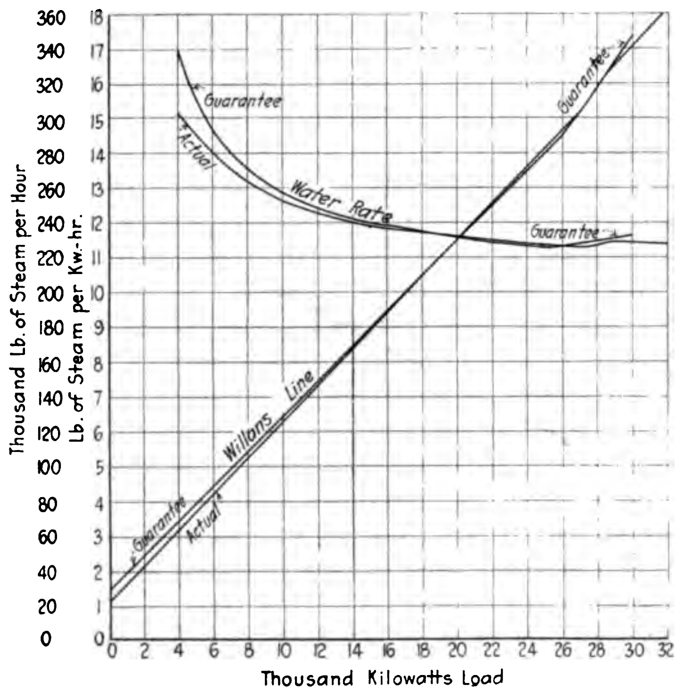

FIG. 20 WILLANS LINE AND WATER RATE FOR 30,000 KW. TURBINE. GUARANTEE AND TESTS

215 lb. absolute; 120 deg. fahr.; 29 in. vacuum; 30 in. barometer

the first eight stages are revolving idly in high-pressure steam, and the exit loss is increased. However, as has already been said, these variations from the curve as drawn from the Willans line are really very slight in actual amount.

FRANCIS HODGKINSON. Mr. Pigott refers to the advantage of examining turbine tests by means of plotting total steam against load. This desirability has already been referred to in my previous discussion, in which I also pointed out the small amount by which the test points departed from the Willans right-line law. The departure from this, however, was nevertheless there and not to be ignored. The tests were repeated too many times for the matter to be relegated to ordinary errors of test.

I do not concur with his conclusion as to the effect on steam consumption of the overload valves, that is, valves which permit the turbine to carry greater load by by-passing certain elements of the turbine. The hydraulic efficiency, and by this I mean the efficiency that is the result of the relation between steam and blade velocities and blade angles, with the angles ordinarily employed, is theoretically a maximum at about 90 per cent velocity ratio. It is readily conceivable that a turbine might be designed so that the velocity ratio with full steam pressure at the primary admission might be arranged to have a velocity ratio of considerably over 100 per cent. Then with full pressure applied to the secondary admission, the steam consumption might actually be better than with full pressure applied to the primary admission only, if the turbine is operating at somewhere near the same speed in each case. This is a feature which is frequently made use of in marine work in order to obtain high efficiency at cruising speeds. The above, however, does not refer to this specific case as the turbine under discussion is designed for a velocity ratio approximating 75 per cent. The amount of the turbine by-passed by means of the secondary admission is a small proportion of the whole turbine, so that the effect on the so-called hydraulic efficiency is small and hence the performance with full pressure on the primary admission only and full pressure on the secondary admission is not materially different, there being intermediary points of poorer performance.

It is customary nowadays for turbines to be sold at a maximum continuous rating corresponding to that of the generator. This, however, has very little to do with the capacity of the turbine itself, as the most important thing in the design of the turbine is the capacity pursue a tortuous course by the baffles, ample opportunity exists for similar superheating. Such superheating would naturally vary decidedly with the rate of steam flow and the entrance pressure. Simultaneous separation of moisture and superheating of steam would occur.

It is not improbable that this unlooked-for phenomenon was at least a partial cause of the unexpected hump in the economy curve between 22,000 kw. and 27,000 kw., as well as of the dip between 30,000 and the maximum load which the machine could carry, approximately 33,000 kw.

HENRY G. STOTT. Mr. Roberts asked what would happen if we added 25 per cent overload on the generator. Of course, obviously, the turbine would begin to shut down. Nothing else would happen except a reduction in speed of about 15 per cent.

In regard to the coal per kw-hr. on the engines before the turbines were put in at all, mentioned by Mr. Dean, we were running about 2.5 lb. of coal for the plant. That included all auxiliaries, coal-handling apparatus, and everything else. In other words, the total coal used per month divided by the kw-hr. delivered to the feeders leaving the power house (what we call the net output), would result in a figure of approximately 2.5 lb. of coal per kw-hr. Since the turbines were put in, and while they have been carrying practically all of the load too, with the steam mains leading to the engines still alive for emergency purposes, the coal consumption is approximately 1.5 lb., a drop of 1 lb. per kw-hr. as the result of the installation of the big turbines. The thermal efficiency of the station for the monthly average is approximately 17 per cent.

In reply to Mr. Foster regarding reheaters between the high- and low-pressure turbines, that practice has been abandoned, as previous investigations in connection with the engines showed no improvement, and there is probably none.

Mr. Reinicker asked why the vacuum went down with the load. That is undoubtedly due to air, which at that time we were unable to overcome, but later, with more care and experience and the use of a gasometer, we have been able to eliminate that trouble.

No. 1552

GRAPHIC METHODS OF ANALYSIS IN THE DESIGN AND OPERATION OF STEAM POWER PLANTS

By R. J. S. Pigott, Bridgeport, Conn.

Associate-Member

Graphic methods of analysis have been used in so many branches of engineering for the solution of problems involving several variables that their application to steam-power-plant design seems entirely obvious. Nevertheless, there have been published only a few isolated examples of the use and value of these methods.

APPLICATION

2 In the power plant, graphic methods of analysis have two distinct and important applications:

a In the design of new plants, based upon the anticipated load curves only

b In the operation of existing plants for determining the efficiency of operation from plant records.

In the design of new plants the graphic analysis is based upon the guaranteed water rates of the various pieces of apparatus proposed for the plant, and will show the effect upon the cost of the plant of varying the combinations, such as substituting electric-driven for steam-driven auxiliaries. By means of the graphic representation of these facts in composite curves, many problems, otherwise settled by professional judgment, can be solved exactly in figures. In the operation of the existing plant, graphic methods are the most flexible and exact for analyzing the actual running conditions.

3 For illustrating the principal features of the graphic method, concrete examples will best serve the purpose. This paper will consider the application of the graphic method of analysis to two distinct and different types of steam power plant.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

a The 90,000-kw. addition to the 74th Street Station of the Interborough Rapid Transit Company, New York City, where the entire output is electrical energy from three 30,000-kw. turbine-driven units, and

b The power plant of the Remington Arms and Ammunition Company, Bridgeport, Conn., where there is a variable output of electrical energy from five turbine-driven units of different capacities, and steam for industrial purposes and heating, this latter output fluctuating, naturally, with the seasons.

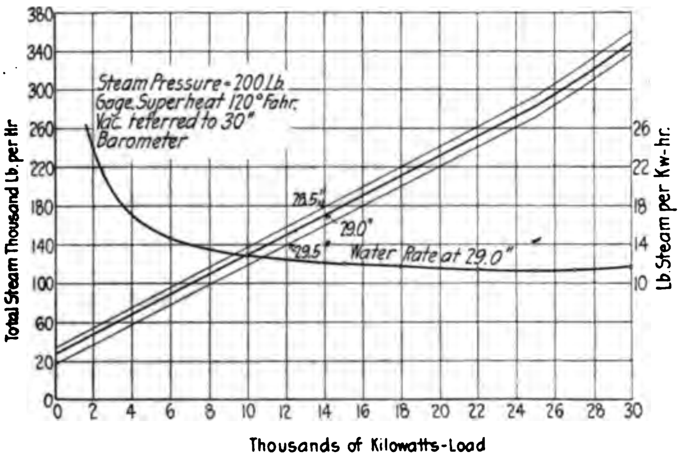

FIG. 1 WILLANS LINES AND WATER RATE, 30,000-KW. WESTINGHOUSE TURBINE UNIT

THE INTERBOROUGH ADDITION

4 *Individual Water Rates.* The Willans input-output lines have been employed considerably in connection with the performance of steam turbines and steam engines, but for a complete analysis it is necessary also to have input-output lines for the boilers and all auxiliaries as well. Whether steam- or electric-driven apparatus is used, these lines are equally important. The input-output lines for the following should be considered in a complete analysis:

a Main units
b Exciters
c Boiler-feed pumps
d Forced-draft fans

e Stoker drive
f Boilers.

5 *Main Units.* The Willans lines and water-rate curve for the 30,000-kw. Interborough steam turbines as guaranteed by the builder are shown in Fig. 1. Here it will be seen that the effect of variation in vacuum is a practically constant variation in the amount of total steam required over the entire range of operation. The percentage increase in economy, therefore, is greater at light than at heavy loads, a fact which should be considered in correcting input-output lines of the plant.

6 *Exciters.* In Fig. 2 will be found the Willans line and water-rate curve of a 300-kw. turbine-driven, geared exciter set. The

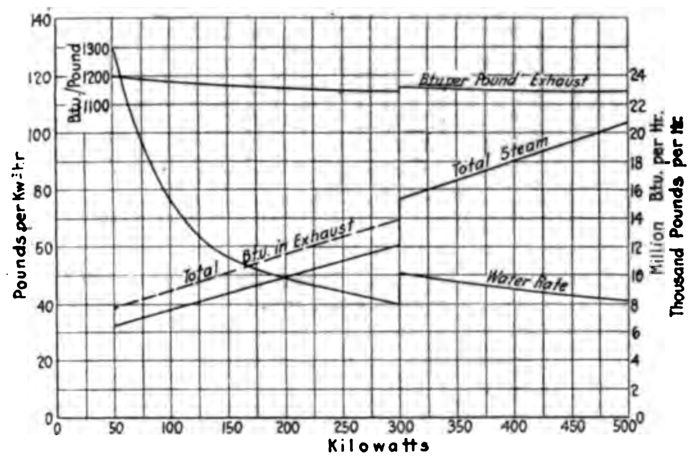

FIG. 2 INPUT-OUTPUT LINE AND ECONOMY, 300-KW. EXCITERS

machine is arranged with a special oversize generator and overload valve, so that, in the event of the failure of one of the two exciters to operate, the remaining one can carry the entire load for a convenient length of time. It is not intended to use the range from 300 to 500 kw. in regular service. In this same diagram are shown the total B.t.u. in the exhaust steam from the exciter, and the B.t.u. per pound of exhaust steam.

7 *Boiler-Feed Pumps.* Willans lines for a 1000-gallon-per-min. turbine-driven boiler-feed pump are given in Fig. 3. The total B.t.u. in the exhaust and the B.t.u. per pound of exhaust steam are also given. The high no-output steam consumption is characteristic of all constant-pressure, turbine-driven centrifugal pumps, in which the

efficiency drops off rapidly at the lower loads. Most boiler-feed centrifugal pumps show from 40 to 45 per cent full-load brake horsepower at no load, and from 50 to 60 per cent full-load steam consumption. The Willans lines for most turbine-driven centrifugal pumps show a greater curvature than those of Fig. 3.

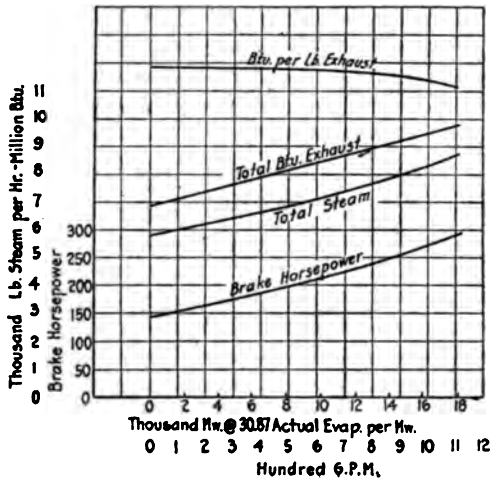

FIG. 3 INPUT-OUTPUT LINE AND BRAKE HORSEPOWER, 100 G.P.M. BOILER-FEED PUMP

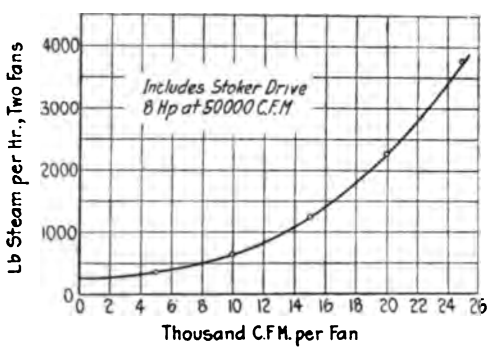

FIG. 4 STEAM CONSUMPTION AND AIR DELIVERED, FORCED-DRAFT BLOWERS

8 *Forced-Draft Fans.* The curve in Fig. 4 is the input-output curve of the forced-draft fans. The units are arranged so that two multivane fans are driven by a single turbine. One unit serves a

battery of two boilers through short air ducts. The sharp curvature is due to the fact that both pressure and volume of air increase with the speed. The curve is drawn on the basis of the steam consumed by both fans, and the air delivered by each. The mechanical stokers are driven from the fan shaft, requiring about 8 h.p. per pair at 50,000 cu. ft. per min. air delivery; this value is from tests.

9 The volume, static pressure and brake horsepower required by fans for the 520-myriawatt[1] B. and W. boiler using the Taylor stoker are given in the curves in Fig. 5, constructed on a basis of 17 lb. of air per pound of coal. The horsepower curve includes that

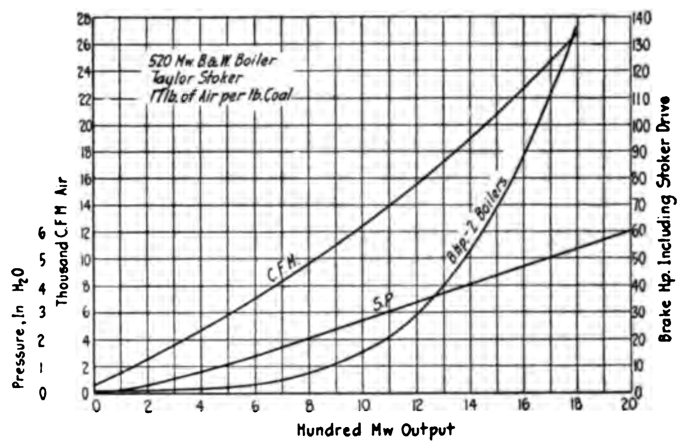

FIG. 5 AIR, BLAST PRESSURE AND BRAKE HORSEPOWER, FORCED-DRAFT FAN

necessary for the operation of the stokers and is given here because it is useful in considering electric drive of the fans. The results of numerous tests show that the CO_2 content and consequently the air per pound of fuel is substantially independent of the rate of driving with underfeed stokers.

10 From Figs. 4 and 5, Fig. 6 has been constructed, showing the relation between the steam consumed by the fans and the boiler output in myriawatts. The total B.t.u. and the B.t.u. per pound of exhaust steam are given.

[1] 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 B.t.u. per hour, the "boiler horsepower" being 33,479 B.t.u. per hour.

11 *Boilers.* In Fig. 7 the input-output curve of a single boiler is plotted from data obtained from actual tests. Heat units supplied are plotted against boiler output in myriawatts. The efficiency at

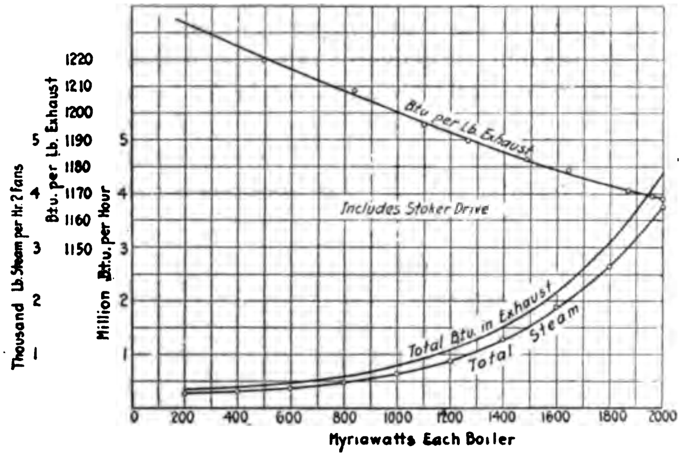

FIG. 6 STEAM CONSUMPTION AND BOILER OUTPUT, FORCED-DRAFT BLOWERS

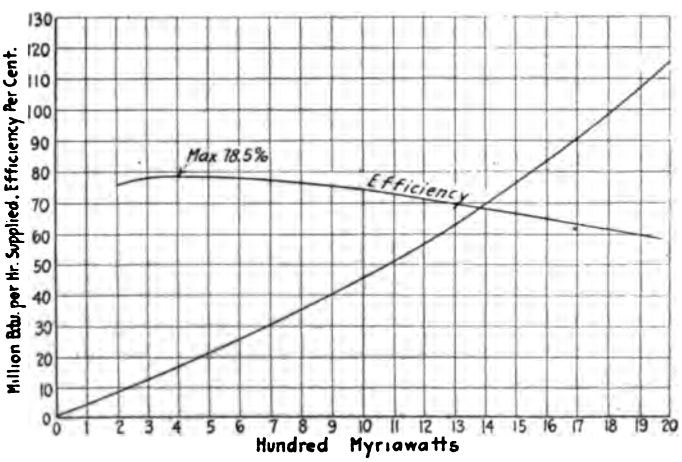

FIG. 7 BOILER INPUT-OUTPUT CURVE AND EFFICIENCY, FROM TESTS

different rates of output is also shown, the maximum being 78.5 per cent; higher efficiencies are frequently obtained in single tests, but the values given represent the average of the best results obtainable in practice.

12 *Condensers.* The Willans lines for the condenser auxiliaries are not essential, as the condenser operates at practically full load, once it is started. The variation in cooling-water temperature is shown in Fig. 8. The average for the year, June 30, 1912, to July 1, 1913, at the 74th Street Station of the Interborough Rapid Transit Company, was 53.9 deg. fahr. The winter conditions only are worked out in the curves, as they illustrate the method fully, and the summer condition is figured in exactly the same way, but with different vacuum and feed-water heat absorption, resulting in higher B.t.u. per kw-hr.

13 *Construction of the Combined Curves.* The method of obtaining the combined curves in Figs. 9, 10, 11, and 13 is quite simple. Fig.

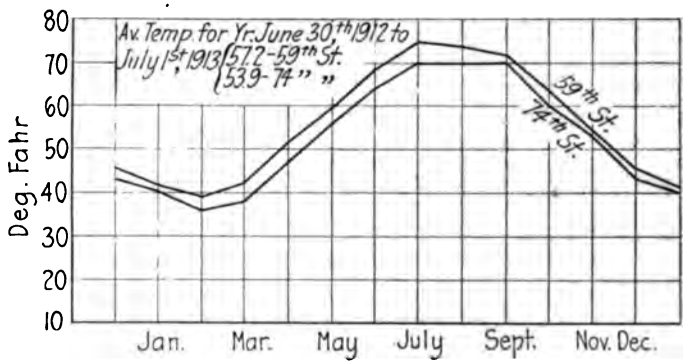

FIG. 8 CIRCULATING-WATER INJECTION TEMPERATURE

9 shows the heat input per hour plotted against myriawatt output of 30 active boilers, 520 myriawatts rated capacity each. By plotting Fig. 9 to a large scale, it is found that the theoretically most economical method is to reduce to zero load with all boilers active, banking none. It is not feasible to operate in this manner, however, because of the difficulties of operating the underfeed stoker at loads less than 80 per cent of capacity and the question of labor costs. Also, opportunity is needed for making minor repairs without cutting a boiler out of commission. Fig. 9 is therefore drawn on a basis of cutting out and banking as the load drops from 12,000 myriawatts at a rate of one boiler for every 400-myriawatt drop. This method is only slightly less efficient than continuous operation of all boilers.

14 *Combined Boiler-Room Auxiliaries, Input-Output Lines.* In Fig. 10 are combined the curves of Figs. 3 and 6 to give the total

steam consumed by boiler-feed pumps, fans and stoker drives per gross output of the boilers in myriawatts, and the total B.t.u. available in the exhaust steam from these auxiliaries. The curves show four sets of auxiliaries cut in at 14, 32 and 48 thousand myriawatts. It will be noticed that, for normal loads, the steam consumption of the boiler-room auxiliaries varies but little from 4.6 per cent of the total steam generated. The exhaust heat contents can be figured quite exactly, because all except about 3 per cent of the losses in a steam turbine

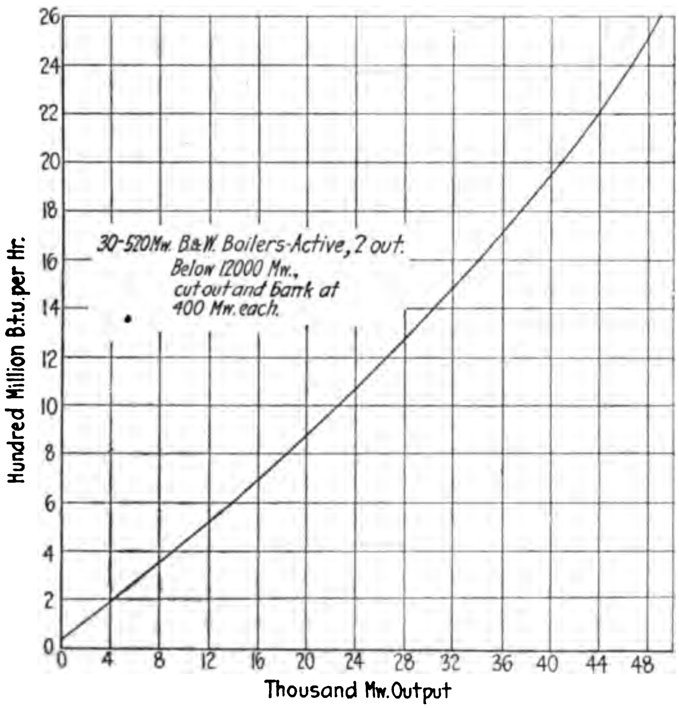

FIG. 9 BOILER-ROOM INPUT-OUTPUT LINE

reappear as heat in the exhaust. Therefore, *Heat in exhaust = Total heat in steam supplied* — (3 *to* 4.5 *per cent*) — *Brake horsepower* × 2547 — 3,200,000 *B.t.u.* The 3,200,000 B.t.u. covers radiation and leakage from the exhaust system. This radiation loss is figured, if possible, from area of pipe-line surface; but for quick figures 2 per cent of the heat of steam passing at full load will amply cover this loss, both on high-pressure and exhaust steam. Usually it does not actually exceed 1 to 1.5 per cent.

15 *Total Steam Demand.* The total steam demand for the main turbine units in pounds per kilowatt-hour developed is shown in one of the curves of Fig. 11, marked "main units." All curves on this sheet are based upon 29 in. of vacuum, and the circulating water is assumed to be at a constant temperature from May 1 to November 1, and at another temperature for the rest of the year. The gain in vacuum due to lowering the water temperature is slightly less than the loss due to an equal rise, but at this stage in the development

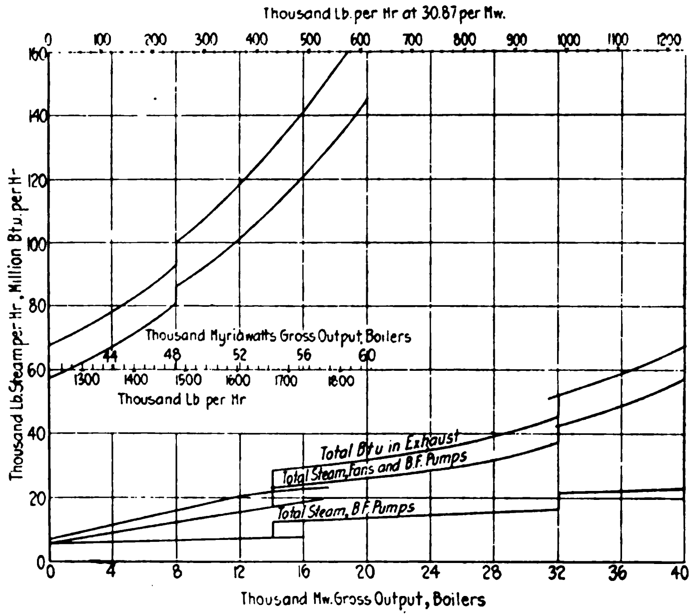

FIG. 10 BOILER-ROOM-AUXILIARY INPUT-OUTPUT LINES

of power-plant design it is idle to pursue such a nicety of calculation. In Fig. 11, the curve marked "main units" is obtained directly from Fig. 1. To this is added 21,620 lb. of steam per hour per unit for the condenser auxiliaries, which comprise two 37,500-gallons-per-min. circulating pumps, two 800-gallons-per-min. hot-well pumps, and one dry-vacuum pump for each unit. The resulting curve is marked "engine-room auxiliaries." Adding the boiler-room auxiliary steam demand, as shown in Fig. 10 the third curve, marked "all auxiliaries," is obtained. There are also added 3 per cent for starting, warming up, and losses dependent upon load, and 25,000 lb.

of steam per hour for pipe radiation, boiler dusting and leakage. These last are substantially independent of the load. From Figs. 10 and 11, the heat demand for the engine room and plant is drawn in Fig. 13, heat units of input being plotted against kilowatts output.

16 As pointed out by H. W. Flashman, in 1911, it is noticeable that the economical point for cutting in units is not at the point of maximum steam economy of the unit under operation, but always at

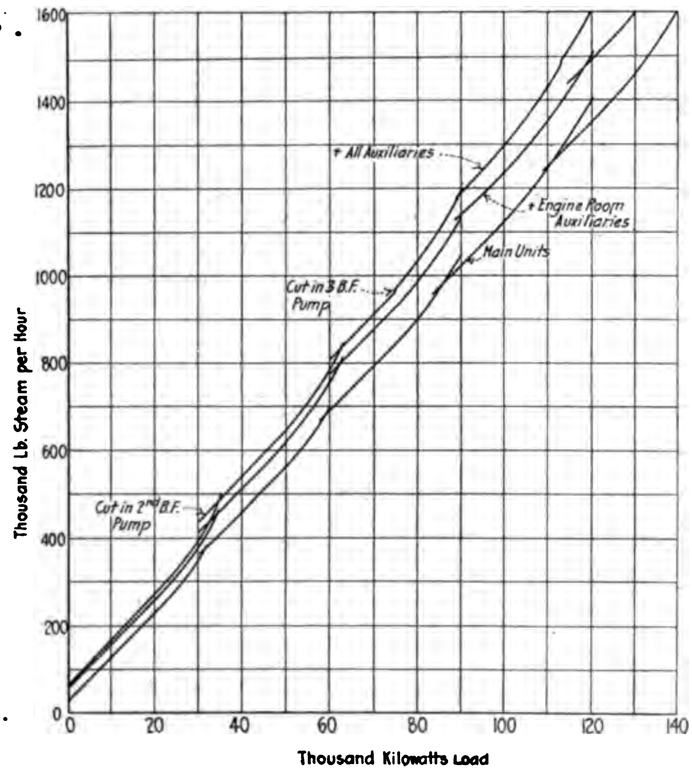

FIG. 11 ENGINE-ROOM STEAM DEMAND

a higher point. This is most marked in the curve showing the total steam consumption of the main units and all auxiliaries. Here it will be seen that the steam required starts on a more rapid rate of increase after 30,000 kw., yet if a new unit is cut in, approximately 8000 lb. of steam per hour additional will be required at once. From the curves, it is apparently inadvisable to cut in the new unit until the load has reached about 35,000 kw., when the total steam demand for the single unit has reached the total steam demand for the two units. Of course the deciding factor is the maximum capacity of the generator, and the character of the load. But this feature proves that main units should seldom be designed for the best water rate at maximum load, *i.e.*, straight Willans line to full maximum rating. This is brought out plainly again in the Remington plant Willans line, Fig. 23.

17 *Bleeding*. The curves at the bottom of Fig. 13 show the heat units required in the feedwater heater and the heat units available from the exhaust from auxiliaries per kilowatt of output. It will be noted that there is more steam available during the lighter loads on the units in service than is required for feedwater heating, and that at the heavier loads the auxiliaries do not supply enough exhaust steam. This is to be expected, because the largest single item of auxiliary exhaust supply is that from the condenser circulating pumps and is constant. The obvious remedy for this unbalanced condition of supply and demand is heat balancing by means of suitable valves, so that steam may be bled from the main units when more heat is required in the feedwater heater, and returned to the main units from the auxiliary exhaust when there is available more exhaust steam than is demanded by the feedwater heater.

18 The conditions are naturally suited to this arrangement. When the amount of exhaust steam from the auxiliaries is excessive, at low loads, the receiver pressure between high and low sections of the turbine is lower than atmospheric pressure, and when the amount of steam is insufficient, at high loads, this pressure is higher than atmospheric pressure. The amount of auxiliary steam required is purposely adjusted to be correct for the load at which the receiver pressure just equals the auxiliary exhaust pressure. This is accomplished by using more or less economical drives for the auxiliaries, as may be needed. In other words, a highly economical auxiliary turbine is not necessary in most cases.

19 *Effect of Bleeding*. The effects of introducing or removing steam from a compound turbine are shown in Fig. 12. Here are shown the Willans lines of the high- and low-pressure turbines and entire unit, with and without bleeding. The plotting of individual water rates is laborious and unnecessary.

20 It will be noticed that the straight Willans line for the low-pressure turbine is represented by the equation

$$W = 45{,}000 + 20.26\,L$$

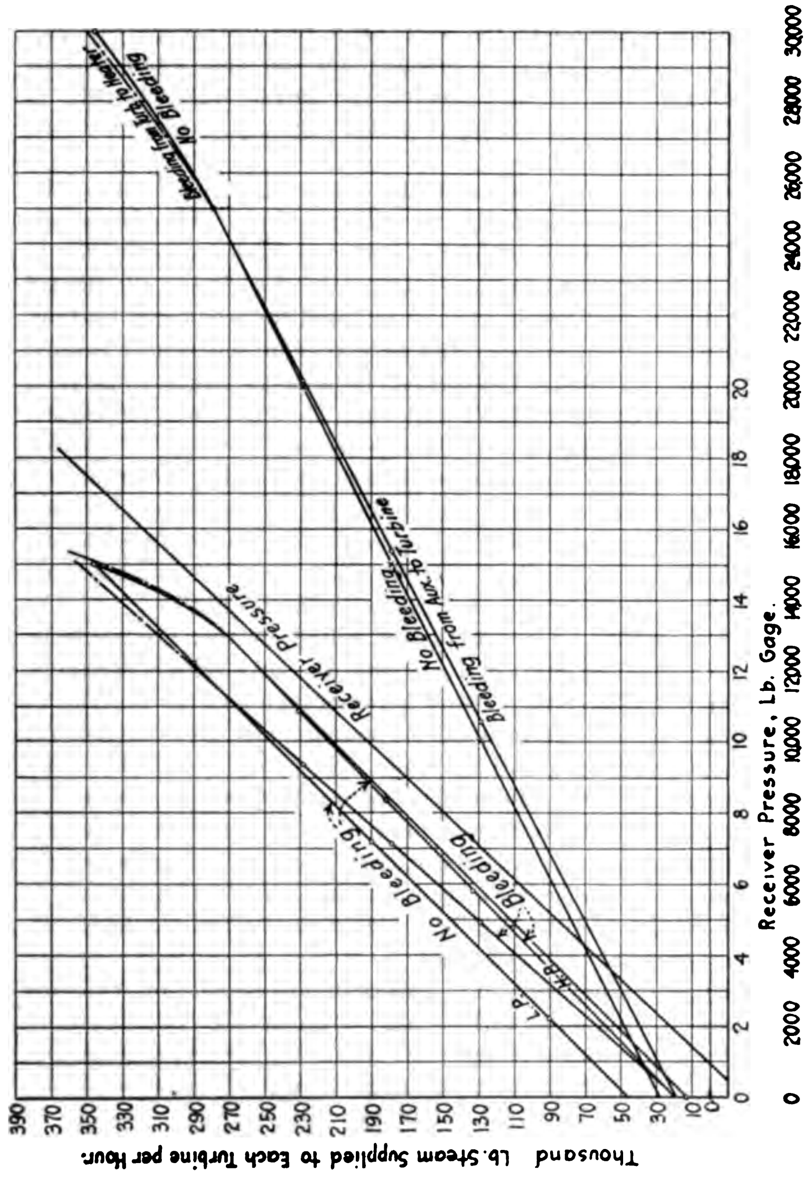

FIG. 12 CORRECTED WILLANS LINES FOR HEAT BALANCING

in which W = total pounds steam consumed
L = kilowatt output of low-pressure turbine only
45,000 = no-load steam consumption, pounds.

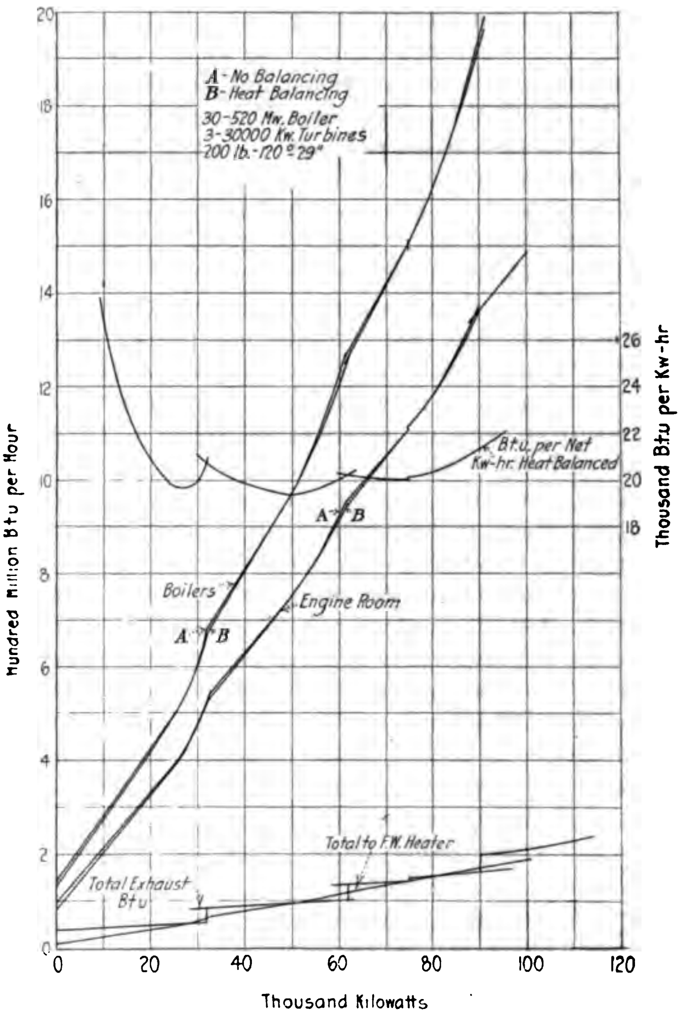

Fig. 13 Plant Input-Output Lines, Steam Auxiliaries

Therefore, the addition of steam in any reasonable quantity at any load will produce the same kilowatt increase in output of the low-pressure turbine, at a water rate of 20.26 lb. of steam per kilowatt-hour. The decrease of output of the high-pressure turbine, due to increased back pressure, is practically in the ratio of the adiabatic heat drop.

21 For example, from the curves at the bottom of Fig. 13, at 60,000 kw. load, the third unit having been cut in, the heat which is available for the low-pressure turbine due to the excess from the auxiliary exhaust over that necessary for the feedwater heater amounts to 20,600,000 B.t.u. per hour. This will give the low-pressure turbine 17,900 lb. of steam per hour above that coming from the high-pressure turbine, and this, at the previously determined water rate, 20.26 lb. of steam per hour, will increase the low-pressure-turbine output by 882 kilowatts. From Fig. 12, the increase in receiver pressure is from 11.75 to 12.65 lb. per sq. in. absolute. This amounts to a loss of 5 B.t.u. in the high-pressure turbine, or about 2.52 per cent of the available heat drop. At the load carried by the high-pressure turbine, 10,800 kw., this loss in available energy amounts to 272 kw. The net gain by introducing the exhaust steam into the low-pressure turbine, without increasing the steam admitted to the high-pressure turbine, is therefore 610 kw.

22 At 90,000 kw., with three units running, the heat required in the feedwater system, to be supplied by bleeding from the low-pressure turbine, is 6,400,000 B.t.u., representing 5560 lb. of steam. The reduction in output from the low-pressure turbine, at the water rate of 20.26 lb. of steam per kw-hr., is 279 kilowatts. The drop in receiver pressure is from 17.25 to 17.00 lb. per sq. in absolute, resulting in a gain of 75 kw. in the high-pressure turbine. The net loss in output is therefore 204 kilowatts. The heat thus deflected from the low-pressure turbine is utilized in the feedwater heater, instead of being partially used in the low-pressure turbine to generate 204 kw. and mainly rejected to the condenser.

23 *Balanced Input-Output Curve.* From the anticipated load curve, Fig. 14, and from Figs. 9 and 13, the B.t.u. consumption for the two conditions is obtained. The data are: 40,739 kw. average load, 71,900 maximum hours, 977,735 kw-hr. per day total output. From Fig. 13, without heat balancing, the total input, at 40,739 kw. output, is 19,823,300,000 B.t.u. per day, while with heat balancing, this amount is 19,636,800,000 B.t.u. per day. With coal at \$0.10 per million B.t.u. and such labor as is affected, which brings the cost up to \$0.11, the saving due to heat balancing is \$204.00 per day.

24 *Effect of Changing Operating Conditions.* The Interborough installation considered in this paper is capable of peak loads of 90,000

kw. because there is adequate reserve in reciprocating-engine units. It will be seen that the B.t.u. per kilowatt-hour would be slightly higher with a 90,000-kw. peak load and about the same load factor than with the original 70,000 peak, but as is pointed out and proved below, there must be a balance set between economy and fixed charges.

25 The Interborough station is now running 22,500 B.t.u. per net kilowatt-hour, or 91 per cent of what is possible with the equipment. The effect of vacuum improvement can be obtained readily.

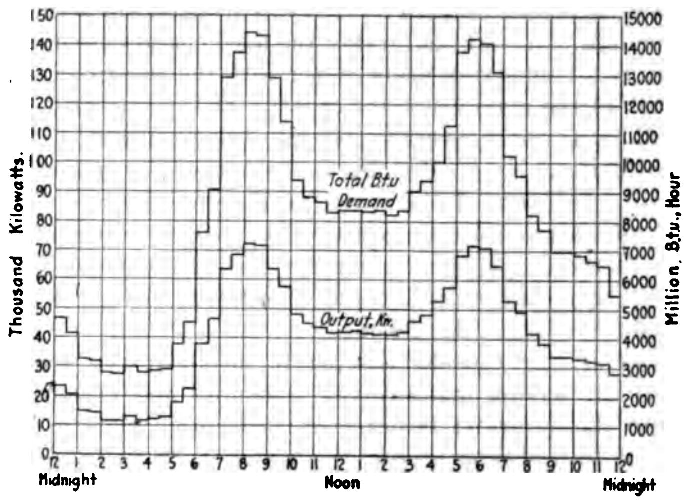

FIG. 14 ANTICIPATED LOAD CURVE AND HEAT INPUT BALANCED

In Fig. 1, as previously noted, the Willans lines at different vacua are practically parallel, showing that an improvement in vacuum results in a constant reduction in total steam in pounds per hour. The effect on the figures obtained above in Par. 24 is found by subtracting the number of pounds of steam, or heat units, saved per day. On the plant and engine-room input-output lines of Fig. 13, it suffices to shift the heat- and steam-consumption scales up or down as required.

26 The correction for superheat is a percentage of the total steam or heat consumption, since it affects main units and auxiliaries alike. It can therefore be applied as a direct percentage correction to the total figures obtained from the load curve. It is unnecessary to redraw any of the curves to make correction for alteration of superheat

or vacuum, because the corrections can be applied so easily to the total figures.

27 *Electric-Driven Auxiliaries.* The choice between steam and electric-driven auxiliaries can be definitely settled for any given set of conditions. The engine-room steam consumption will be that of the main units only, with the same allowances for losses varying with load and radiation, but the capacity of the main units is reduced by the amount used by the auxiliaries, the effect being to raise the input-output line. The greatest difference will appear in the boiler room, as there is no recovery of heat in the feedwater heater.

28 Fig. 15 shows conclusively that this arrangement is thermally less economical. In addition to this, the fixed charges are increased, the prices of all auxiliaries being higher with the electric drive, and the portion of the main generator capacity and engine-room equipment used for auxiliary power supply also being charged. The saving in auxiliary piping is offset by the cost of wiring and switching equipment. The cost of the steam auxiliary station per effective installed kilowatt is $65.00, against $69.10 for the electric auxiliary station. As the simple electric auxiliary station is defeated both on grounds of operating economy and investment charge, it can be eliminated from further discussion.

29 Let us now consider a separate turbine-driven generating equipment for serving motor-driven auxiliaries and exhausting its steam into the feedwater heater. An economizer will also be considered. With feedwater leaving the heater at 208 to 210 deg. fahr. and an average flue temperature of 450 deg. fahr. or less, the economizer is of doubtful value. Cold water cannot be used in the economizer because of the difficulty encountered with sweating, which causes clogging and renders the economizer inoperative. This is borne out by the experiences with the original 74th Street Interborough station, in which all of the auxiliaries, except the exciters, were electric-driven, and economizers were employed.

30 A short analysis will show the comparative value of this arrangement. Assume a load of 50,000 kw., which will give about the best economy for both steam and electric drive. The data are: main steam units, 561,200 lb. of steam, auxiliary power requirements, 1989 kw., 52,140 lb. of steam, including induced-draft fans and economizer scraper drive. In the steam-electric units the temperature of the feedwater would be raised from 70 to 155 deg. fahr. by the exhaust from the auxiliary turbines, and the possible rise in the economizer would be 147 deg. fahr. (limited by the fact that flue-gas temperature

cannot be brought lower than 150 deg. fahr. above water temperature). The use of the economizer will lower the B.t.u. per kilowatt-hour to 18,000, as against 19,250 without it, or about 6.5 per cent.

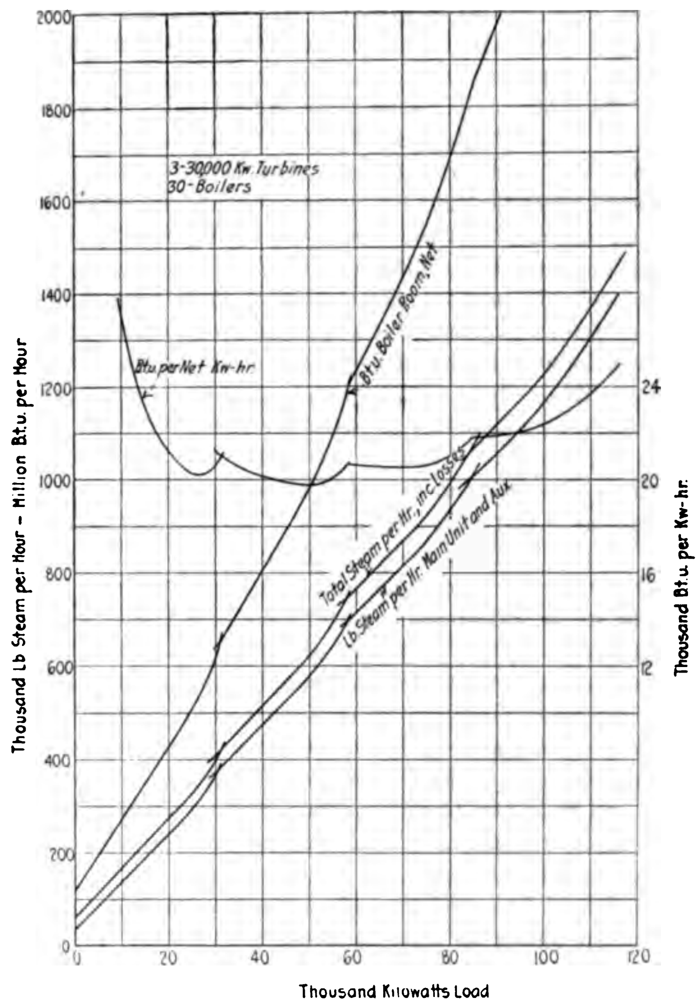

FIG. 15 PLANT INPUT-OUTPUT LINES, ALL-ELECTRIC AUXILIARIES

Economizers could be used with the all-steam units, but on account of the feedwater temperature, 208 deg. fahr., the saving would be only 4.1 per cent, making the B.t.u. per kilowatt-hour for this type of plant 18,540. The steam-electric plant, therefore, shows an improvement of 2.9 per cent. This, of course, is only at one load; the steam-electric combination does not necessarily operate more efficiently at all loads. The investment cost per kilowatt, however, is increased from $65.00 to $70.31. Whether or not it would pay to use the steam-electric combination, viewing it from the point of view of cost of power only, would depend upon the load factor. However, there are other items to be considered. There is an increase in the costs of labor and maintenance due to the extra turbines, generators, and economizers. The electric-driven auxiliaries are no easier to handle than the turbine-driven auxiliaries, and flexibility is lost to a certain extent, especially in dry-vacuum pumps and circulating pumps. Reliability, certainly, is not as good with the electric-driven units, as two additional links have been added to the auxiliary system. In the writer's opinion, electric-driven auxiliaries should never be used for boiler feed, and only with steam support for excitation. While dry-vacuum pumps and circulating pumps are less affected, they are better off with turbine drive from the operator's point of view.

31 Since the adoption of higher boiler pressures, the use of the economizer has grown more undesirable. It is not in the first year or two that trouble is likely to be encountered, but as the economizer ages, and rust, both inside and out, has weakened the tubes. At 225 lb. per sq. in. boiler-feed pressure there was trouble with tube breakage, and with the pressures of 300 and 350 lb. per sq. in. needed for the higher pressures now employed the reliability is very poor. If the economizer is to be employed at all, it should be of the steel-tube form, integral with the boiler, as in European practice. The method recently suggested, of operating the economizer at lower pressures and then boosting to boiler pressure, holds some promise. But it is to be remembered that this scheme requires double the number of pumps and piping.

32 It should be noted that the same results could be obtained with all-steam plants as with steam-electric, and with none of the disadvantages of the electric plant except the economizer. Another method is to use part steam and part electric auxiliaries, as is done in many plants, but the limits of this paper will not permit the analysis of all possible combinations. The above reasons show why the slightly more economical combination steam-electric auxiliary was not considered for the Interborough plant; reliability was paramount.

33 *Number of Boilers to Install.* The best number of boilers and stokers to install for a given load curve can be determined readily. The cost per myriawatt-hour to operate a boiler and stoker is given in Fig. 16. The curves are based on the following data: Coal at

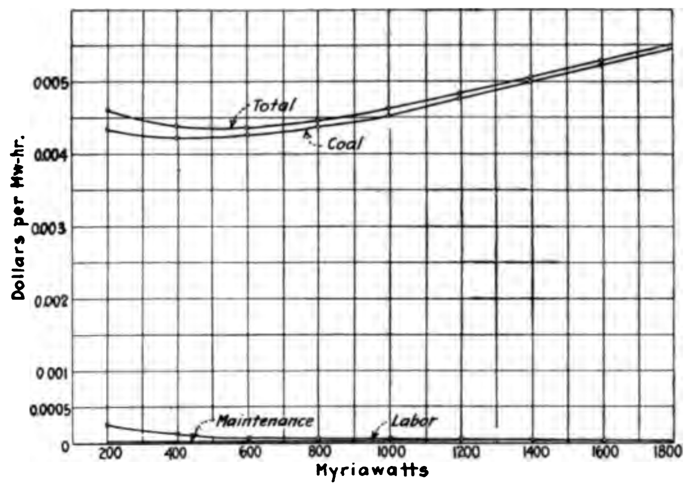

FIG. 16 COST OF OPERATION PER MYRIAWATT-HOUR, 520-MYRIAWATT BOILER

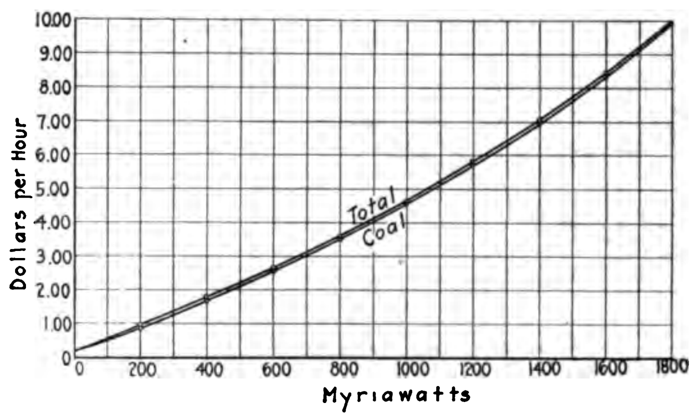

FIG. 17 TOTAL OPERATING COST PER HOUR, 520-MYRIAWATT BOILER

$0.103 per million B.t.u. ($3.30 per ton, heat value 14,250 B.t.u. per lb. of dry coal); maintenance at $0.03 per ton of coal fired; labor, one operator at $0.40 per hour and one helper at $0.20 per hour for every twelve stokers. Fig. 17 shows the total operating cost per hour per myriawatt load, and the total cost of coal. The total cost plainly follows the heat input-output line. Figs. 18, 19 and 20 show the boiler-room cost lines in dollars per myriawatt-hour for different numbers of boilers in operation.

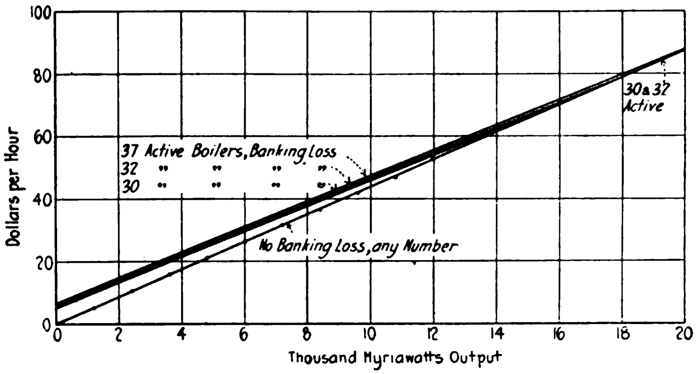

FIG. 18 COST PER HOUR, BOILER PLANT

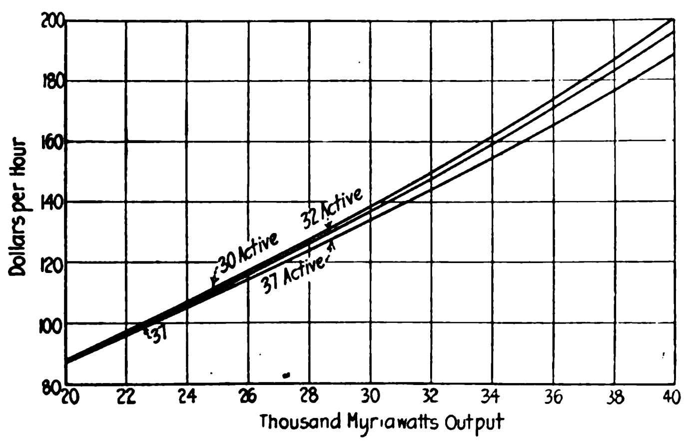

FIG. 19 COST PER HOUR, BOILER PLANT

34 Using the total heat demand as before, but separately for different numbers of boilers, Fig. 21 is obtained. This shows 30 boilers more economical at light loads and 37 more economical at heavy loads, and for a typical day shows a net saving of $59.20 by the larger number of boilers. In a new plant, unquestionably the greater number of boilers would not pay, as the fixed charges more than offset

the operating saving. The actual case in question required stokers and equipment, but not boilers, as these were already installed. The

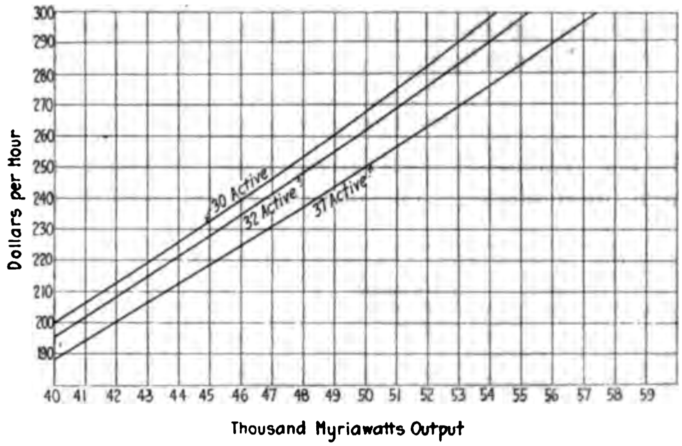

FIG. 20 COST PER HOUR, BOILER PLANT

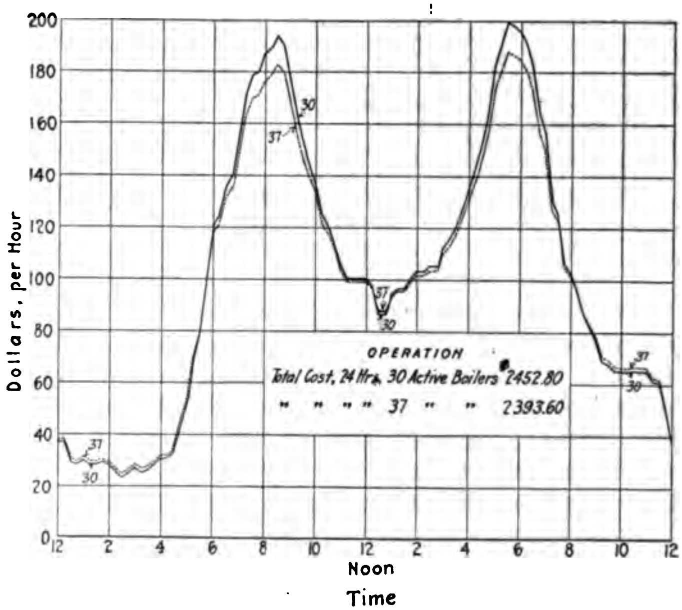

FIG. 21 HOURLY VARIATION OF BOILER-ROOM OPERATING COST, ANTICIPATED WINTER LOAD

comparison was that of 37 boilers in commission, 40 installed, and 30 in commission, 32 installed. The saving would not pay the fixed charges on the greater number of boilers even in this case. It is evident that the load factor is the controlling item: one must know the shape of the load curve as well as the peak loads and daily output. For a high-load-factor plant, the larger number of stokers. forced less, will generally pay.

REMINGTON POWER PLANT

35 The Remington plant offered some interesting studies for graphical interpretation. Low-pressure steam, at 15 lb. per sq. in.

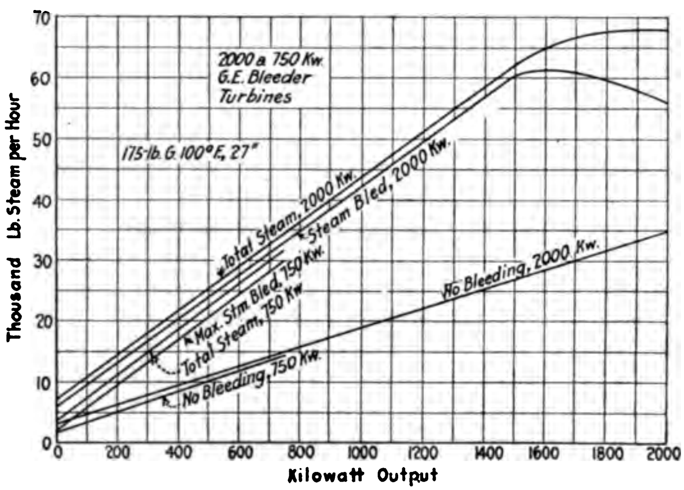

FIG. 22 WILLANS LINES OF BLEEDER TURBINES

gage, is used for industrial processes, such as soda kettles, browning chests, dry kilns, and restaurant cooking apparatus. Bleeder turbines are used, operating condensing in connection with a spray pond. The original plant had one 750- and two 2000-kw. units, but was rapidly increased by the addition of two 4000-kw. straight condensing turbines. The first two turbines are bleeders, and the third (2000 kw.) is arranged for both bleeding and mixed-pressure service. This is very desirable, as about 125,000 pounds of steam per day are used for air compressing, the exhaust being available for industrial purposes or for the mixed-pressure turbine. Approximately 385,000 lb. per day of high-pressure steam for hammers, etc., and 90,000 lb. of low-pressure steam are delivered direct to the factory. In winter, all heating service is from the bleeder turbines, and the hot-water lavatory service is from them during the entire year.

36 Fig. 22 shows the Willans lines for the bleeder turbines, which operate with steam at 175 lb. per sq. in. gage pressure, 100 deg. fahr. superheat, and 27 in. of vacuum. The auxiliaries are developed graphically in the same manner as noted with the Interborough plant. The steam consumption without bleeding, all units, is shown in Fig. 23. The winter condition, showing the effect of bleeding 60,000 lb. of steam per hour, is found in Fig. 24. The boiler-room curves are not given, as they are covered in the same manner as for a straight condensing plant.

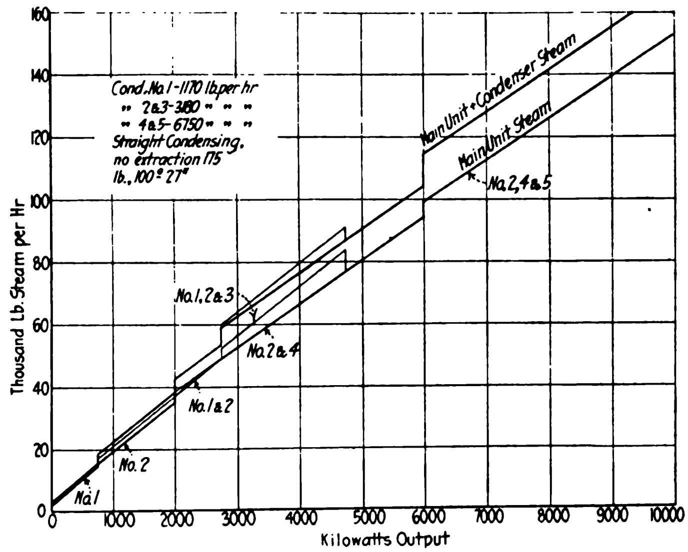

FIG. 23 ENGINE-ROOM STEAM DEMAND

37 The engine-room steam demand, no mixed-pressure service, as determined from Fig. 24, is shown in Fig. 25. The effect of bleeding is very marked, especially at the lighter loads, when a large percentage of the total steam is bled. It is found that power can be made for 0.28 cent per kw-hr. in the winter months, increasing to 0.52 cent in summer when the plant becomes practically straight condensing. This is with a ratio of average load to installed capacity of 19 per cent and a ratio of average hour to maximum hour of 62 per cent. The figures actually obtained show 95 per cent attainment of the above.

38 Of course the plant is much underloaded at present, but the

above figures will not be greatly altered by increase of load, as shown in Fig. 24. The decrease of cost due to bleeding also affects the cost

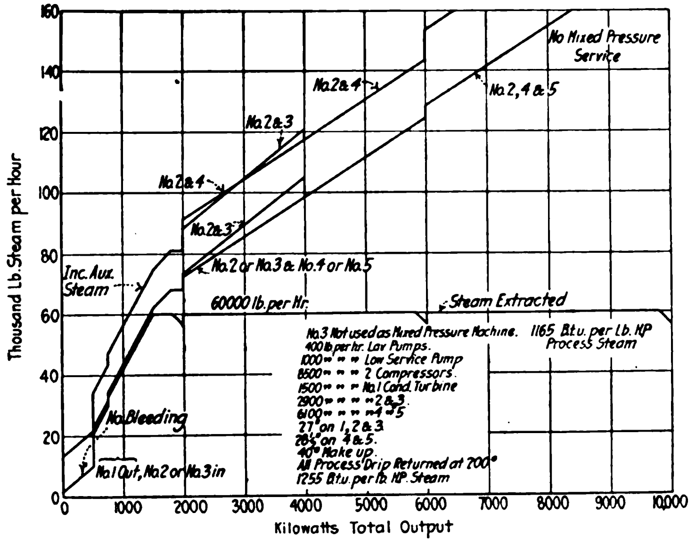

FIG. 24 ENGINE-ROOM STEAM DEMAND

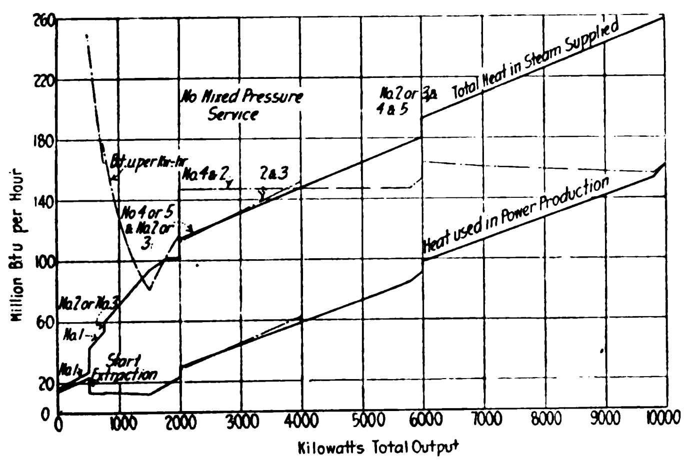

FIG. 25 ENGINE-ROOM HEAT DEMAND

of steam. Low-pressure steam is nearly 20 per cent cheaper than high-pressure steam, and both are reduced by the increased load factor on the apparatus, the installation of boilers being nearly 50 per cent less than a straight high-pressure-steam service operating condensing without bleeding. It is interesting to note that the flue-gas temperature, 400 deg. fahr. and less, average for the month, with boilers operated at 150 per cent of rating, renders the economizer useless in this plant. The boilers are arranged with a distance of 12 ft. between stoker tuyeres and bottom of header, and with special baffling to make the velocity of the gas over the tubes constant in all passes.

CONCLUSION

39 For rapid estimates, it is evident that the curves presented in this paper will give some useful relations for analyzing operating records. For instance, the engine-room steam demand with all-steam auxiliaries, in Fig. 13, balanced, might be represented with fair accuracy by the straight line $H = a + bL$,

where a = the no-load losses in the engine room = 60,000,000 B.t.u. per hr.
b = the intrinsic B.t.u. rate of the engine room = 14,000 B.t.u. per kw-hr.
L = the kilowatt load on the plant
H = heat demand, B.t.u. per hour.

This average curve smooths out the humps in the original line. It is found, by investigating eight cases, that the total heat input-output amounts between 0.30 and 0.80 load factor will lie on the curve of total B.t.u. plotted from the above straight-line engine-room demand and by use of Fig. 9, for any load curve. Variations of peak load within 15 or 20 per cent are also found to agree. In short, the average heat consumption for any load curve can be taken from this chart as the amount corresponding to the average load without correction. This is of great convenience in predicting coal consumption for change of load factor.

40 A report of the American Electric Railway Company, Power Generation Committee, 1915, gave proof of the above facts in data presented by E. F. Scofield on the Minneapolis power stations. Apparently it was not recognized that, since the boiler input-output line is a second-degree curve, the variation in coal consumption with the load cannot be a straight-line function, although it may be nearly so for short ranges if the boilers are not used at high ratings.

41 The writer has indicated only a few of the valuable features

of the graphic method of analysis in steam power plants, which not only shorten the labor in power-plant design, but reduce the chances of error and encourage a rigorous heat analysis. It is hoped that these curves will prove of value to others investigating similar problems.

DISCUSSION

Walter N. Polakov called attention to several other methods than that of using the Willans lines as advocated and used by the author. The limitations of analysis of the prospective working of a power plant purely by the Willans method has the disadvantage, which the author touched upon himself, that it does not cover entirely factors other than thermodynamic efficiency, for instance, financial factors, that are just as important and sometimes more important than the efficiency of the equipment.

Norman G. Reinicker said he used the total station water-rate curve as proposed by the author, but obtained satisfactory results without quite so much trouble. By plotting the feedwater demand, as indicated by venturi-meter readings for the same load over various numbers of days, he found that the effect of putting on an additional turbine would show up in that curve. This method saves the testing of various machines. The graphical method of analysis is valuable where the load curve can be predicted accurately; but in some plants where lighting is a large proportion of the load it is hard to predict what may be expected, and for that reason the boilers cannot be operated at the best point of their efficiency curve, but must have enough excess capacity at all times to carry the load which may be demanded.

James W. Parker said that the assumption has been made that the purpose of an auxiliary power system in a power plant and of the feedwater arrangement is to obtain auxiliary power. The problem, as he saw it, is always to heat the feedwater. At all times it is desirable to generate as much auxiliary energy as will provide exhaust steam for feedwater heating. If it is possible, during the period of light load, to take power for the auxiliaries from the main bus, and during the period of heavy plant load to transfer that from the auxiliary generator to the main bus, the main electrical system will act as a reservoir for the storage of energy, and at all times the best of the auxiliary steam will be turned into auxiliary energy.

Albert A. Cary pointed out that the size of the combustion space is not a fixed quantity for any furnace, but must be varied according to the amount of volatile matter in the coal.

Joseph Harrington took up the question mentioned by the author of the necessity of large combustion chambers in order that complete combustion may take place before the boiler tubes are reached by the gases. He believed that while the underfeed stoker is making this need less pronounced, the use of pulverized fuel in the future will make it still more urgent.

C. F. Dixon (written). We are building in Buffalo a station which will be equipped with three 20,000-kw. Curtis turbo-generators and five 11,500-sq.-ft. B. & W. boilers and Green economizers. We placed in operation on November 26 one turbo-generator and two boilers. Our operation is in conjunction with Niagara power, so that the steam plant is in operation now only about twelve to fourteen hours a day, and we are confronted with the problem of starting and stopping these large units from no load to full and reverse on very short notice. The starting is quite simple, as the underfeed stoker responds very rapidly to load demands. Our chief difficulty is in the loss of auxiliary steam during the starting period before the main unit is started. This is the only period I have ever found where electric-driven auxiliaries would show any value, and this period does not extend over 20 to 30 min., and would never occur in a 24-hour plant.

Stopping these large boiler units is the most difficult problem we have, especially with the high volatile coal used. We have found that banking with green coal is out of the question. We are able, however, to take our boilers out without loss of steam if we have 20 min. to burn down fires before load is withdrawn.

I cannot agree with the author's statement that the economizer has grown undesirable through the use of higher boiler pressures. With higher boiler pressures have also come higher boiler ratings, with their consequential higher flue temperatures, and the economizer can be made to pay for itself under these conditions. The construction, however, must be materially changed for the higher pressures.

F. A. Wardenburg (written). In this paper Willans input-output lines are ingeniously applied to the design of the power plant.

In the paper two examples are worked out, where most exact information on each unit is plotted. It is not possible to secure such accurate data except from actual test on an operating plant, and in very few cases are these data available in making a design. There is no advantage in working out a design so minutely when the figures ordinarily obtainable are speculative. A great many things vital to the design of a power plant are not subject to exact analysis and must be left to the judgment of the designer; this being the case, it is straining a point to apply such exact analysis to the parts of the design for which figures of more-or-less doubtful value are available. Further, the proposed method does not take into account return on investment, which is one of the most important considerations in the design of a power plant. The designer can, by direct figuring, determine the necessary features of design more easily than by the use of the Willans input-output curves.

The Author. In reply to Mr. Cary's inquiry as to the temperature of the steam in the Remington plant where the *average* flue temperature was said to be at or below 400 deg., I would say that the temperature of the superheated steam is 520 deg., with a saturation temperature of 380 deg. The temperature corresponding to 175 per cent rating is 450 to 470 deg.

The high setting of the boilers in the Remington plant is for the purpose of complete combustion. With former types of stokers the mixture of air and volatile from the coal was imperfect and made necessary the large combustion chamber. With the underfeed stoker the mixture is better and more intimate. The gases are formed in a porous bed of coal and pass through the incandescent fuel bed. Being thoroughly mixed with air, the process of combustion is going on during the passage through the fuel bed, with the result that surface combustion with comparatively short flame takes place in the furnace. The size of the combustion chamber may then be materially reduced. In fact, the size of the furnace is more a question of the type of stoker than the nature of the fuel.

Several who have discussed this paper seem to have mistaken entirely the source of the data on these curves. The method is practically without use if the data must be plotted from tests; in neither case were any test data available at the time the graphic curves were made up. They are made entirely from guarantees, and the data are no more exact than is usually the case in obtaining guarantees, except in that they are complete; but the use of the input-output line allows one to get the complete characteristics of a unit from two or three guaranteed points. This is especially true of such apparatus as boiler-feed pumps and fans.

The criticism that this method cannot be made to take care of financial features is not true; in one case in the paper I have done it, and it is just as easy to treat the investment costs in the same manner by the use of the double panel curve used by Mr. Stott when figuring total cost of power. If Mr. Wardenburg's statement about direct figuring being easier than the graphic method were true, there would be no necessity even for such things as load curves, because they can always be figured from the log sheets. I do not believe that anybody who has made extensive use of graphics would agree with this.

With regard to Mr. Dixon's and Professor Greene's questions raised relative to the economizers, I would say that if the use of the steel-tube unit were adopted, most of my objections to the economizer in its present form would disappear; but in this paper I have been considering constructions as they are standard on the market at the present time. Certainly the reliability of the economizer would approach that of the boiler, if put into steel-tube form.

Mr. Reinicker's method is simply a short cut permitted in an operating station by the presence of venturi meters. Naturally, advantage should be taken of the opportunity to elide some of the steps. I must repeat again that the paper is a mere indication of the application of the method. The variations in detail and in general are endless. To sum up, the use of input-output lines, both of individual apparatus and combinations of apparatus, is the easiest and safest way of getting information for all loads. Individual calculations for a single point at a time do not show the important changes taking place around cut-in points, and are not as likely to be accurate.

No. 1553

POWER-PLANT EFFICIENCY

By Victor J. Azbe, St. Louis, Mo.

Junior Member of the Society

This paper will deal not so much with the efficiency available in an up-to-date power plant, as with the improvement possible in the thousands of plants that supply power for small factories, office buildings, and various institutions, since these are by far the most prevalent and, also, the most uneconomical. While conditions are much better than they were years ago, there is still an enormous quantity of fuel wasted, and this waste is increasing as more power plants come into existence. Reduction of this waste of national wealth is imperative, for the coal resources are not inexhaustible (diminution in the supply of anthracite coal is being felt now), and even if they were, it still is the duty of the mechanical-engineering profession to conserve the financial power of the commercial public, for use in other ways of more good to humanity.

2 There are unlimited possibilities abroad for American-made goods, but to compete with foreign manufacturers we must make and sell our goods as cheaply as they do. One of the elements in the final cost of an article is the cost of power and fuel, which, ordinarily, is quite a large percentage, and a difference in its magnitude of 30 per cent plus or minus might mean the loss or profit of the manufacturer.

3 In 1915 about six hundred million tons of coal was mined in the United States, and this enormous amount is steadily increasing. Suppose the average cost of coal delivered at the power plant is \$1.80 per ton, then the total cost will be \$1,080,000,000. In the ordinary power plant, conditions as shown in Fig. 1 exist, where the shaded areas (excluding the one marked "Indicated Energy") represent the preventable losses, aggregating about 30 per cent. Of our total expenditure for coal this is \$324,000,000 each year, to say nothing of the natural gas and crude oil produced, burned and

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers. Received Honorable Mention, Junior Prize Contest, 1916.

wasted, also without the cost of handling and firing, and investment in unnecessary boilers, machinery and labor. If everything could be computed, the total loss would exceed half a billion dollars, and all due to *inefficiency of man,* which can be subdivided into:

a Lack of foresight and business ability
b Improper design
c Improper management and inefficient operation.

4 Often when a power-plant installation is planned, the owner fails to consult a competent designing engineer; ill advice from

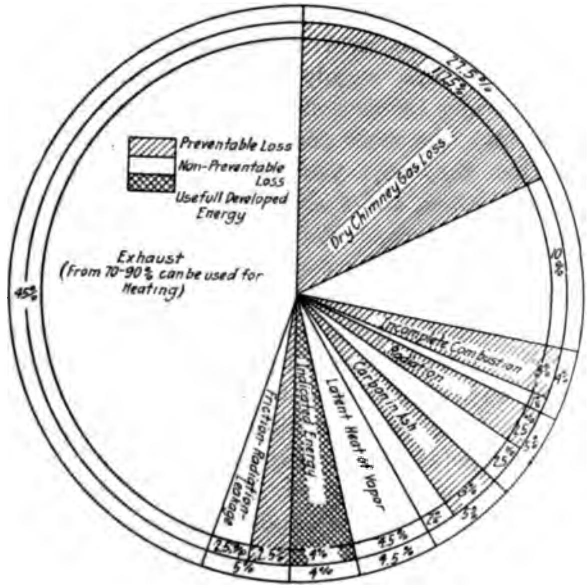

Fig. 1 Distribution of Heat Losses in the Average Power Plant Operating Non-Condensing

incompetent persons, prejudice and fear of investing needless money do the rest. The owner is not an engineer and cannot be expected to know where a greater initial expenditure for more economical power-generating equipment will pay. Therefore, every power-plant owner should have a consulting engineer proficient in economical power-plant design and operation, and every such consulting engineer should have a staff of specialists capable of examining power plants in regard to their proper upkeep and operation and of instructing the operating force thereupon. The time spent by such specialists in a plant will depend upon the condition and size of the plant, but every plant should be examined at regular intervals, and the cost of such examination will be amply rewarded by the savings due to greater efficiency.

5 The investor, ordinarily, knows little about power-plant economics, and when contemplating purchasing new apparatus may not consult an impartial, competent engineer, but deal directly with the manufacturer, who, however honest, cannot give unbiased recommendations. Often the plant operator has the first word, and although thoroughly knowing the old plant, ordinarily is not competent, from lack of experience, to make large changes and design new plants. All in all, therefore, owners are as much in need of education in power-plant economics as their employees. They should be impressed with these possibilities:

- *a* From 10 to 30 per cent fuel-consumption reduction without additional investment, simply by more efficient operation
- *b* Up to three times the amount of power from the same amount of fuel by installing proper equipment
- *c* Investments can be made in a power plant bringing from 10 to 100 per cent interest
- *d* To obtain the above, specialists on economical power-plant operation and design should be consulted.

6 Next in importance is the proper planning and designing of power plants. There are large and small power plants in operation to-day that are monuments of folly and are wasting enormous amounts of money, chiefly due to lack of foresight when designing the plant. No important change should be made in any power plant and no new power plant should be built without advice from a consulting engineer.

VALUE OF INSTRUMENTS

7 Instruments to show whether a plant is operated efficiently are generally wanting. In boiler rooms from 10 to 20 per cent of the fuel, and more in some of them, is wasted chiefly from lack of indicating devices to detect losses. Every boiler should have a draft gage and if possible a steam-flow meter. No boiler room should be without some form of gas-analysis instrument and flue-gas thermometer or pyrometer. Too much guessing is done in the average power plant, which is, in part, responsible for the miserable results obtained. Few plants pay particular attention to obtaining economic combustion; many operators have no idea what high CO_2,

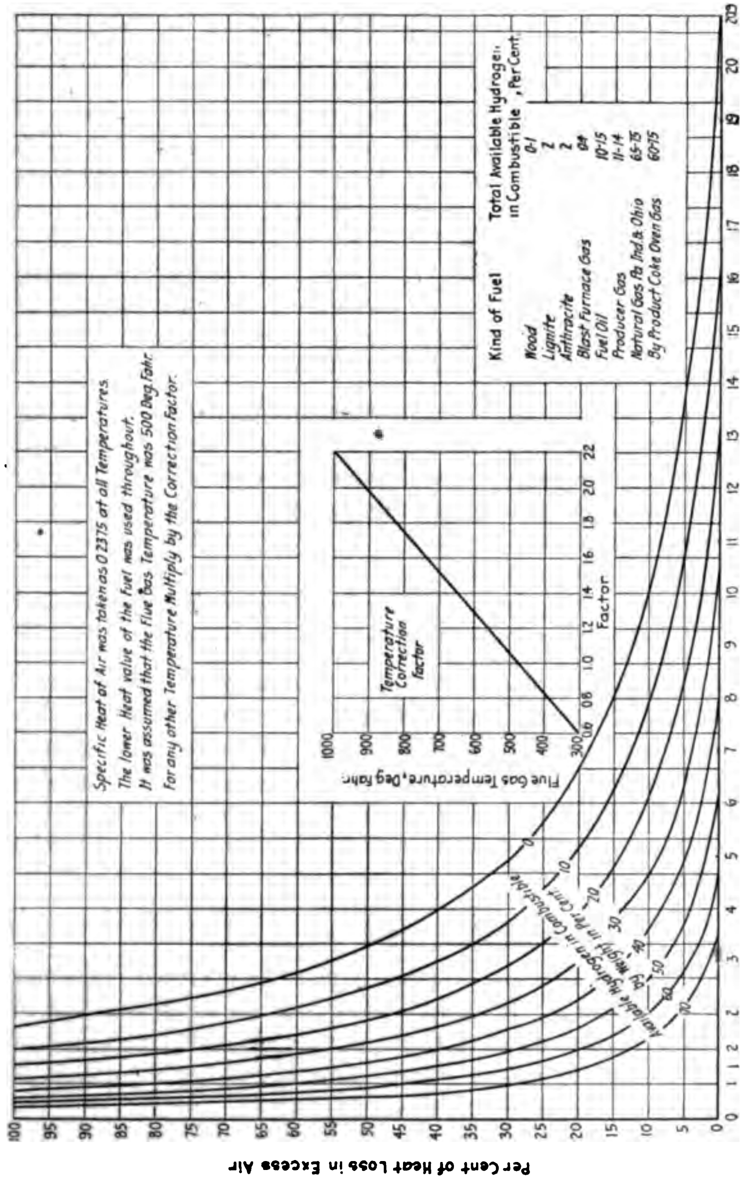

FIG. 2 HEAT LOSS FROM EXCESS AIR WITH FUELS OF DIFFERENT HYDROGEN CONTENT

low CO, and other such expressions mean. Hence it is not surprising that there are boilers operated at 40 per cent efficiency.

8 An automatic CO_2 recorder has a great moral effect upon the fireman, and numerous boiler plants can thank this apparatus for a large saving in fuel consumption. While it is of great value in any boiler plant, plants having a variable load with hand-fired boilers, and those having chain grates, Murphy and Roney stokers, etc., especially need this apparatus, since in them CO_2, ordinarily, is found the lowest.

9 The steam-flow meter on each boiler in connection with proper interpretation of the pressure drops through boiler gas passages, is also of great value and, with intelligent, trained men in charge, even preferred to the CO_2 recorder.

10 The common differential draft gage is also important, for, when properly connected, it will tell whether the boiler is being operated at, below, or above rating; and whether the fuel bed is too thick, or dirty, or full of air holes. But the draft readings must be intelligently interpreted, and they seldom are. Few operating engineers know the real value of the draft gage.

11 Flow meters are quite important in testing reciprocating pumps for their displacement efficiency, and centrifugal pumps for decrease of efficiency from wear of internal parts. When a plant is installed, provision should be made in various steam, water, and air lines for connecting a flow meter, and if none is put up permanently, a test meter should be kept to check the performance of various departments or single units of power machinery and the keeping up of the $\frac{output}{input}$ ratio.

REDUCIBLE LOSSES

12 The greatest and most neglected loss in the average boiler plant is from excess air. The curves, Fig. 2, show the magnitude of this loss at different CO_2 percentages and are applicable to any fuel, liquid, solid, or gaseous, if its available hydrogen percentage is known.

13 The impression prevails that coal must be burned with about 50 per cent of excess air. Some authorities claim that going beyond 10 to 12 per cent CO_2 the loss due to incomplete combustion will offset the saving effected by reducing the excess air. This opinion is based upon certain types of installation only, and should not be accepted and disseminated as a general condition. Some few up-to-date plants are averaging 17 per cent CO_2 with bituminous coal,

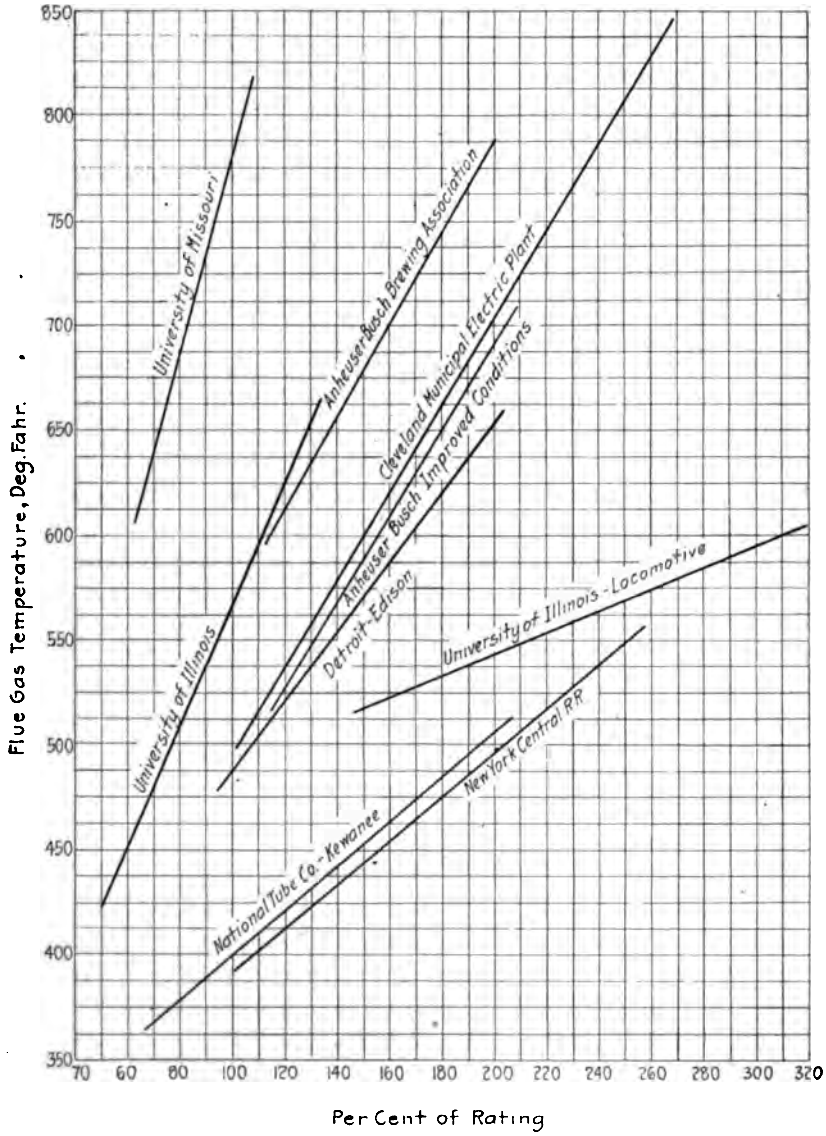

FIG. 3 FLUE-GAS TEMPERATURES AT VARIOUS BOILER RATINGS IN SEVERAL DIFFERENT PLANTS

without serious losses due to incomplete combustion. The author himself obtained an analysis giving CO_2, 17.2 per cent, O, 1 per cent, CO, 0.1 per cent, which appears remarkable with bituminous coal, but is easily possible if the furnace construction is right. Hence it is ridiculous to use 200 per cent excess air as is often done; also, 50 per cent is more than is needed, and with proper furnace design not more than 10 per cent excess air is required.

14 One objection to high CO_2 is the extremely high furnace temperature and consequent rapid depreciation of furnace linings. For this reason many maintain 10 per cent CO_2 as the maximum. On the Detroit Edison Sterling boilers, however, where CO_2 is maintained at from 15 to 17 per cent, the furnace lining shows no effect of high temperatures.

15 With low percentages of excess air, proper design of combustion space becomes much more important, to prevent incomplete combustion and too high furnace temperatures. Exceedingly high furnace temperatures have no real advantage and several disadvantages, therefore heating surface exposed directly to the fuel bed to reduce the furnace temperature will not only cause a decrease of the slagging effect upon the firebrick, and thus increase the CO_2 possible to maintain, but will also increase the efficiency of the boiler due to the heat radiated directly to it, and will increase the boiler capacity without increasing the flue-gas temperature. With a proper furnace volume smoke will not form if provisions for gas mixing are maintained and the temperature does not drop off too low or the boiler is not forced too much.

16 Heat transmission in steam boilers will not be treated at length here, but it should be emphasized that velocity of gas, time of contact, and hydraulic mean depth are so important that they deserve thorough study under all conditions of installation. Not much is being done in this direction and quite large boiler builders ignore the matter altogether.

17 Fig. 3 shows the flue-gas temperatures with different well-known installations at various boiler ratings. It shows that results obtained are very varied, and that some installations obtain the same final temperature at 300 per cent of rating as others at 100 per cent. In one case at 200 per cent of rating the temperature of the escaping gases was 285 deg. higher than in another, and this in spite of the fact that in both cases the amount of air used to generate 10,000 B.t.u. was the same. The seriousness of this will be realized from the fact that this condition causes a loss of about

11 per cent in fuel consumption to the high-temperature plant. This is at 100 per cent overload; at higher ratings this difference will increase as indicated by the curves.

18 Boilers can be operated at two times the rating, and more, but not economically unless designed for it; with the rates of heat absorption obtained at present in most installations, if operated at very high ratings the loss will be too great, unless the temperature of the escaping gases be reduced in economizers. The increase of velocity will cause draft loss, but this is not so serious, for the necessity of installing mechanical draft brings other advantages.

19 With high ratings and high velocities the power required to produce the draft becomes of interest. Fear that so much of the steam generated will be required to produce the draft that the advantages obtained will be offset is unfounded, if the blower is designed for efficient work and driven by an efficient engine or motor.

20 The author has found reduction of flue-gas temperature possible in most boiler installations by reducing excess air, increasing velocity, eliminating dead-gas spaces, re-baffling or installing auxiliary baffles, exposing some of the heating surface to direct radiation from the fuel bed, and last, but not least, maintaining clean heating surfaces by blowing with steam at regular intervals and scraping the heating surfaces when the boiler is down for cleaning.

ADVANTAGES OF ECONOMIZERS

21 Boiler-plant efficiency can be considerably increased by adding economizers, which are used in most European plants, but find little favor in this country. The objections ordinarily offered are:

- *a* Their initial cost
- *b* Reduction of available draft and, in many plants, necessity of installing mechanical-draft systems
- *c* Cheap fuel (speaking comparatively)
- *d* Low load factors
- *e* Complication of the power-plant outlay.

The chief reason that they are not used more generally is that their value is not realized. It is difficult to reason out just what effect load variations have upon an economizer as a heat absorber. If plants were operated continuously for 10 or 12 hours and completely shut down the rest of the time, the saving would be in proportion to the load factor, but where the load remains on, and only varies from overload to underload, the value is difficult to determine. But since such plants are ordinarily operated uneconomically at both high and low loads, and since economizers are of greater value the lower the boiler efficiency, they are particularly needed in plants having variable loads, and the saving in fuel, if not more, will be at least proportional to the load factor. Much depends upon properly proportioning the installation, and, therefore, before an economizer is decided upon, a plant must be closely studied and the governing factors, such as flue-gas temperature, feed-water tem-

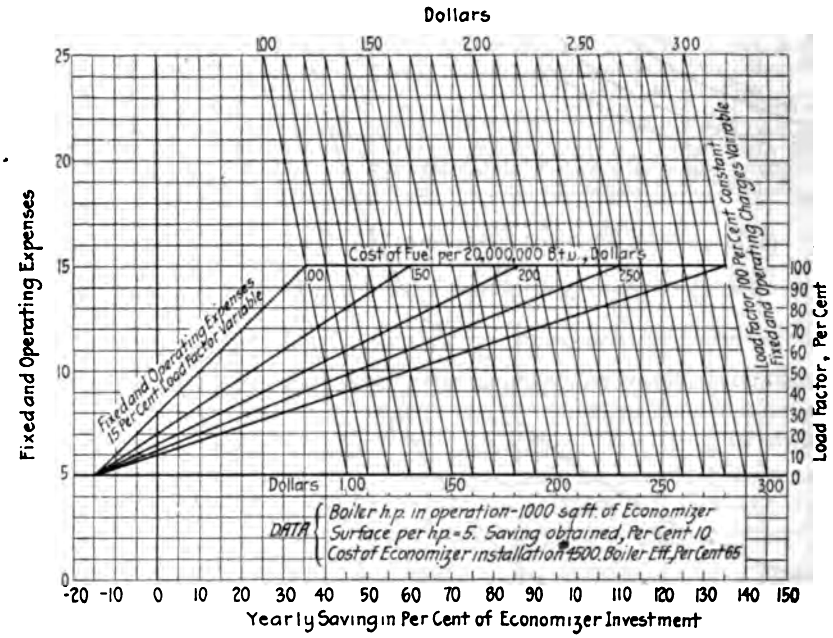

FIG. 4 EFFECT OF FIXED AND OPERATING EXPENSES AND LOAD FACTOR UPON SAVINGS OBTAINED FROM ECONOMIZERS

perature, weight of air per pound of coal, load factor, character of load, etc., very closely predetermined.

22 Fig. 4 shows that economizers are economical, not only in large, but also in small, power plants. It shows that the initial investment is warranted and usually will bring good returns. It was calculated upon the assumption that the fuel saving is 10 per cent; 5 sq. ft. of economizer area was allowed per boiler horsepower, and it was assumed that the boiler efficiency, without an economizer, is 65 per cent. The total cost of the economizer installation was

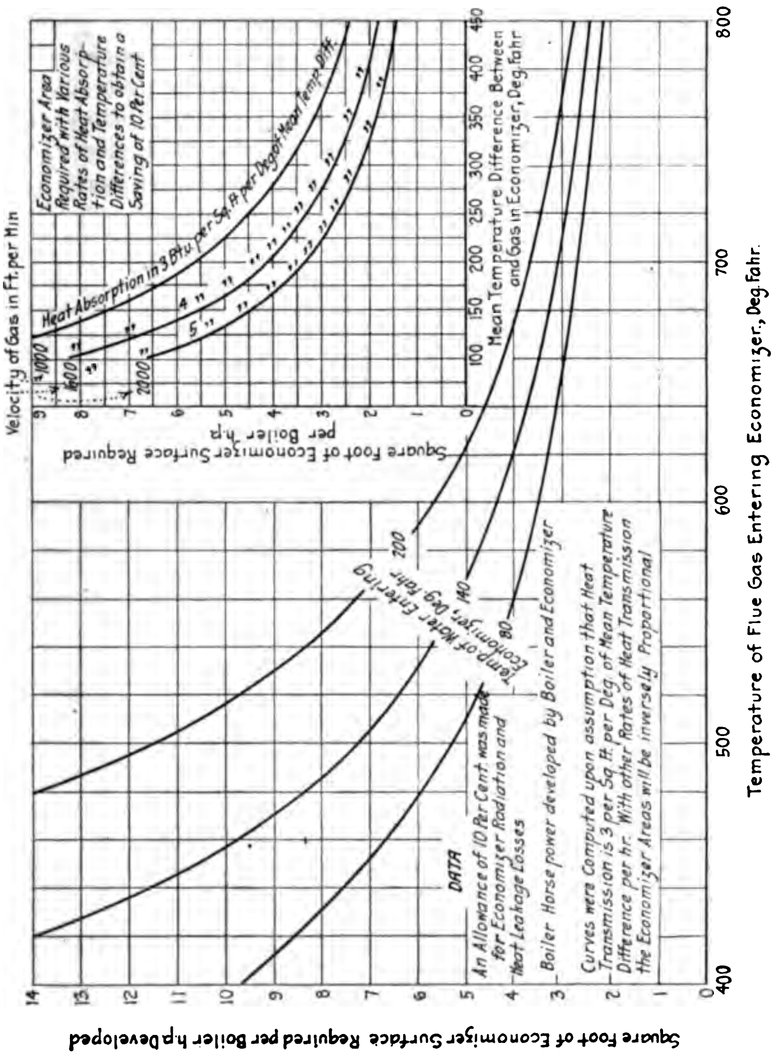

FIG. 5 ECONOMIZER SURFACE REQUIRED TO SAVE 10 PER CENT IN FUEL WITH DIFFERENT FLUE-GAS AND FEED-WATER TEMPERATURES

taken at $4500.00, which is about the actual cost except where extensive alteration must be made to accommodate it. This chart also plainly shows what effect fixed and operating charges and load factor will have upon the savings obtained from economizers.

23 Fig. 5 shows the square feet of economizer area required per boiler horsepower to save 10 per cent in fuel with different flue-gas and feed-water temperatures. Other charts and data dealing with economizer theory and practical performance are given in Figs. 6 to 9, and a table of installation costs follows:

AVERAGE COST OF ECONOMIZER INSTALLATION IN MIDDLE WEST OF UNITED STATES

Boiler Horsepower Developed	Cost in Dollars		Boiler Horsepower Developed	Cost in Dollars	
	Economizer Surface Per Horsepower			Economizer Surface Per Horsepower	
	3 Sq. Ft.	5 Sq. Ft.		3 Sq. Ft.	5 Sq. Ft.
250	1100	1800	2,000	6,000	8,600
500	1800	2850	4,000	11,500	16,800
700	2300	3600	6,000	17,000	25,000
1000	3000	4500	10,000	26,700	42,000
1500	4500	6600	......	......	......

24 Economizers do considerably reduce the draft, but often much of the total loss is the fault of the gas passages from the boiler to economizer and from them to the stack. The drop of draft through an economizer 40 sections long will be about 0.25 in. of water pressure if the mean gas velocity is 1500 ft. per min.; for other lengths of economizers the draft will vary with the length, and for other velocities as the square of the velocity. (See Fig. 6.) With natural draft there will be additional reduction of available draft resulting from the reduction of the stack temperature.

25 The loss of draft because of economizers is partly offset by the higher efficiency and therefore smaller amount of coal burned, and also by the ability to operate more economically the auxiliaries that previously exhausted into a feed-water heater, which, saving steam, will somewhat reduce the amount of coal burned and, consequently, also the draft required. Maintaining high CO_2 percentage, together with having tight breeching, clean-out doors and dampers on idle boilers will tend to increase the available draft. If the stack height is so low or the boiler load so high that all the above will not insure sufficient draft at all times, mechanical draft will have to be resorted to.

26 Installing mechanical draft will in one way reduce the saving, due to added interest on investment, depreciation and operating charges with the power required, but all this can be offset by the

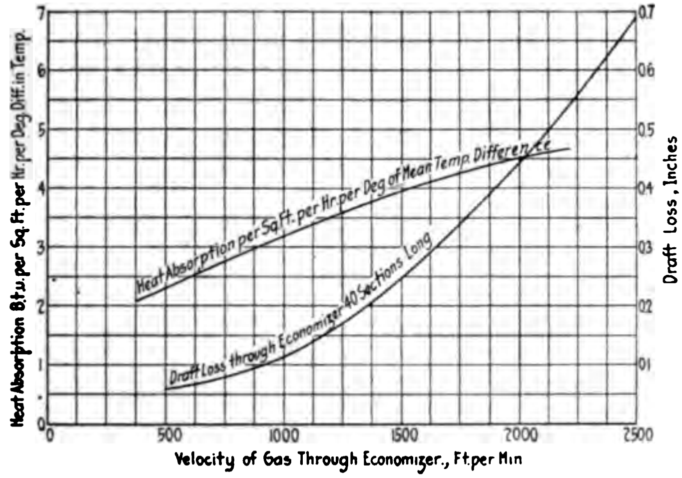

FIG. 6 EFFECT OF VELOCITY UPON HEAT TRANSMISSION AND DRAFT LOSS IN ECONOMIZERS

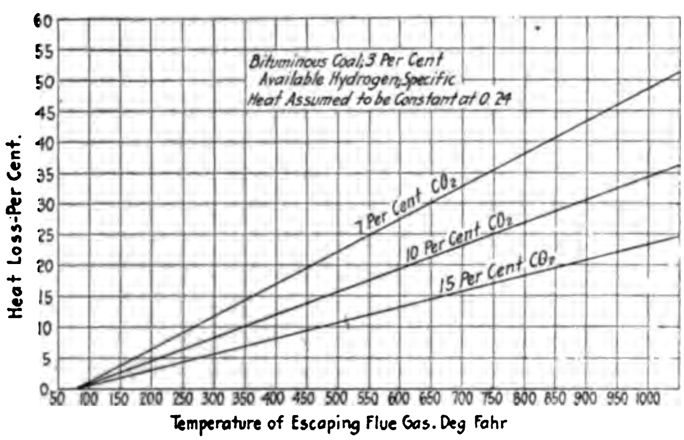

FIG. 7 HEAT LOSS IN DRY FLUE GAS WITH VARIOUS CO_2 PERCENTAGES AND DIFFERENT UPTAKE TEMPERATURES

greater boiler efficiency possible with thicker fuel beds and, consequently, higher CO_2, and greater rate of heat absorption attending higher velocity of the gas over the heating surface of the boiler and economizer.

27 Cleaning economizers is not simple. In some plants air and steam are used for blowing off the soot with considerable success, but this is effective only with loose matter. The hardened soot will not

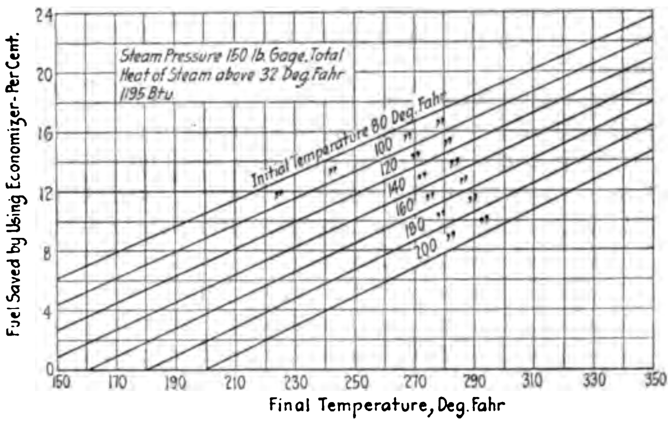

FIG. 8 SAVINGS FROM ECONOMIZERS WITH VARIOUS INITIAL AND FINAL FEED-WATER TEMPERATURES

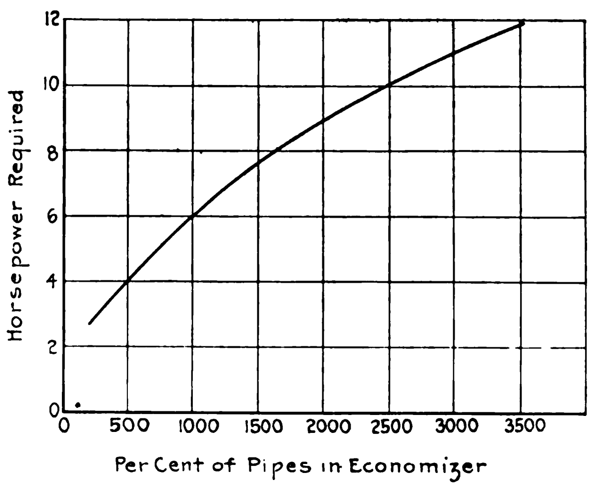

FIG. 9 APPROXIMATE POWER REQUIRED TO DRIVE SCRAPER MOTORS ON VARIOUS SIZES OF ECONOMIZERS

be affected, and the blower tubes cannot be so arranged that blowing will be equally effective all through. Best and most effective are scrapers, and when they give trouble by sticking, it is better to force them by hand until again working smoothly, than to let them stop and allow the tubes to scale up so much that the further constant operation of the scrapers becomes almost impossible. The labor required to keep economizers operating and clean should not be charged against the economizer, since there will be a fully compensating reduction in boiler-room labor because of there being less coal to be burned.

28 Where the natural draft is too weak to warrant a fuel economizer and conditions are such that mechanical draft cannot be considered, there is still a way to improve the working conditions. At least 40 per cent of the boiler heating surface is inefficient as a heat absorber because of the low temperature difference, and any increase in temperature difference will reduce the amount of heating surface required to cool the gases down to a given point. Fig. 10 shows the effect of establishing a greater temperature difference by reducing the boiler heating surface per nominal boiler horsepower and adding economizer surface. With a feed-water temperature of 100 deg. fahr. the total heating surface per b.h.p., without an economizer, to cool the gases would be 10 sq. ft. With an economizer absorbing 3 B.t.u. per sq. ft. per degree of mean temperature difference per hour, the total heating surface of the boiler and economizer would have to be only 8.55 sq. ft. or 14.5 per cent less, and with a rate of heat absorption of 4 B.t.u. by the economizer (see Fig. 6) the total heating surface necessary would be 7.25 sq. ft. or 27.5 per cent less than when water is fed directly into the boiler without preheating.

29 Fig. 10 shows also the ratio of boiler to economizer surface required with different initial feed-water temperatures, and also the total heating surface required in each case. The saving of heating surface by establishing a greater temperature difference will become less with higher feed-water temperatures, but even if the temperature increases up to 210 deg. fahr. it still is possible to absorb the same amount of heat with 10 per cent less heating surface.

30 The draft loss will not be any more with this installation than when no economizer was attached, but even less, which can be taken advantage of by increasing the gas velocity over the added economizer heating surface. The installation could follow the European practice of building the economizer directly into the boiler setting, which would further reduce the draft and radiation losses. The economizers could also be of boiler steel since the temperature would never drop to a point where danger would exist from external corrosion, and to further prevent this, water entering should never be below 100 deg. fahr.

31 Some engineers object to economizers because of their cast-iron construction, and do not want to employ them with the high steam pressures maintained in modern plants. One way to over-

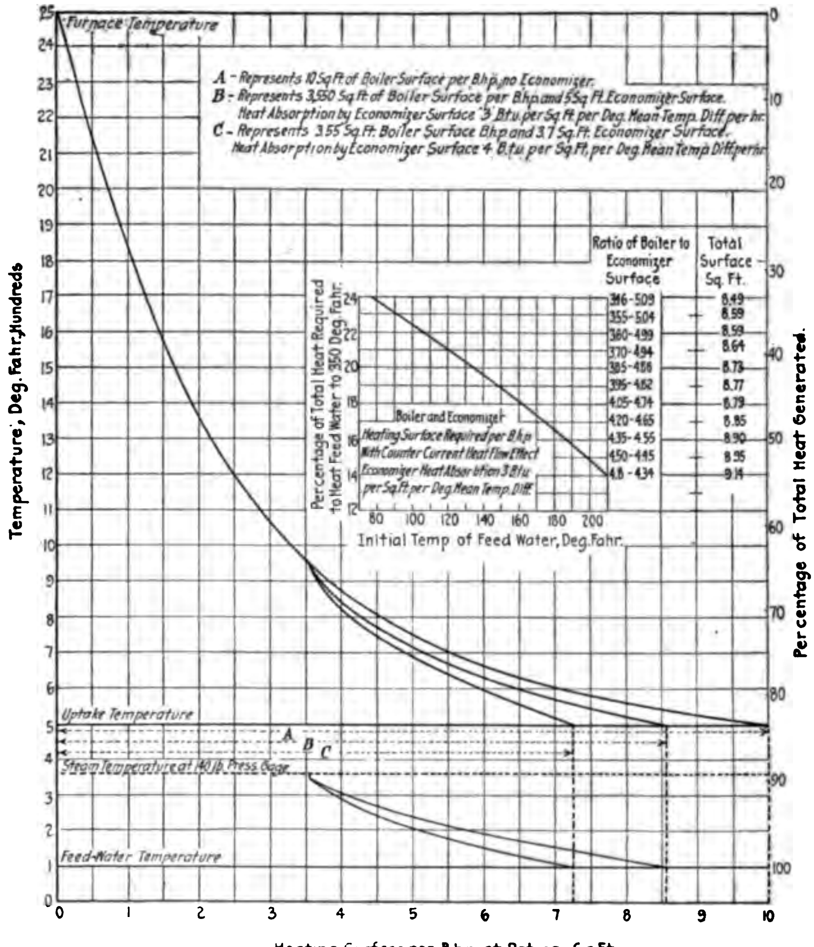

Fig. 10 Effect of Establishing Counter-Current Heat Flow by Reducing the Boiler Heating Surface per Nominal Boiler Horsepower and Adding Economizer Surface

come this objection would be to locate the boiler-feed pump between the economizer and the boiler, but with this arrangement full benefit could not be obtained from the economizer, since the feed water could not be heated any higher than the evaporating temperature at the pressure maintained.

32 A solution for this would be to build economizers in two sections, as shown in Fig. 11, one for low-temperature gases, through which water at low pressure would circulate and be heated up to somewhere around 210 deg., depending upon the pressure, and another in series with this for high-temperature gases and full boiler

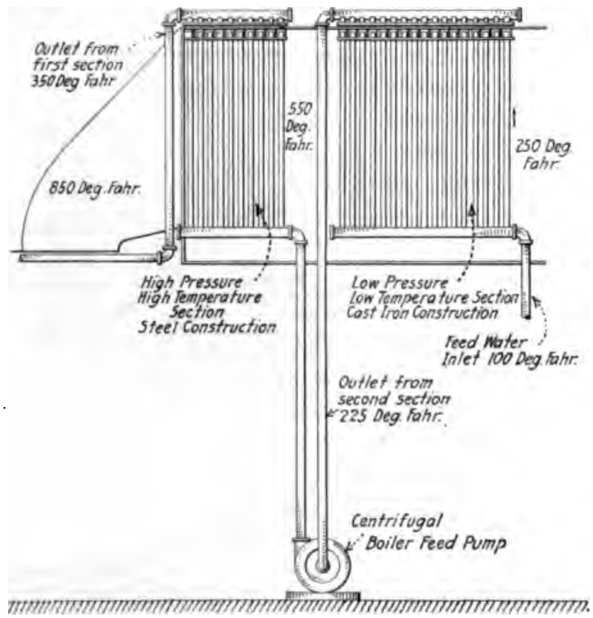

FIG. 11 A TWO-SECTION ECONOMIZER CONSTRUCTION — CAST IRON FOR LOW PRESSURE AND TEMPERATURE; STEEL FOR HIGH PRESSURE AND TEMPERATURE

pressure. The low-pressure, low-temperature section would be of cast iron because the low-temperature gases are highly corrosive of steel, and the high-pressure, high-temperature section, where the temperature will never drop below 350 deg., would be of boiler steel. The boiler-feed pumps would be between the two sections.

33 The ratio of high-temperature to low-temperature section would depend upon the initial and final flue-gas and feed-water temperatures.

34 With the above arrangement the internal corrosion would be prevented, since the low-temperature tubes would be cast iron, and by the time the water reached the steel section it would be at a

temperature where it deposits inorganic precipitate which would form a protective coating over the heating surface. The external corrosion would be prevented by the fact that both the water and the gas would be at high temperature when in contact with the steel section, and there would be no danger of condensing any vapor in the flue gases. We must not forget, however, that care has to be taken to prevent the temperature in the steel section from dropping too low, or corrosion will set in. But, ordinarily, the above arrangements will not be necessary, since economizers can be built of a cast-iron and semi-steel composition with a safety factor of five for any boiler pressure.

PROPER FEED WATER

35 Much heat is wasted by dirty boilers, and the cost of cleaning in most plants is considerable. Using distilled water as boiler feed should be advocated whenever possible. In large plants the condensate from the condensers of the main units could be used to condense the exhaust steam from the auxiliaries, and all the make-up water obtained from a multiple-effect evaporating system, or flue-gas evaporators working under vacuum. In other plants more attention should be paid to treating the feed water.

36 Properly treated feed water will not only improve the efficiency of a boiler as a heat absorber, reduce the cost of boiler cleaning and its repairs and increase its life, but will also greatly increase its safety, since most boiler explosions and accidents can be traced to unsuitable boiler feed water.

37 City boiler-inspection departments, with the aid of water departments, could be of great aid to the operating engineers in the particular cities by keeping them informed as to what treatment the water requires to make it suitable for boiler feed. Much good could also be done by the boiler-insurance companies, not only to the engineers and power-plant owners, but also to themselves, if they had chemists to inspect samples of feed water and advise proper treatment.

PREHEATED FURNACE AIR

38 Preheating of furnace air with the heat in the escaping flue gases would save considerable, but it is never practiced. A saving of 5 per cent is easily possible and air preheaters to effect this saving could be constructed cheaply, but as they would be quite bulky they could be located outside of the boiler room. Most simply constructed

they would consist of a long sheet-iron compartment through which the air and flue gas would pass, as shown in Fig. 12. The cost of installing would be low, except if an extensive building alteration were required, but mechanical draft would be necessary since there would be not only more friction of the flue gas and less stack draft because of lower temperature, but also more power required to force the air into the furnace. These heat abstractors could be easily cleaned by hand, or like economizers.

39 The air in summer could be taken from the boiler room and thus maintain the ventilation and keep the temperature down, while in winter it could be taken from outside, thus insuring a comfortably

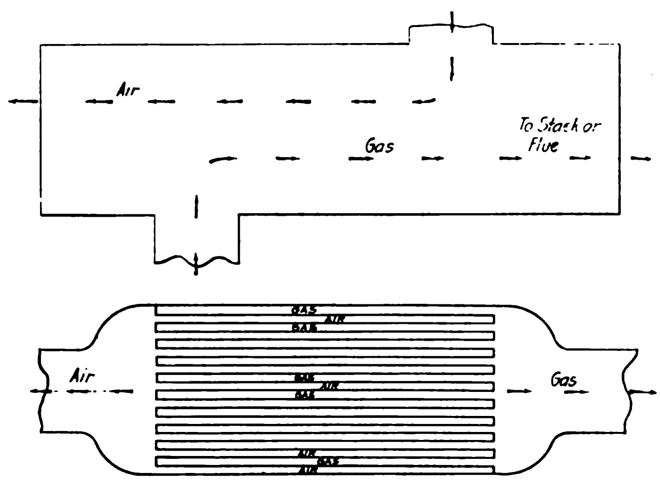

FIG. 12 DESIGN FOR FURNACE-AIR PREHEATER

warm boiler room and reducing somewhat the loss by radiation from the boilers and steam piping.

40 In larger central stations the use of air from the turbo-generators, as practiced by the Detroit Edison Co., is commendable not only because it saves heat, which, in this case, is small, but chiefly because it cools the engine room, increasing the comfort of the men and making them more efficient.

ECONOMICAL AUXILIARIES NEEDED

41 While in many power plants, especially the larger ones, considerable attention is paid to the main power-generating units, the auxiliaries are ordinarily neglected. Throttling steam engines and

especially direct-acting double and single pumps are too generally used. Unless they exhaust into feed-water heaters or other heating appliances, they should not be used except in very small units, or only intermittently for short periods. Substituting motor-driven triplex plunger pumps, or motor- or turbine-driven centrifugal pumps would usually result in a substantial saving. Where duplex steam pumps are preferred and the plant operates condensing, in a small plant, the exhaust steam from the pumps could be discharged into the condenser of the main units, or in a large plant, a separate condenser could be installed especially for the auxiliaries, and the discharged circulating water from the main condensers, with the make-up water, could be used to condense the steam from the auxiliaries.

42 Many power plants that have economical prime movers have auxiliaries that are so uneconomical that the efficiency of the whole plant is greatly reduced.

43 Using exhaust steam from reciprocating steam pumps in the low-pressure stages of a steam turbine designed for this purpose is also quite a saving, and there are many other such means that would effect large or small savings of power, but the operatives must be trained to reason and discover the ways and means to accomplish these savings.

44 The use of high-pressure steam in manufacturing and heating should be discouraged; while sometimes its use is warranted, generally the use of low-pressure steam will effect a saving if the pressure is reduced through small turbines, engines, or pumps, as, thus, the mechanical energy will be extracted between the high- and low-pressure limits, without materially reducing the heating qualities of the steam. Using large reducing valves to reduce steam pressures, while not a waste of heat, is a distinct waste of power.

UTILIZING EXHAUST STEAM

45 The value of exhaust steam for heating and the consequent saving effected has been demonstrated, and many plants take advantage of it, but the practice should be more general. Even plants operating condensing could bleed the low-pressure receiver of a compound engine or the low-pressure stages of steam turbines and distribute this steam through the buildings. In power plants where economizers are not employed and the temperature of the water is low when fed into the boilers, this partly expanded steam could also be used to advantage in feed-water heaters.

46 In plants where large quantities of low-pressure steam are

required and the pressure cannot be expansively reduced in a cylinder or turbine, a saving still could be made by generating this steam in a boiler under low pressure. It has been experimentally determined that there is about 6 per cent difference in boiler efficiency between generating steam at near atmospheric pressure and at 350 deg. fahr. because of the increased temperature difference between the furnace gas and the boiler water, greater heat absorption and consequently lower flue-gas temperatures.

EXTRAVAGANT REFRIGERATING PLANTS

47 About the most wasteful power plants are the ice-making and refrigerating plants; many make three or less tons of ice per ton of coal, a few about six tons, and only the very exceptional plants reach the ten-ton mark. All except the very small ones and those near coal mines should be making ten tons of ice per ton of coal burned, and none less than six tons.

48 One reason for the existing bad conditions is the low load factor. While during warm weather the equipment is forced far beyond its economical rating, in winter it stands idle. This low yearly load factor, which, in most plants, is between 20 and 40 per cent, is to a certain extent responsible for many plants not being more successful and profitable. There are ice plants, for example, the Polar Wave of St. Louis, that operate with a load factor of over 100 per cent the year round. This is made possible by providing proper-size ice-storage houses, which reduces the necessary size of power plants, machinery, ice tanks, etc., and consequently also the fixed and operating charges.

49 Besides the variable load conditions, the following are factors responsible for uneconomic conditions:

a Inefficiency in the boiler houses, where often the coal is burned with from two to four times the air actually required

b The fact that distilled-water ice is made and uneconomical prime movers and auxiliary machinery are operated so that sufficient exhaust steam for distilled water will be obtained. This type of plant also condenses the exhaust steam in atmospheric condensers, with the result that the engines operate against a back pressure instead of under a vacuum

c Due to a lack of thorough understanding on the part of many operators of the physical laws governing the compression of gases, refrigeration is often obtained with from 20 to 40 per cent more power than actually required. Many operators insist that ammonia compressors operate best with low back pressures, while exactly the opposite is true.

50 To be able to operate with high back pressures, ice tanks, brine tanks, and coolers should be designed and piped so that proper temperatures will be maintained without too great a temperature difference between the air (or brine) and the evaporating ammonia. Where two (one high and one low) temperatures must be maintained, the power required per ton of refrigeration will be greatly reduced by installing high- and low-pressure suction lines connecting with a multiple-effect compressor attachment, thus enabling the compressor to operate with very nearly the efficiency obtained with the higher back pressure.

51 A compressor operating against a condenser pressure of 185 lb. and with 12 lb. suction pressure, will require 3.07 b.h.p. per ton on ice making, while the same compressor operating with 35 lb. suction pressure will require only 1.9 b.h.p., or a gain of 38 per cent. An economical ice plant will cool the liquid ammonia leaving the condenser down to the temperature of the coolest water by passing the water on the way to the condenser through a liquid-ammonia cooler. The heat extracted here will be a clear gain. From the water cooler the ammonia will pass to a second cooler where the liquid ammonia will be further cooled by the evaporation of ammonia on the outside of the liquid coil at a high evaporating pressure, preferably 35 lb., or by gas returning from the expansion coils in heat exchangers. The distilled water, in the meantime, will be cooled to 40 deg. in the fore-cooler by liquid ammonia evaporating at 35 lb. pressure. Thus about 40 per cent of the work will be done at 35 lb. and 60 per cent at 12 lb. The ammonia gas from the liquid ammonia, water fore-coolers and ice tanks will then be led into two separate suction lines to either two different compressors, or to a compressor with the two ends separated and one compressing gas at 35 lb. and the other at 12 lb. suction pressure. By this arrangement about 15 per cent less power will be consumed, and the capacity of the compressor and ice tanks will be greatly increased.

52 Raw-water-ice systems have been so perfected that ice can be made from raw water that is wholly equal to distilled-water ice. Where the water is very bad, it can be purified or a multiple-effect

distilled-water system used, avoiding any occasion to employ uneconomical steam machinery to get the required quantity of ice water.

53 The compression system of ice making with simple non-condensing Corliss engines and its production of six tons of ice for every ton of 12,000 B.t.u. coal burned under its best conditions, high load factor and excellent operation, could be improved upon by various other combinations, as follows:

	Tons of ice per ton of coal
Compression, simple Corliss, non-condensing.....	6
Compression, compound Corliss, non-condensing..	7
Compression, simple Corliss, exhausting into generator of absorption machine.....	8.5
Compression, compound Corliss, condensing......	9.5
Compression, compound Corliss, exhausting into generator of absorption machine..	11

54 These estimates are for well-operated ice plants, and the production could be further increased by adding special devices and by special arrangements, economizers, multiple-effect compressor attachments, flooded and jet ammonia condensers, vertical small clearance compressors, heat exchangers, flooded systems, etc., also by using Uniflow engines supplied with highly superheated steam. In some refrigerating plants Diesel and semi-Diesel engines are used to drive ammonia compressors with great success, while in others they are not profitable, but this is chiefly the fault of improper design, management and operation and not of the oil engine.

55 By operating ammonia compressors properly, that is, with a high discharge temperature and leading the gas through a feed-water preheater, considerable heat could be saved and thus allow, if economizers are not used, employing more efficient auxiliaries or discharging their steam into the generator of the absorption machine.

56 Condenser efficiency is also very important in ice- and refrigerating-plant operation and it is regrettable with what persistence the low-efficiency straight-pipe condensers are used, while there are excellent types of flooded, jet and other condensers on the market. Some of these condensers are from two to three times as efficient as those in general use because they transfer heat from liquid to liquid instead of from liquid to gas. An entirely new type of condenser, employing high velocity and counter-current effect, gives a very high efficiency and remarkable results.

57 Many large plants are operated non-condensing, and while

the exhaust steam, in winter, is used for heating, in summer, a large part of it is ordinarily wasted. It would pay some one to buy this steam and use it in an absorption ice machine. Another good combination would be an exhaust-steam absorption ice plant and an electric-light plant, with the resulting high efficiency of the two combined.

58 Opportunities for economy in ice plants are great, not only by better equipment and arrangement but also by better operation, for many plants are ridiculously inefficient. While operating engineers are coming to realize the importance of frequently indicating the steam engine, they are slow in becoming impressed with the importance of indicating the ammonia compressor. Although simply inspecting the ammonia-compressor indicator card will not reveal much, by drawing the adiabatic and isothermal curves and comparing their relative position to the compression curve, and by drawing the suction and discharge pressures existing in the pipes leading to and from the compressor and noting their relation to the suction and discharge pressures in the cylinder, a good idea of conditions will be gained. Thermometers in the suction and discharge pipes at the compressor and proper interpretation of the readings obtained are also of great value in the efficient operation of an ammonia compressor.

59 The author knows of ice plants where compressors operate with only about 70 per cent of their volumetric efficiency and, in addition, the wire-drawing of ammonia gas through improperly designed suction piping and suction valves causes an increase of 10 to 15 per cent in the power required per ton of refrigeration, because the ammonia in the cylinder before compression is more rarefied than in the evaporating coils. These conditions would not have been discovered without the indicator.

ECONOMICAL PRIME MOVERS

60 The uniflow engine is important to the economical operation of the smaller power plants, and has these advantages:

a Simplicity of valve gear and reduced friction
b Large overload capacity
c Flat steam-consumption curve through various ratings
d Reduced piston leakage and no exhaust valve leakage
c Poppet valves permitting use of very high superheat without complications

f Remarkably low steam consumption because of great reduction of cylinder condensation and re-evaporation

g Consequent to the foregoing, smaller investment in boilers, stokers, chimneys, etc.

61 The cost of uniflow engines is not excessive, ranging from \$14.00 per i.h.p. in 320-h.p. units to \$16.00 in 190-h.p. sizes, unerected and without foundations.

62 The locomobile also deserves honorable mention and for most small plants is ideal. Its concentration and high efficiency are remarkable and with it a small power plant can produce one i.h.p-hr. on 1½ lb. of coal and thus be equal in efficiency to some of the largest power plants. Exhaust-steam turbines operating under proper conditions have demonstrated their benefits and should be used more generally.

63 Superheated steam also should be taken advantage of more than it is. In very many plants substantial savings could be made by installing superheaters. Some that now have them pay little or no attention to whether the proper amount of superheat is obtained, and others have them almost cut out of service, ordinarily because of fancied objections. Very often, only enough superheat is obtained for the steam to reach the engine cylinder barely saturated, and thus little or no return on the superheater investment is obtained.

64 There are difficulties in operating Corliss engines with superheated steam; but, ordinarily, not enough effort is made to prevent them and often they are greatly exaggerated. Obstacles encountered should be overcome instead of condemning superheated steam without further investigation and trials. Often the change to a better grade of cylinder oil will settle the whole difficulty. If, after thorough inspection, the trouble is not eliminated, the highest possible superheat to maintain without difficulties should be determined and maintained.

65 Producer-gas plants, gas engines, and Diesel engines should find a much greater application. With proper care, gas and oil engines to-day are entirely dependable and much more efficient than steam engines. The advisability of their use depends upon the price of the fuel, but, in any event, the burning of oil and gas under boilers should be discouraged, since they are much more economically used in gas and heavy-oil engines. The gas producer's chief economic advantage is that it can use low-grade fuels unsuitable under steam boilers. The quantities of fuel wasted at the mines could

be used in gas producers supplying gas engines driving electric generators and the current distributed for sale.

66 The economical power plant is necessarily more complicated, and very often objections offered to producer-gas plants and gas and oil engines are that they require better operating men. This is so, and therefore we must begin to develop men thoroughly fitted to operate the up-to-date power plant. One-sided development such as inventing more efficient machinery should be balanced by educating the men to handle it properly.

67 Economical power-plant machinery, equipment and methods of design are available, still there remain many extremely uneconomical power plants.

RELATIVE EFFICIENCY AND HEAT CONSUMPTION OF VARIOUS POWER INSTALLATIONS

68 Fig. 13 serves to show at a glance the relative value of various power installations from a thermal standpoint. Effort was made to have the given values, as far as possible, represent the average conditions, and while better results can be obtained (and also worse), the ones given will be ordinarily found.

69 All sizes of prime mover varied from 300 to 400 b.h.p. except the simple non-condensing engine and the locomobile which were 150 b.h.p., the turbine, 4000 k.w. and the Nürnberg gas engine, 1200 b.h.p.

70 Boiler efficiency in each case was assumed to be 70 per cent; an allowance of 3 per cent was made for the boiler feed pump, and 2 per cent for leakage and condensation. For the condensing engines the auxiliary steam consumption was assumed to be 5 per cent of the total steam consumption. The temperature of the feed water entering the boiler was assumed to be 200 deg. fahr., to which point it was heated by either exhaust steam or an economizer.

71 For the uniflow engine when operating non-condensing with superheated steam, the superheat was 120 deg. fahr., and when operating condensing, 100 deg. fahr. with 25½ in. of vacuum.

72 The turbine tests represent a 4000-kw. horizontal turbine operating under 28 in. of vacuum, 175 lb. steam pressure and saturated steam. The effect of superheat is shown by the extension of the curve at full load.

73 The locomobile curve represents tests made on a 150-b.h.p. locomobile with a varying high-pressure superheat of 80 deg. at

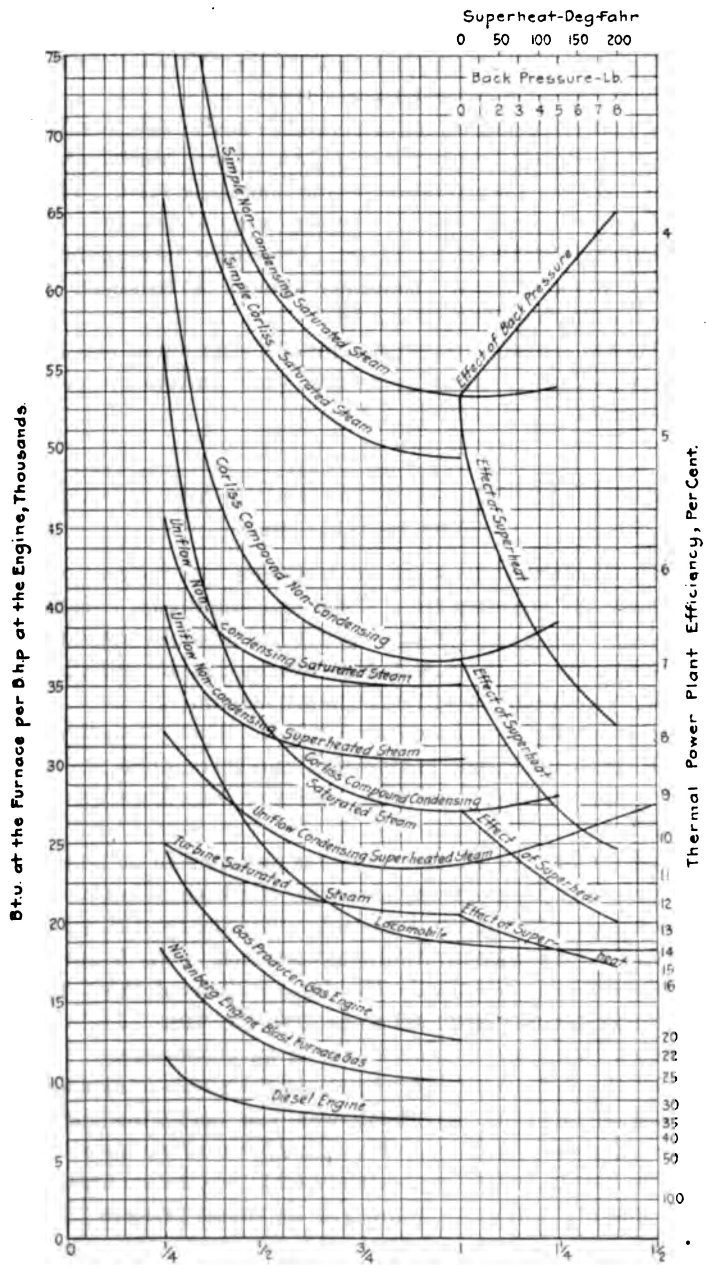

FIG. 13 RELATIVE EFFICIENCY AND HEAT CONSUMPTION OF VARIOUS POWER INSTALLATIONS

one-fourth load to 235 deg. at one and one-fourth load, and ordinary intermediate superheat. Since the chart was drawn upon the assumption that the boiler and grate efficiency is 70 per cent, for any other efficiency the value will be

$$\frac{\text{B.t.u.} \times 0.70}{\text{Boiler Efficiency}} = \text{Actual B.t.u.}$$

OPERATION AND MANAGEMENT

74 To obtain the greatest efficiency possible in a power plant is not only a matter of design, but a good deal a matter of proper operation and management. Most power plants could make large savings in this way, and there are exceptionally few where some improvements could not be made without spending a dollar for new equipment. To obtain full benefit from the equipment on hand, the men operating it must be made more efficient, and this can be done by:

a Education
b Strict supervision
c Pay according to merits. (Bonus systems)
d Competition and advancement
e Welfare.

75 Education, while the most important, is also the most neglected factor in economic operation of a power plant. This is especially true in the boiler room where the waste is greatest. Skilled labor should be employed. Ordinarily, anyone is hired who will work for low wages and maintain the steam pressure. Every boiler room should have a man with authority to give orders and with sufficient personality to have his orders respected. This man should predetermine, by the proper use of various instruments, the most economical ways of burning coal and instruct the firemen as to the best methods, and then insist that these methods be carried out.

76 License laws and examinations would conduce to better education of operating engineers, but in 60 per cent of the United States, by population, there is no license law, and the other 40 per cent consider only ability to operate a plant safely — seldom economic operating ability. It is declared unconstitutional to examine for efficiency. Since the welfare of our descendants and the ability of the American nation to be a world power and commercial leader depend upon an ample coal supply, if laws for its conservation do not exist they should be made.

77 License laws should be more stringent, examinations more complete, licenses graded and issued for intended plants. Since fuel-loss prevention is a national problem, efforts should be made to obtain a Government standard license law all over the United States, and members of examining boards should be civil-service employees of the United States Government, thus eliminating politics.

78 Jointly with extending the license laws and making examinations more strict, opportunity should be offered to men intending to become operating engineers to acquire the necessary knowledge. While the practical part can be learned in the power plant directly, the fundamentals of theory which are all important to the up-to-date operating engineer are difficult to obtain properly. Coöperation of educational institutions is needed by the forming of classes at the various universities and other schools to give courses on economic operation of power plants. Instruction should be given by correspondence to those unable to attend in person.

79 This year, Washington University of St. Louis made the initial step in this direction by starting two night courses especially adapted to the operating engineer. The first course is of a classroom nature and covers elementary thermodynamics; selection and operation of boilers; draft; furnace design and operation; flue-gas analysis; economizers; selection and operation of simple and compound steam engines, steam turbines, gas and oil engines; condensers; power-plant records and management. The second deals with commercial testing of power-plant machinery and includes gas and coal analysis, boiler tests, efficiency and economy tests of gas engines, pump tests, condenser tests, lubrication and oils. This last course is thoroughly practical; the men perform the various experiments in the university laboratory and thus become proficient and interested enough to perform them in their own power plants.

80 The author was in position to follow closely the agitation for these two courses and is thus able to speak authoritatively of the great interest they inspired among St. Louis engineers, which proves the willingness on the part of the men to learn if opportunity is offered to them and their natural bashfulness is overcome.

81 These courses should be given in all cities by men of thorough technical education and prominently engaged in the practical engineering field. They should also be given coöperatively between the different schools and be based upon one standard with the inten-

tion that in the future every applicant for an engineer's license will either produce a certificate of having attended such a course, or be able to stand an equivalent examination.

82 Opportunity should be offered the fireman also to learn his trade under competent instruction, and he should be required to stand an examination under practical conditions.

83 The effect that proper training of operating men would have on general power-plant conditions and economy cannot be estimated, but it would be of immense value to the owner, mechanical engineer, and also to the men themselves; in fact to the whole community.

SMOKE PREVENTION

84 There has been considerable agitation against smoke emission, and in all important cities smoke-prevention departments have been formed and laws made against smoke emission. The objections to smoke, as stated, are "public nuisance" and "fuel loss." The methods employed by smoke departments are effective in preventing the first, but the value of some of them as to the second is very doubtful. It is well known that the chief loss in a boiler plant is due to excess air (low CO_2) and not due to incomplete combustion, of which the result might be smoke.

85 To prevent smoke effectively, it is necessary to center attention on the quantity of air used to burn the coal and keeping it down to a minimum, proper furnace temperature and possibilities for well mixing the combustible gases and the air. At present (with exceptions) most attention is paid to smoke and smoke alone, without paying the attention to excess-air loss that it deserves. As a result many plants are wasting more fuel since operating smokelessly than before, because of using more excess air.

86 Smoke-prevention departments could greatly aid in preventing national fuel waste and some of them are doing so, but where they are not it can be laid to:

- *a* Political influence and interference
- *b* Lack of funds to employ expert inspectors
- *c* Employment of methods having no scientific basis and lack of effort to educate the engineer and firemen, while at the same time more force is employed than is good for the cause
- *d* Lack of confidence on the part of the power-plant owner because of all of the above.

87 Exceedingly few firemen, not even 3 per cent, know the simplest rudiments of combustion. Many who are considered good firemen persist in operating their furnaces so that at least 20 per cent of the fuel is wasted; at the same time claiming that it is the better and proper way. To overcome such conditions they must be educated.

88 It would be a great help if more of the simplest literature could be obtained, describing the economical process of firing, and this written in an interesting way. It is not meant that there is no such literature, for an immense amount of good has been done by *Power*, *Practical Engineer*, and other journals of this character, but the man who needs it most does not get it, and it would pay to bring it to him.

89 The U. S. Bureau of Mines would increase its usefulness by including in its excellent editions on fuel technology from time to time, bulletins of a thoroughly practical nature especially adapted to the average boiler-room man. These bulletins should be written by men who know the fireman and who have been through the mill themselves. After such bulletins are ready they should not be kept until those who need them write for them, but they should be given a forced circulation with the aid of boiler-insurance companies, smoke-inspection departments, business-men's leagues, etc.

CARE FOR THE HUMAN ELEMENT

90 In larger power plants, reading rooms for men waiting for the change in their shift, with some good literature, will cause the men to come, instead of ten minutes before work time, an hour or more. Also good engineering literature should be kept on file for employees to take home. Meetings should be arranged between employees and foremen at regular intervals for discussion and instruction upon the special problems coming up, new work, and better and more efficient operation. Lectures upon different subjects pertaining to efficient power-plant operation should be given, and the men kept in close touch with not only the outside but also the inside details of their work. As a result they will be more interested in their work, eager for learning and, in general, better men, by whom the power plant will be operated much more economically.

91 In smaller plants, typewritten instructions and discussions should be edited at regular intervals and given to the men to be studied.

92 The proper amount of work and the most economical way of getting the results for each man should be predetermined. He

should then be given his instructions, and it should be seen that he carries them out. Absolute strictness and discipline should be maintained, and, at the same time, if the men do the proper amount of work, in the proper way, they should receive the right compensation for it. Good men will seldom work for low wages.

93 A good power-plant manager should have the welfare of his employees at heart, yet usually not even proper washing facilities are provided and the men are allowed to live under unhealthy conditions, with the result that they are not vigorous enough to give efficient service.

94 Proper treatment of men and just compensation for service rendered by them is necessary, and of much greater importance than ordinarily realized. Blindness on the part of the employers to the fact that workingmen must be treated as human beings has done an inestimable amount of harm to the industry, since organizations are springing up that directly counteract efficient performance of labor.

95 Competition is the best stimulus to the attainment of more and better work, and therefore should be developed wherever possible. One power plant should be compared with another. The engineers of each should be put in competition with one another whenever possible, and so also the firemen, by posting their fuel loss, obtained from average CO_2 readings, upon a blackboard in the boiler room. When practical, a bonus system should be provided and men given some compensation for the extra efforts made.

96 Coöperation among power-plant owners and the formation of supervising engineering staffs to take care of a number of plants would be of great value, as in this way they all would obtain expert supervision that each alone, ordinarily, could not afford.

97 Replacement of small, uneconomical power plants by large, efficient central stations will also promote conservation of the nation's coal supply.

98 It was the author's aim to outline in this paper the existing weak points and expose the wasteful conditions, whose prevention is not only our duty, but also for our immediate benefit. A nation-wide educational campaign should be started and persistently carried out until results are obtained. It is suggested that The American Society of Mechanical Engineers start the movement by forming a "Committee for the Prevention of Fuel Losses and General Betterment of Power-Plant Conditions," which would devise ways and means, by which the best results could be obtained.

DISCUSSION

Joseph Harrington expressed his general approval of the author's paper, and contended that the question of increasing the operating efficiency of a power plant was not only a question of apparatus, of equipment, or of instruments, but was a question ultimately of the human element. The man in the office must have an incentive for better power-plant operation in the form of a substantial financial return on the money invested for instruments, and the man in the boiler room also must have an incentive for obtaining the maximum efficiency and utilizing the instruments provided for his aid. It is the man in the boiler room who presents the more difficult problem: how to supply an incentive for him is an unsolved difficulty. Some schemes more or less subject to criticism have been worked out. The human element, the man, must be studied, of course, and the scheme must be adapted to all the surroundings. It is one requiring tact, judgment, and perseverance. Something should be done for the compulsory education of the fireman, and a bonus system of some kind may supply the incentive which is now lacking.

Walter N. Polakov took strong exception to the point of view expressed by Mr. Harrington, and said that the trouble was not with the firemen, but with the owners of the plant. How can the fireman do good work if he does not have a competent leader? If the manager or owner does not appreciate the importance of studying the processes involved, if he does not take into consideration the psychology of his employee, if he does not care to make a small investment for instruments, the fireman cannot be blamed. If, as in a case cited, an improvement of 40 per cent in economy was made by the change of a head fireman, the blame for the former wasteful condition lay not so much with the fireman as with the management that tolerated such a condition. In other words, if there is to be an educational campaign with the object of protecting our national resources, primarily fuel, the beginning must be made at the Boards of Directors, not in the fireroom — at the money end and not with the coal passer. A proper bonus system is the best solution of the problem of properly stimulating the firemen to work intelligently.

He could not subscribe to Mr. Azbe's statement that the replacement of small plants by large central stations will be beneficial to everyone. In small plants, exhaust steam may be used for heating and industrial purposes. From an unsophisticated standpoint the question is: Is it better to pay 10 or 12 cents per kw-hr. to a central station, or to develop this power in your own isolated plant at a cost of 2½ cents per kw-hr.?

Edward A. Uehling (written). Mr. Azbe's paper covers the whole steam-power plant, and he makes many valuable suggestions as to how the heat now wasted could be saved. I shall confine my specific remarks to the operation of the steam generators, in which at least 50 per cent of the preventable heat losses occur, especially in the type of plants to which Mr. Azbe most particularly refers.

To obtain highest efficiency from a boiler three things are necessary, these in the order of their importance being (1) efficient combustion of the fuel, (2) efficient absorption of the heat generated by combustion, and (3) efficient rate of driving.

The difficulty in maintaining boiler efficiency is that so many continually changing variables are involved in its operation that fixed adjustments are out of the question. To maintain maximum boiler efficiency the fireman must have before him the information necessary to enable him to make the required adjustments intelligently, as well as the facilities necessary to make them. The draft must be varied to burn the coal necessary to produce the steam required. The steam gage tells him when to increase or decrease his draft. The thickness of the fire must be adjusted to the draft, so that complete combustion takes place with the minimum excess of air. Unless some means are provided by which the fireman can tell whether the relation between the draft and thickness of fire is right, he cannot know with any degree of certainty whether his fire is too thick or too thin for economic combustion at the changed rate of driving. A CO_2 indicator at the boiler front tells him at a glance whether this adjustment is right, and what to do to make it so. Although the necessary draft adjustment to the steam demand and the fire adjustment to the combustion-efficiency demand can and must be made regardless of what the draft gage may indicate, it is none the less of great value to the fireman in making adjustments. He should have before him not only the boiler draft but also the furnace draft, the latter to indicate the condition of his firebed and the former to indicate the rate of driving. Where positive information is required regarding the rate of driving, steam meters also must be installed.

The temperature of the escaping gas depends on four variables: (*a*) the condition of the boiler setting, including the baffling; (*b*) the condition of the heating surface; (*c*) the rate of driving, and (*d*) the percentage of CO_2 in the escaping gas. It is an index to absorption efficiency provided condition (*a*) is perfect and (*c*) and (*d*) are known. It is of no value to the fireman as a guide to combustion efficiency, because it depends on conditions beyond his control. A record of the temperature of the escaping gas considered by itself is also of questionable value to the operating engineer and may be entirely misleading, inasmuch as a low gas temperature may result from air in filtration as well as efficient absorption of heat by the boiler, and high temperature may be due to a high rate of driving, dirty heating surface, or broken-down baffling. To discriminate in the former case, we must know the percentage of CO_2, and to locate the cause of the latter we must know the boiler draft.

Although the flue-gas temperature by itself is of no value to the operating engineer as a control, it becomes of some value when considered in combination with the boiler draft, and is of the utmost value in connection with the percentage of CO_2, because these two factors determine the heat loss up the chimney.

Boiler efficiency can be correctly ascertained only by accurately measuring the heat input and heat output. This, however, is neither feasible nor desirable under operating conditions. A system of efficiency control on the heat input-output basis was organized by Mr. Polakov for the Penn Central Heat & Power Co. in their Warrior Ridge plant, and described by him in a paper read before the Society in December, 1913. This system of control approached scientific accuracy and gave excellent results, at least while under Mr. Polakov's able management, but it is too elaborate and cumbersome for general application, and to my knowledge has not been duplicated at any other plant.

Automatic coal weighers alone are of no value as a control for boiler operation, though they may be of some value as a check on the shippers' weights. Water meters are of more value because a more or less approximate estimate of the coal burned can be made at convenient intervals, perhaps monthly, and knowing the weight of water evaporated, a calculation of steam produced per pound of coal burned can be made and a rough idea of the combined efficiency of all the boilers obtained.

Diagnosis must precede the selection and application of the remedy, or a cure cannot be effected with any likelihood of success. Every boiler must be treated individually. The gas must be analyzed, temperatures must be measured and draft conditions observed continuously. To do this the necessary instruments must be supplied by the manager and intelligently used by the operating personnel, and if so used the highest efficiency which the conditions of the plant permit can be attained, and in no other way is it attainable. It has been accomplished many times by thorough combustion experts with the aid of an Orsat apparatus and a portable draft gage and pyrometer. This is very well and good. The trouble is the high efficiency established by the expert will not stay put. He has scarcely left the plant when it will begin to drop off, and in the course of a few weeks or months at most it will be down nearly, if not quite, to where he found it. High efficiency cannot be continuously maintained without instruments that will guide the fireman as to what to do, and will indicate properly the effect of what he did, and will autographically record the performance of both fireman and boiler as a control for the operating engineer.

Autographic records to be of the greatest benefit must be regularly and thoughtfully scrutinized and co-related, and the information they contain must be promptly acted upon, whether to bestow a praise, administer a rebuke, or correct a shortcoming in the operator or plant. The moral effect of the autographic record is very great if wisely used. Scientific apparatus cannot serve their purpose unless they are kept in continuous operating condition. They will not prove a paying investment if they are operated perfunctorily. They should receive the same regular and conscientious attention that must be given to the machines and apparatus which are vital to the operation of the plant.

R. J. S. Pigott spoke about the possibility of high CO_2 content, and said that even with an underfeed stoker it was a fact that the highest efficiency could not be obtained by operating at the exact proportion of air to coal and gas. His personal experience was that there is no advantage in a higher content than 12 or 13 per cent.

He considered that the deterioration of the firebrick furnace wall was not due to high temperature but to the erosion effect above the firebed. Commenting on a statement in Par. 15 of the paper, that exceedingly high furnace temperatures have no real advantages and several disadvantages, he said that in his opinion the temperature of the furnace had little to do with the advantages claimed for it by the author. It is an advantage to have the heat-absorbing surface directly over the fire, because heat absorption by radiation is enormously rapid compared to that by conduction. It is attained at no loss of efficiency due to dirt on the tubes.

Regarding another statement in Par. 42: "Many power plants that have economical prime movers have auxiliaries that are so uneconomical that the efficiency of the whole plant is greatly reduced," he said that in many cases the economy of the auxiliary has almost nothing to do with the thermal economy of the plant until the maximum possible amount of heat is absorbed in the feedwater heater. The use of an economizer will depend upon the other factors to be considered. The question should be settled on the actual working conditions of the plant, and not on a set of assumed conditions with one load only.

William B. Jackson. Referring to Mr. Azbe's paper, it is always worth while to have placed before us what it is possible to do in a power plant, but it is also essential that we do not permit ourselves to confuse the ideal with the practical in the operation of such plants. It is true that there are cases where it is even practical to so change the operation of small power plants as to make the difference between complete failure and superlative success, and I have had the pleasure of having a hand in some such cases. But there can be no doubt that owners of small plants frequently receive very exaggerated ideas of what can be done to improve the net earnings of their properties, by having brought to their attention possibilities which are ideal but not practical. A person can go into almost any small power plant and show the manager how he can obtain apparently perfectly astounding operating economies, but there is always the question to be answered whether the introduction of the improvements will really prove a net gain in the long run, when taking all things into consideration, including the human element. Probably the most difficult problem that confronts an engineer, in advising as to improvements that should be made in the operating conditions of a power plant, is to differentiate between the many improvements that apparently can be made but which will really not show a net gain for the property and those which will show a net gain.

E. N. Trump stated that he was interested in the question of the economizer, which he considered paid exceptionally well in his plant. He could not subscribe to Mr. Pigott's statement that the economy of the auxiliaries makes no difference until there is more heat in the exhaust steam than is needed for feedwater heating. In this case the water comes to the economizer so warm that the furnace gases do not give up all the heat possible otherwise. In his plant he found it possible to obtain 90 per cent efficiency in summer and 85 per cent in winter. The plant has a steady load. There are five 2300-h.p. boilers. The gas temperatures are about 135 to 150 deg. cent. The CO_2 runs about 12 and 13 per cent. The steam from the boilers is superheated. The exhaust steam from the turbines driving the stoker fans and feedwater pumps is all condensed in the feedwater just before it enters the economizer.

John Hunter thought that the CO_2 recording machine would be more useful in the boiler room if it did not take a skilled mechanic to keep it in operation, and asked the author if he referred in his paper to some special machine. He questioned Mr. Trump's 90 per cent efficiency. [Mr. Pigott explained that the boiler-plant efficiency was meant, and not boiler efficiency.]

H. R. Cobleigh commended the paper generally and emphasized certain points. Managers, and particularly directors, of power plants should be impressed with the saving possible without adding to or changing their equipment. If they would operate their plants as they now exist more intelligently, they could save more than they would otherwise by putting in all the latest and most modern apparatus and not operating it correctly.

He did not feel that the author's method of educating the fireman would be very effective. The only way of teaching improved power-plant operation is in the power plant itself. It cannot be taught in a class room, much less by correspondence. He wished particularly to emphasize the importance of something like a bonus system which makes a man automatically desire to do better work and keep it up.

P. C. Idell thought that one way to make the paper bring results, would be to have an appendix giving summaries from different lines of business, showing how preventable waste has been dealt with and the waste overcome.

William F. Schaller, in line with Mr. Idell's remarks, cited a case where in the first year a saving of 47 per cent on an investment of $22,000 was made by purchasing engines to operate with heating boilers adapted to high pressure, and modifying central-station service to form an auxiliary source of current.

A. T. Baldwin suggested that the conclusion reached by the author be brought to the attention of the plant owner. Having had the pleasure of meeting both the operating engineer and the plant owner, it had been his experience that the engineers were already alive to the importance of efficiency, and that such questions were discussed by them in national and local meetings; the plant owner, however, was not easily converted.

Albert A. Cary corroborated Mr. Pigott's statement, saying the temperature in the furnace had but a secondary bearing upon the slagging effect on furnace walls. If the ash in contact with a furnace lining is composed of certain chemical compounds which will unite with the constituents found in the firebrick so as to form eutectic mixtures, the fusing temperature of both the ash and the firebrick is lowered below the normal fusing temperature of either one and then the lining is apt to be eroded and a slag is formed.

If, on the other hand, the ash contains sufficient amounts of fluxing compounds (such as Fe_2O_3, CaO, or MgO) to react strongly upon its own refractory compounds (SiO_2 or Al_2O_3), such ash will fuse at a relatively low temperature and become a more or less hot pasty or sticky mixture which will adhere to the furnace walls. Such a deposit continues to grow, becoming hard when cooled, and is detached from the lining with considerable difficulty.

Another form of less troublesome clinker is occasionally found in boiler furnaces (generally of a light-gray color), in which the fluxing compounds do not exist in the ash in such excessive percentages as they do in the ash forming the hard, troublesome clinker just described. The result is that these clinker formations do not fuse together or attach themselves to the furnace lining tenaciously, and this weak, more or less friable clinker gives but comparatively little trouble.

If means are provided to handle the firebed in such a manner as to prevent the ash from coming in contact with the linings (to any great extent), little or no slagging effect occurs on the furnace wall, even with the hottest fires.

The place for the ash resulting from the process of combustion is right down upon the top surface of the grate. There should always be a thin layer of ash there, to protect the grates from the hot firebed above, and further means should be provided to keep the thickness of this ashbed down to a thin layer, so as not to prevent the free passage of an ample amount of air to the firebed above. With the cool entering air constantly passing through this ashbed the temperature of the greater part of the ash is kept below its fusion temperature, and under such conditions the temperature of the furnace will have little or nothing to do with the formation of clinkers.

Personal experience with furnaces operated in this manner for over a year, has proved that with furnace temperatures running from 2600 deg. to 2800 deg. and with fires operated continuously for twenty-four hours per day, no troublesome clinkers have been formed, notwithstanding the continuous combustion of over 42 lb. of eastern bituminous coal per square foot of grate per hour; and the doing away with slagging effect on the furnace linings is shown in the fact that the original linings that were put in when the stoker was installed have not been replaced, but have been kept in good, serviceable condition by only slight and occasional patching.

In line with Mr. Idell's remarks, he told of experiences in Cleveland, fifteen years ago, where a considerable number of plants were equipped with various improved furnaces and stokers to suppress the smoke nuisance.

A record was kept at these plants of cost of fuel used for operation before and after these improvements had been made, showing in all cases that the means used to reduce smoke emission had resulted in a material saving which gave good returns upon the investments made. A summary of these results was then issued by the Smoke Commission to the architects of that city and was also sent to power-plant owners and others upon request.

Owing to legal and other difficulties encountered by the Commission, this good work could not be carried on as vigorously as they earnestly endeavored to push it, and although the limited results they succeeded in obtaining were very encouraging, the general results were very discouraging.

W. S. Gould (written). In so many discussions and papers on this subject, I am impressed by the fact that the most essential point seems to be overlooked: namely, the question as to the character and quality of the fuel used. It seems to me utterly useless to discuss the dollars-and-cents losses in any power-plant tests when nothing is known or said regarding the particular fuel used. In many cases there are greater preventable losses found in the character of the coal used than in the operating conditions of the plant.

In his paper the author evidently refers only to Illinois coals, although he bases his figures as to losses on the total fuel consumption of the whole country, using the price of Illinois coal at St. Louis as the "average cost of coal." Coal in the New York district, in normal times, costs from $3.00 to $3.50; and in some parts of New England, $5.00 or more.

Coal is too often considered simply as "coal," and assumed to be of some average quality, known to someone or to everyone. This is only too obviously in error. Illinois coal varies greatly, even in the same district and from the same mine. So it is with coal from every district. Illinois coal varies greatly both in character and quality from coal mined in the Eastern States, and coal in the Eastern States varies with every hole in the ground. This fact is susceptible of easy proof.

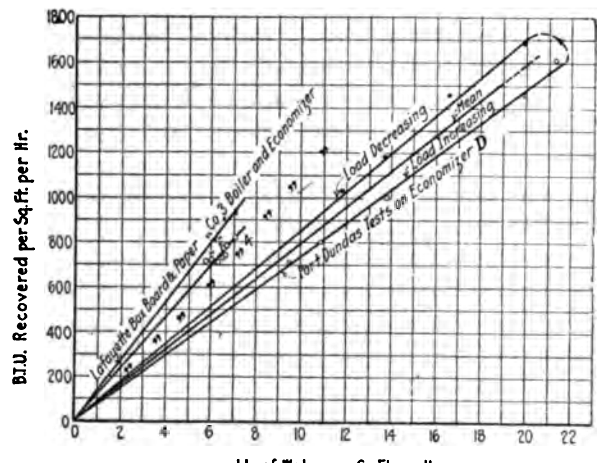

Fig. 14 Relation of Heat Recovery of an Economizer to Rate of Water Flow

Geo. H. Gibson (written). The author states that "it is difficult to reason out just what effect load variations have upon an economizer as a heat absorber." However, in a given installation, if the heat absorption at any one load be known, the heat recovery at other loads will be approximately in a direct proportion to the rate of water flow, as indicated by the chart of Fig. 14, which gives the results of three different tests with variable loads. From the Port Dundas test it would appear that the heat recovery while the load is increasing is somewhat less than while the load is decreasing. This is due to heat storage in the large mass of water in the economizer. If the heat recovery of the economizer for a given steady load is known, however, it is only necessary to draw a straight line through this point and the point for zero load, in order to determine the heat recovery at every other load.

One of the graphs in the paper purports to show the relation between velocity of gas through the economizer and heat absorption per square foot per hour per degree difference of temperature. However, the coefficient of heat absorption varies with the temperature of the gases as well as with the velocity. The relations between rate of gas flow, temperature of gases and average coefficient of transmission are shown in the accompanying chart, Fig. 15, which I have compiled from tests upon a large number of commercial economizers in various conditions of actual service. The temperatures

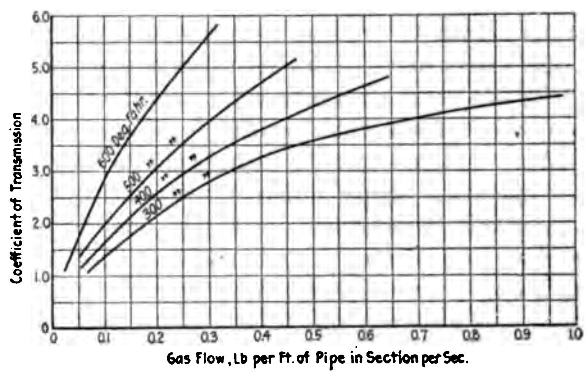

FIG. 15 RELATIONS BETWEEN RATE OF GAS FLOW THROUGH ECONOMIZERS, TEMPERATURE OF GASES, AND COEFFICIENT OF HEAT TRANSMISSION

marked upon the graphs are the mean temperatures of the gases, that is, the temperature of the gases entering the economizer plus the temperature of the gases leaving the economizer, divided by two. The rate of gas flow is stated in pounds of gases per foot of pipe in a section per second, that is, if there are ten pipes in a section of the economizer and each pipe is 10 ft. long, there will be 100 ft. of pipe per section, and the total gas flow per second would be divided by 100 to obtain the quantity set off on the horizontal axis.

THE AUTHOR. In connection with Mr. Pigott's opinion in regard to maximum CO_2 desired, I wish to say that this is altogether a matter of type installation. With many installations it is not desired to go over 10 per cent; with others again 16 per cent and 17 per cent can be obtained without serious losses due to incomplete combustion.

The most important factors in this problem are proper facilities for gas mixing and proper air distribution over the grate. We also must not forget that on one hand at high CO_2 percentages CO might not be the only combustible gas escaping, and on the other hand that a given percentage of CO at high CO_2 percentages represents a great deal smaller loss than with the low CO_2.

Supplementing Messrs. Harrington and Polakov's excellent statements, I wish to say that while the manager is the one who is mostly responsible for the present condition, nevertheless the operating engineer shares also the blame to a very great extent, and the chief reason for the bad condition is that efficiency in power plants is taken, generally, too lightly, and this even by a great many professional men who really should be the leaders in agitation for better conditions.

I do not agree with Mr. Hunter's statement on CO_2 recorders, for the reason that I have had five CO_2 recorders of three different types in operation for two years, obtaining with them the most satisfactory results in our two St. Louis Anheuser-Busch plants. One man—not a college graduate — takes care of, changes charts and keeps in constant proper operation eighty different instruments, such as CO_2 recorders, recording pyrometers, steam-flow meters, venturi and other water-flow meters, etc., and in addition finds time to help with testing work. Where CO_2 recorders or similar instruments are not successful, it is generally due to not taking time to learn their principles of operation, and to improper installation.

Fig. 6 of my paper was not drawn to show exactly the economizer heat transmission, but rather the approximate relation between heat flow and draft loss, and represents conditions in the economizer at about 300 deg. fahr. mean temperature difference. Mr. Gibson mentions that the rate of heat transmission varies with the temperature. I wish to say that it varies rather with the mass of the gas and the velocity, both of which are, of course, dependent upon the temperature. Heat transmission in economizers with a given gas velocity will also vary, due to the influence of such factors as cleanliness of outside and inside heating surface, water velocity, and distribution of gas over the heating surface.

Where the flue-gas temperature is below 500 deg. fahr. it will hardly pay to install an economizer, except in those cases where the cost of coal is very high. In ten years, however, coal will probably cost twice what it now does, and then economizers will be of great value. It would probably be better to eliminate the last pass and add an economizer.

With Mr. Gould's opinion that I overlooked the most important point, "the character and quantity of fuel used," I do not agree, for the reason that while the quality and cost of coal are prime factors in planning efficiency methods and efficient equipment, they are not necessarily essential factors, so far as my paper is concerned. Preventable power-plant losses are general over the whole country and where high- as well as low-grade coal is burned, and if it pays to prevent the losses in small plants burning low-grade and at the same time cheap coal, it certainly should pay in larger plants or with higher-grade and more costly coal.

No. 1554

THE IMPACT TUBE

By Sanford A. Moss, Lynn, Mass.
Member of the Society

The impact tube as a part of the pitot tube is well known as a means for measuring flow of fluids. When used as a separate instrument, however, it is a much more valuable apparatus, and has not received the attention it deserves. This paper deals with applications of the impact tube to a number of purposes not generally known, particularly relating to compressible fluids. The paper is based on laboratory work of the Steam Turbine Department of the Lynn works of the General Electric Co., Richard H. Rice, Mem. Am.Soc.M.E., chief engineer.

ADVANTAGES OF THE IMPACT TUBE OVER THE PITOT TUBE

2 The writer believes there is no reason for using the common form of pitot tube. This is a double-walled tube consisting of an impact tube pointed upstream surrounded by a chamber with openings for determining the static pressure. It has been known for years that the impact portion of this tube gives reliable results, regardless of its shape. On the other hand, the static holes give very indefinite results and the shape and sizes of all of the parts have influence on the static readings. Specific arrangements have been found by experiment which seem to give exact static pressure.

3 Static pressure is defined as the pressure determined by a hole in the wall of a surface parallel to the fluid flow. Now, in no case can reliance be placed upon readings with an impact tube, or any of the common forms of pitot tube, unless the instrument is inserted at a point where the stream is uniform. Ten to twenty diameters of straight pipe should precede the tube and about five diameters should succeed it. Hence, if we have a pitot tube which measures true static pressure, and a uniform jet, the static readings, at every point across the pipe section, are identically the same as would be given

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by a static hole in the pipe wall. Therefore, measurements made by use of a plain impact tube in conjunction with a static hole in the pipe wall will give results identical with those given by an accurate form of pitot tube.

4 It might be thought that an accurate form of pitot tube takes account of any slight variation there may be in static pressure from one side of a pipe to another when a traverse is made across the pipe. This, however, is not so. If there is any difference in the static pressure across the pipe section, the streams will not flow parallel to the pipe walls but at various angles. Parallel flow in a pipe is only possible when the static pressure is absolutely the same at all parts of a section perpendicular to the stream. Therefore, if there is variation in the static pressure the stream angles will be such that even an accurate form of pitot tube will give erroneous results. The pitot tube, therefore, offers no advantage over the combination of an impact tube and a static hole in the pipe wall. On the other hand, it is much easier to make a reliable static hole in the pipe wall than it is to make a reliable static hole in the pitot tube. The usual trouble with pitot-tube static openings is that they give too low a reading for static pressure. This increases the small differential reading between impact and static sides and indicates a quantity larger than actually exists.

5 All of the above refers to the common form of pitot tube, consisting of an impact opening and one or more static openings in the side walls immediately adjacent. Such a tube is used to determine the flow in a pipe by means of a traverse across the stream, readings being taken at each of a number of different points. There are several forms of nozzle plugs, special pitot tubes, etc., which provide a number of holes completely across a jet. Some of these are said to average the velocities and static pressures in an irregular jet, automatically determining the flow at a single reading. The writer has had no experience in the use of such instruments for accurate work and this paper does not discuss them.

LAW OF THE IMPACT TUBE

6 The law of the relation between velocity and the differential pressure (difference between the readings of an impact opening and a true static opening) is well known for the case of incompressible fluids or cases where the compressibility effect is negligible, such as air and steam at comparatively low velocities. This law is usually given in the form $V = \sqrt{2gh}$, where V is the velocity in feet per

second, and h is the height, in feet, of a column of the fluid in question having a pressure equal to the differential pressure.

7 The use of the impact tube is by no means restricted to cases where the above formula applies, however, and very high velocities with compressible fluids can be measured accurately if the complete law is used. This law is that *the impact tube perfectly converts velocity into pressure.* That is to say, the velocity of the fluid in front of the impact tube is exactly that which would be produced by theoretical adiabatic flow from a region where the absolute pressure is that shown by the impact tube, to a region where the absolute pressure is the static pressure. We may, therefore, use the theoretical thermodynamic formulæ for adiabatic flow through orifices deduced with the assumption that the initial velocity in the pipe preceding the orifice is practically zero. This law holds for steam, air and all gases, as well as for liquids.

8 The flow of a gas thus computed by the thermodynamic laws for adiabatic flow is somewhat different than if computed by the incompressible fluid law of Par. 6. This difference is negligible if the differential pressure is a small fraction of the absolute static pressure. The error is appreciable with air with a pressure of 1 lb. per sq. in. above atmosphere, with velocities of about 50 ft. per sec., and becomes more and more serious as velocity increases.

PROOF OF IMPACT-TUBE LAW

9 The law given in Par. 7 follows directly from the general law that *impact pressure is constant at all points as we proceed up the stream along an orifice fed by a very large pipe.* It has been known for some time that this law holds for the case of liquids, and it may be extended to the case of high velocities and compressible fluids. Demonstration of this law is readily made by means of apparatus shown in Fig. 1. A stationary impact tube is clamped at a convenient point in the jet from a convergent orifice preceded by a large pipe. An impact tube, moved by hand, is arranged to be pushed up inside of the orifice. Differential pressure between the two tubes is shown by a water U-tube. It will be found that the differential pressure is zero. The law is certainly accurate to within about $\frac{1}{2}$ of 1 per cent of the orifice impact pressure, and probably is absolutely accurate. Complete sets of readings and detailed descriptions exist for this test, as well as for many others referred to later, but must be omitted for lack of space.

INITIAL VELOCITY HEAD

10 The apparatus above mentioned shows, when the pipe preceding the orifice is more than about three times the orifice diameter, that the jet issuing from the orifice into the atmosphere has a constant impact pressure, and hence a constant velocity, at all points over its area except very close to the edges.

11 The velocity in a long straight pipe preceding an orifice is not at all constant but is proportional to the ordinates of an ellipse, being a maximum in the center and gradually decreasing toward the sides of the pipe. When this pipe velocity is not negligible compared

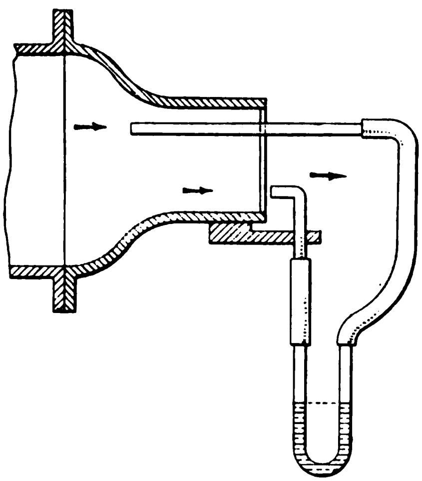


FIG. 1 CONSTANCY OF IMPACT PRESSURE OF AN ORIFICE

with the spouting velocity, the experiment of Fig. 1 shows that the static pressure in the pipe preceding the orifice plus the average velocity head at this point is equal to the impact pressure in the jet issuing from the orifice. The sum of the initial static pressure and the average velocity head is given by an impact tube in the pipe preceding the orifice placed at such a radius as to give the average velocity head. If the pipe preceding the orifice is quite large, the point where the velocity head has the average value is at about 0.7 of the radius. Hence, a water U-tube arranged as in Fig. 2 reads zero. For the same reason, the water U-tube in the experiment of Fig. 3 reads the average velocity head in the pipe preceding the orifice.

CONVERGENT-DIVERGENT ORIFICES AND VELOCITIES GREATER THAN VELOCITY OF SOUND

12 As is well known, for a compressible fluid where the final pressure is greater than about half the initial pressure, an orifice must be wholly convergent just as in the case of incompressible fluids. In such cases, the velocity of the jet issuing from the orifice is less than the velocity of sound. The above impact-tube laws, as well as all of the work with impact tubes here discussed, apply only to such cases. With a gas, when the final pressure is less than about half the initial pressure, the orifice must be first convergent and then

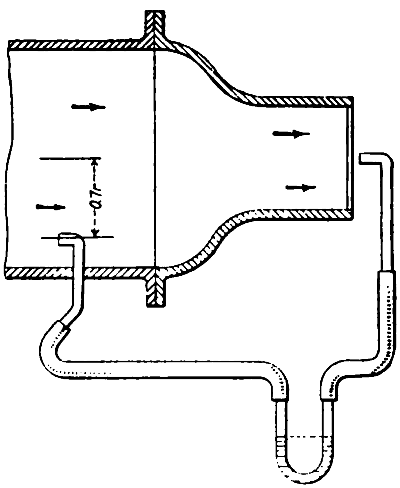


FIG. 2 EQUALITY OF IMPACT PRESSURE IN PIPE AND JET

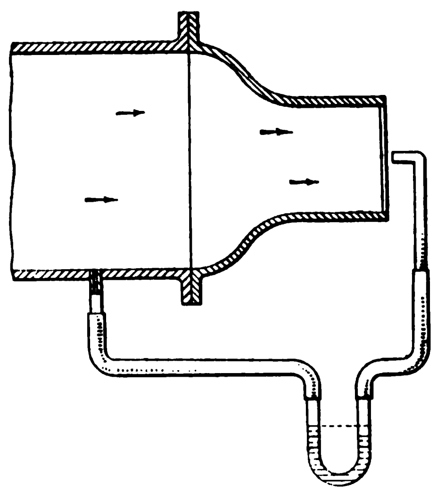


FIG. 3 VELOCITY HEAD SHOWN BY DIFFERENCE BETWEEN PIPE STATIC AND JET IMPACT PRESSURES

divergent, for well-known reasons which will not be discussed here. The writer has not carefully investigated impact tubes in such cases. There is an indication that, under these conditions, the impact tube does not convert velocity into pressure with an efficiency of 100 per cent. It can be said, however, that with a final pressure less than half the initial pressure, and with a wholly convergent nozzle (a shape which does not give full theoretical velocity), the impact pressure in the jet close to the end of the orifice is equal to the initial pressure. In this case, it is known that the static pressure of the jet is about 52 per cent of the initial pressure for air, and 58 per cent for steam, regardless of the pressure of the region into which the jet discharges, and that the jet velocity is the velocity of sound. The jet itself forms stream lines, creating an extension of the orifice, and representing a divergent portion where pressure decreases to the pressure of the surrounding region and where velocity increases from velocity of sound to a somewhat greater value (less, however, than the value obtained with a properly shaped orifice).

DEDUCTION OF THE IMPACT TUBE LAW FOR GASES

13 We must remember that a gas in a closed envelope consists of a number of molecules darting hither and thither at high velocities, and that a gas flowing in a pipe consists of molecules with velocities such that the average velocity down stream has a definite value, the velocities across the stream cancelling so that the average is zero. Hence, if gas is flowing from a large vessel through a properly shaped orifice into a region of lower pressure, the molecules in the jet issuing from the orifice are really in the same condition as those in the vessel, except that there is a general net velocity which we call the velocity of the jet. The net velocity is at once destroyed, however, if anything is inserted in the jet, so that the molecules immediately in front of the impact tube in a discharge jet are in exactly the same condition as those in the large vessel from which the gas issues. Hence, the impact tube in the jet must measure the same pressure as would a static hole in the pipe wall of the large vessel.

IMPACT TEMPERATURE

14 A similar situation occurs if a thermometer is placed in a high-velocity jet in an attempt to measure its temperature. The molecules surrounding the thermometer are immediately restored to the same condition as those in the large vessel from which they originally proceeded. Hence, as can be demonstrated by experiment, a thermometer inserted in a high-velocity jet issuing from an orifice will give nearly the same reading as a thermometer in the large vessel from which the jet discharged. A very interesting form of the experiment can be performed with a steam orifice discharging into the atmosphere, with an initial pressure of about 25 lb. per sq. in. absolute and with the initial steam slightly wet. The laws of adiabatic expansion of steam show that the steam in the jet issuing from the orifice will also be wet. Hence the jet will actually be at the temperature of wet steam at atmospheric pressure, which is 212

deg. fahr. Nevertheless, a thermometer inserted in this jet will indicate a temperature which is considerably higher than 212 deg. fahr.

15 A gas discharged from an orifice, from the ordinary point of view, has a very low temperature resulting from adiabatic expansion. Actually, however, this temperature is only relative to a thermometer moving with the velocity of the jet. The individual molecules have the same average kinetic energy and hence the same temperature as they originally had.

16 Temperature measurements have been made and published which show temperatures in a high velocity jet of air much above the theoretical adiabatic temperature of the jet. It was alleged that these experiments showed that the velocity of such a jet was not really the theoretical velocity due to adiabatic expansion, and that it was not possible to expand a gas in a nozzle so as to obtain such a velocity. This conclusion is incorrect, however, and with a properly shaped orifice or nozzle, nearly the full theoretical velocity due to adiabatic expansion can be attained with air or any gas, just as with steam in a steam turbine. If the energy is extracted from the jet by means of a turbine wheel or the like, the exhaust will show low temperature. If, however, no energy is extracted but if the thermometer is inserted directly in the high-velocity jet, nearly initial temperature will be shown.

17 The temperature of the walls of a pipe, or orifice end, in which a high-velocity jet is flowing, is another interesting matter. There is reason to believe, if the tube is perfectly smooth from a molecular point of view, that the temperature of the wall will be the theoretical adiabatic temperature. If, however, the wall is rough, as is usually the case, a temperature intermediate between the initial temperature and the adiabatic temperature will be read, depending upon the roughness.

18 A similar situation probably arises in the case of a thermometer inserted in the jet. If this thermometer has a perfectly square end which is the only part exposed to the direct jet, it will probably read initial temperature. If the thermometer has a rounded end and some side walls exposed to the stream, a value a little lower than initial temperature may be read. If the tip of the thermometer were shielded so that only the thermometer walls parallel to the jet were exposed, a value nearer the adiabatic temperature would be read. Intermediate arrangements would give temperatures between the adiabatic and initial values.

19 It is to be noted that the temperature shown by a thermometer which had only a square end exposed to the jet would probably be initial temperature rather than the temperature which would result from destruction of the initial velocity at the lower pressure. This latter temperature is that obtained in a porous-plug experiment or any other case of expansion with constant total heat, and is cooler than the initial temperature by the amount of the Joule-Thomson effect. Experimental measurements of this have baffled physicists for years. Possibly such experiments may be made by using a thermo-couple in a high-velocity jet, one side exposed to the impact temperature, and the other shielded from the impact temperature and exposed only to gas whose velocity has been destroyed.

20 In the Bradley and Hale experiments on temperature of air after great expansion,[1] thermometers directly in the high-velocity jet showed a lower temperature than those a little further away where the velocity had been destroyed. This is probably due to the fact that the orifice which was demanded by the pressure ratios involved should have been first converging and then diverging. The actual orifice was a straight passage succeeded by a sudden enlargement to a chamber where the thermometer was inserted. This probably formed an orifice mouth with area much greater than necessary, giving expansion to a pressure much lower than atmosphere. The adiabatic temperature with such an excessive expansion would be very much lower than with expansion to atmospheric pressure. It could easily be lower than the temperature of the jet after it had gone up to atmospheric pressure and had had its velocity destroyed.

IMPACT-TUBE ARRANGEMENT

21 The exact shape of the impact tube is not important. It is only necessary that there be a pipe or tube facing squarely up stream in a jet undisturbed except by the impact-tube end. The impact-tube end, therefore, should have a straight portion parallel to the jet about 10 times the length of the impact-tube opening, before the tube bends away from the jet. The open end of the tube should be slightly rounded on both inner and outer edges and should be smooth. For ordinary work, the tube is $\frac{1}{4}$ in. to 1 in. in diameter. For work with small pipes, and for special exploration work, the tube may be of any size. The writer has obtained satisfactory results with tubes as small as $\frac{1}{32}$ in. outside diameter.

[1] Physical Review, 1909, v. 29, p. 258.

22 The impact tube must face squarely upstream. There is no necessity for exact measurement in this connection, however, and estimation by eye is accurate enough. Readings in a constant jet with impact tube rotated show that angles of several degrees on either side of the exact upstream position give no effect on the readings. As the tube is rotated further, however, the readings fall off rapidly. At an angle shortly before the tube becomes perpendicular to the stream and while it is still partly facing upstream, the differential pressure becomes zero and the impact tube reads the static pressure. When the tube is perpendicular to the stream the differential reading becomes negative.

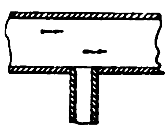


Fig. 4 Single Static Hole in Pipe Wall

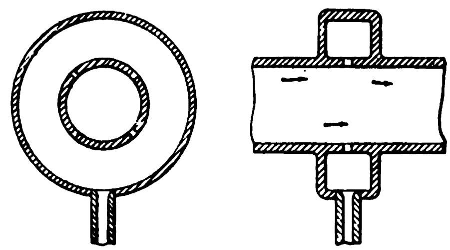


Fig. 5 Ring of Static Holes in Pipe Wall

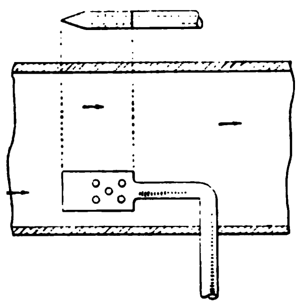
23 In order to be certain that a tube inside of a pipe is properly facing upstream, it is desirable to have the outer end of the tube bent in the same plane as the inner end, as shown in Fig. 8.

STATIC HOLES IN A PIPE WALL

24 Figs. 4, 5, 6 and 7 show different methods of obtaining static pressure. Fig. 4 shows how actual pressure is obtained by means of a hole in the pipe wall. The inner surface of the pipe adjacent to the hole must be smooth. The hole must be at right angles to the surface and there must be no burrs. The inner corners should be rounded with a radius 1/10 to 1/50 of the diameter of the hole. Fig. 2 shows a number of static holes. This form is in very common use and it is supposed that any variation in static pressure is averaged. It is questionable whether any advantage is obtained, however, from the multiplication of holes. Possibly the pressure reading is the average reading of the two holes on either side of the connection to the gage. This is because any difference in the static pressure will create a flow in the outer ring with different pressures at

various places. There will not be a constant pressure at all points of the chamber at the average of the different static pressures. When there is a proper length of straight pipe preceding the point of measurement, such arrangement as Fig. 5 is not necessary. When there is not a proper length it is questionable whether the arrangement of Fig. 5 will give good readings.

25 Static holes such as Fig. 4 are often made with burrs on the inner surface or with the pipe leading to the measuring instrument projecting into the stream. These practices cause serious errors and must be avoided. If it is not possible to make certain of proper holes of the type of Fig. 4, as, for instance, when work is done in the field, the arrangement of Figs. 6 or 7 must be used.



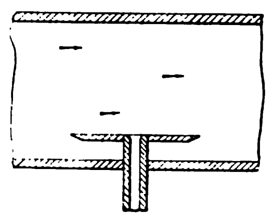


FIG. 6 STATIC PRESSURE BOAT FIG. 7 STATIC PRESSURE BY SIDE GAGE

26 For any of these types, the pressure holes must be of good size. They should be at least ¼ in. in diameter and preferably ¾ to 1 in. in diameter in order to avoid a drop of pressure which might occur with a small hole and a minute leak. One of the disadvantages of the common form of pitot tube is that the static holes are often so minute that the pipes to the pressure-measuring instrument must be of extraordinary tightness in order to secure accuracy. On the other hand, if a large hole is used in the pipe wall, or the equivalent, as in Figs. 4, 5, 6 and 7, and if the impact tube is made of good size with the opening ¾ to 1 in. in diameter, a slight leak will not have very serious effect. Nevertheless, precautions should be taken to keep the pressure piping tight.

TIGHTNESS OF PIPING

27 It is desirable to use large pipes for connecting the static and impact tubes with the measuring instrument. Standard ¼ or

$\frac{1}{8}$-in. piping should never be used for such purposes unless the connecting lines are very short. When the piping is of any great length, $\frac{1}{2}$-in. pipe or even 1-in. pipe should be used, because it is much easier to make the larger-size pipe tight, and because even a small leak will not cause much of a pressure drop. It is to be noted that the differential pressure measured is usually quite small, so that even a slight error in either pressure will cause an appreciable error in the differential.

28 The following methods of test for tightness of piping apply to pitot and impact tubes, flow-meter work, venturi-meter work, etc.:

a In case the connecting pipe is filled with water, a leak will always be shown by a drop. Such leaks tend to rust tight, however, so that the drop will soon disappear.

b In case the pipe is filled with gas, the entire outer surface including the surface of all fittings and joints must be gone over with a paint brush when pressure is on. If there is steam or hot gas in the pipe, this brushing must be done with cylinder oil. If there is air or cold gas in the pipe, it must be done with a thick lather of soap suds. Every part of the surface must be thickly coated. Leaks will be shown by bubbles.

c If there is a sensitive differential gage and if the piping is of considerable length, valves should be inserted on both the static and impact sides immediately adjacent to the pipe in which the flow is to be measured, as well as a by-pass valve connecting the two pressure pipes immediately adjacent to the measuring instrument. Such valves can be used for testing leakage as follows: The by-pass valve should be opened. Then the valves on the pressure pipes should be closed as rapidly as possible in the following order: First, nearly but not quite close the valve on the pipe giving the lowest pressure, then nearly but not quite close the valve on the pipe giving the highest pressure. Now completely close the valve on the pipe giving the lowest pressure, and then completely close the valve on the pipe giving the highest pressure. Lastly, close the by-pass valve. Fluid under pressure will be trapped in each of the pressure pipes. The measuring instrument will indicate a definite pressure difference which will remain unchanged if there is no leak on either side. The instrument will, of course, remain unchanged if there are equal leaks in the two sides, but this condition is almost inconceivable. If, however, the instrument does not remain unchanged the valves themselves may leak, even though the pipe be tight, so that the test gives no information under such conditions.

IMPACT TUBE AND HOLE IN THE PIPE WALL FOR MEASURING PIPE FLOW

29 The apparatus is to be arranged as shown in Fig. 8. However, instead of the single static hole as shown, other means of securing static pressure can be used such as shown in Figs. 5, 6 or 7.

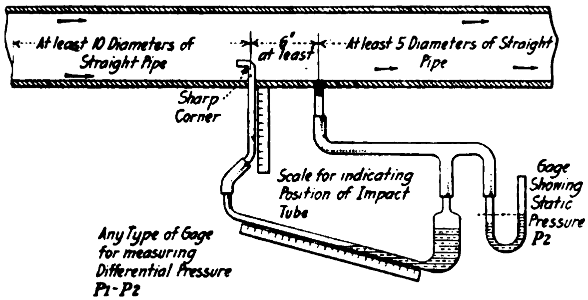


FIG. 8 IMPACT TUBE AND HOLE IN PIPE WALL FOR MEASURING PIPE FLOW

If methods of Figs. 6 or 7 are used, the impact tube must be placed at such a distance from the static tube that neither instrument will disturb the flow past the other.

30 There are numerous well-known forms of the differential gage which may be used.

31 If any of the tests give flows large enough to show a differential pressure of 1 in. or more of water vertical, it is desirable to connect up a vertical water U-tube in parallel with the differential gage. A steel scale with graduations on both sides should then be provided. On one side these graduations should be 1/64, 1/50, or 1/100 of an inch. The center lines of the two legs of the U-tube should be separated an amount equal to the width of the scale. The two legs of the glass must be of exactly the same diameter. By laying the scale against the glass and using a magnifying glass and a good light, very satisfactory readings can be obtained. If a number of readings are made with the differential pressure in the neighborhood of 1 in. of water, the average of the ratios of such readings to

the readings of the differential gage gives a satisfactory constant for calibration or checking of the differential gage.

32 The impact tube should be pushed clear across the pipe, readings being taken at both sides with the tube in contact with the pipe wall, and at a number of points between. One method is to space the positions at distances proportional to the square of the radius. Then the arithmetic mean gives a number properly weighted for the various areas involved.

33 The writer has found it satisfactory to measure the actual radius with each setting, spacing settings somewhat more closely

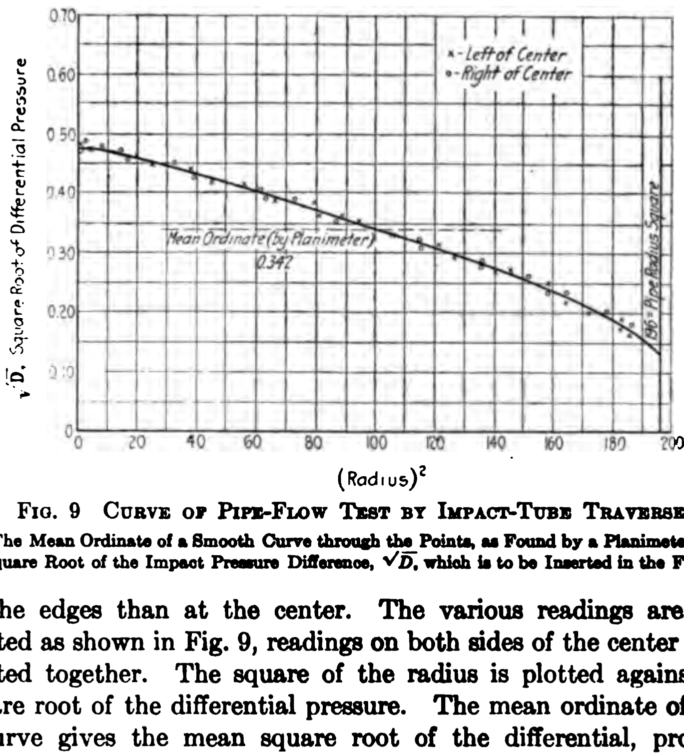


FIG. 9 CURVE OF PIPE-FLOW TEST BY IMPACT-TUBE TRAVERSE

The Mean Ordinate of a Smooth Curve through the Points, as Found by a Planimeter, Gives the Square Root of the Impact Pressure Difference, $\sqrt{D}$, which is to be Inserted in the Formulæ.

at the edges than at the center. The various readings are then plotted as shown in Fig. 9, readings on both sides of the center being plotted together. The square of the radius is plotted against the square root of the differential pressure. The mean ordinate of such a curve gives the mean square root of the differential, properly averaged for use in the formulæ given later. The static pressure is also to be measured independently. The computation work is slightly simpler if this is taken rather than impact pressure.

IMPACT TUBE FOR MEASURING PIPE PRESSURE

34 When a fluid, either liquid or gas, is flowing in a pipe, it is frequently desirable, for general purposes, to use the impact pressure

rather than the static pressure. An example of this is pressure in a pipe preceding a venturi meter or other form of orifice in a pipe line. The ordinary formulæ involve this pressure as well as the pressure at the throat or orifice end, and include a very troublesome corrective term for the initial velocity at the point where the initial static pressure is measured. If, however, this initial pressure is measured by use of an impact tube, the velocity head at this point is automatically added to the static pressure and a net pressure obtained, equal to the pressure which would have been obtained if the pipe preceding the orifice had been large enough to render the velocity head in it negligible. The formulæ for this case (given later)

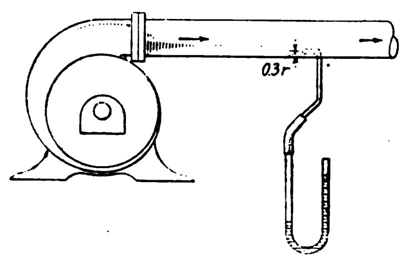


FIG. 10 IMPACT TUBE GIVING TOTAL PRESSURE RISE WITH ATMOSPHERIC INLET PRESSURE

are very much simpler than those in which the initial velocity has to be taken into account. Of course, if there is any foreign matter in the stream, it is likely to clog the impact tube, in which case this arrangement can not be used. In such a case, initial pressure must be measured by means of a hole in the pipe wall and the ordinary formulæ which take account of initial velocity must be used.

35 Another case in which it is convenient to measure the impact pressure is in finding the total pressure rise produced by a centrifugal pump or centrifugal compressor. If the discharge pipe of such a machine is very large, the velocity head in it is negligible, and pressure produced is the static pressure. If, however, the pipe is small enough to give an appreciable velocity head, as is often the case with fan blowers and centrifugal compressors producing a pressure of 2 or 3 lb. per sq. in., the energy actually put into the air by the machine is completely shown by the impact pressure. It is really correct to compute the efficiency of the machine on the basis of impact pressure. The arrangement is shown in Fig. 10. A similar arrangement can be used in the case of centrifugal pumps. If the inlet and discharge pipes are both of the same size, impact tubes give

the same difference as static holes. If, however, the inlet pipe is larger than the discharge pipe, as is often the case, impact tubes in both pipes automatically take account of the difference in velocity heads. The difference in the pipe pressures as shown by the impact tubes, plus the head equal to the vertical height difference gives the total work done by the pump. The arrangement is shown in Fig. 11. These remarks do not apply to the case of pulsating flow, as with a reciprocating machine.

36 In all cases, the impact-tube center should be about 0.3 of the pipe radius from the pipe wall.

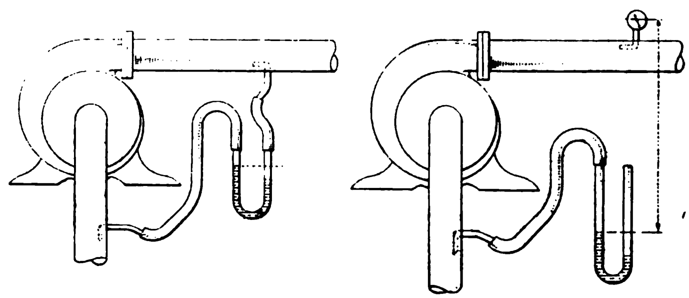


FIG. 11 IMPACT TUBES GIVING TOTAL PRESSURE DIFFERENCE BETWEEN INLET AND DISCHARGE PIPES

37 As an additional advantage in the use of the impact pressure rather than the static pressure, it should be noted that the pressure is obtained more accurately. The tapping of a pressure hole in the pipe wall is very uncertain and it is often easier, particularly in the case of field tests, to insert an impact tube which is certain to give correct pressure, than it is to take the necessary precautions to make sure that a hole in the pipe wall is free from burrs and projections.

SHAPE OF ORIFICES FOR MEASURING FLOW OF FLUIDS

38 The "orifice" frequently discussed in this article is a smoothly formed, convergent passage. Such an orifice is used at the entrance of a venturi meter, or for measurement of fluid discharged into the atmosphere, as discussed later. The exact shape by means of which the convergent portion is secured is not very important. The writer prefers the general shape shown in Fig. 20, formed by circular arcs. A cone with straight sides however is probably equally satisfactory, if the angle is gentle and if there is a smooth circular arc connecting with a straight portion at the end of the orifice. It

seems probable that any shape will be satisfactory which will obviously have a gentle action on the stream, without interference and eddies due to corners or sudden variations of section. When the orifice is being machined, it is not particularly important that the shape be made to fit a templet. The orifice wall should feel smooth when the hand is run along the surface, for if any lumps or ridges can be felt, they will cause irregularities in the fluid flow.

39 It is highly essential that there be a parallel portion at the end of the convergent portion, of sufficient length properly to size the jet. At the end of the convergent portion there may be a con-

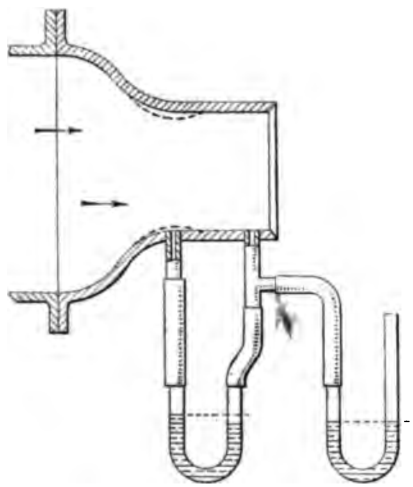


FIG. 12 APPARATUS TO SHOW CONTRACTION AT END OF CONVERGENT PORTION OF AN ORIFICE

traction of the jet, due to the fact that the orifice is not shaped to avoid contraction. The existence of such an effect can never be known except by an actual experiment, as indicated in Fig. 12. This figure represents a case where a comparison was actually made between pressure in a parallel portion of the orifice with atmospheric pressure on one side, and with pressure at the end of the convergent portion on the other side. It was found that pressure in the parallel portion was equal to atmospheric pressure, while pressure at the end of the convergent portion was below atmosphere, indicating the contraction in the jet shown in Fig. 12. In other words, there was a venturi-meter effect, and the jet at this point did not quite fill the orifice. Hence, if the orifice had ended at the end of the convergent portion, giving an orifice of the shape of Fig. 13, the jet escaping into the atmosphere would be of smaller diameter than the orifice end. The quantity would have appeared to be larger than it actually was. A similar situation is given by the other orifice of Fig. 13. If, however, we have an orifice of the shape of Fig. 20, or a venturi meter with the shape of Fig. 16, the parallel portion insures that the jet whose measurement is sought has the exact diameter of this parallel portion. The length of such a parallel portion should be from half diameter to diameter of the parallel portion. Greater length might introduce appreciable friction drop.

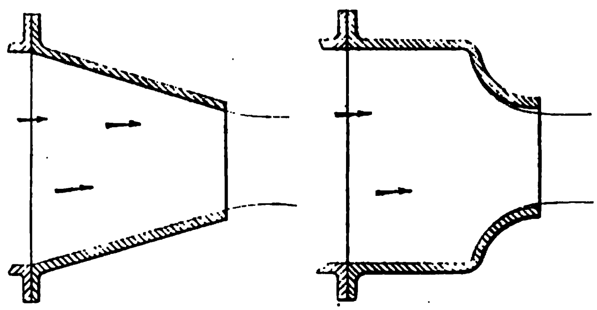


FIG. 13 POORLY SHAPED ORIFICES CAUSING JET CONTRACTION AND EFFECTIVE JET DIAMETER LESS THAN ORIFICE END DIAMETER, SO THAT MEASURED FLOW IS GREATER THAN ACTUAL FLOW

40 A minor point, of practical importance, is the chamfering of the outer end of an orifice. This prevents the edge from being burred or dented during handling and so preserves the diameter of the parallel portion everywhere. The parallel portion must, of course, be carefully machined so as to be exactly parallel and round.

VENTURI METER WITH IMPACT TUBE

41 The general arrangement is shown in Fig. 16. The use of such an impact tube avoids the computation work necessary to take account of the initial velocity head and the necessity for care in drilling a static hole. From the discussion already given it is obvious that theoretically the impact tube could be placed in the throat of the meter adjacent to the static tube. In the case of a large meter where the impact tube can be made so small that it will not affect appreciably the area nor the readings of the static tube, this arrangement is perfectly practical. However, in many cases

an impact tube of reasonable size would disturb the jet at the throat.

42 Fig. 16 also shows some additional readings which the writer has found convenient in the case of venturi meters used for experimental testing. For permanently installed venturi meters, these extras are of no particular advantage. The extra apparatus consists of an impact tube, C, in the pipe beyond the meter, and shows the pressure drop resulting from the use of the meter. It is quite necessary that the tube, C, be placed at a considerable distance from the meter in order that the jet may have wholly recovered from the divergence due to the meter. The streams, and the static pressure, particularly, are not normal at points near the end of the divergent part of the meter. The impact tube, C, is connected so as to show the final pressure directly, and also, the difference between this pressure and the initial impact pressure, A, giving the meter drop or pressure difference caused by the insertion of the meter. This meter drop, AC, is usually from $\frac{1}{4}$ to $\frac{1}{8}$ of the meter differential pressure, BA. The system shown involves four U-tubes, one of which is, of course, superfluous. However, the use of this extra U-tube gives a very valuable means for making a check on the readings. The readings will check as follows, if all pipes are tight and the apparatus properly set up. (See Fig. 16.)

If B and C are above atmosphere, the difference between B and AC will equal the difference between C and BA.

If B is below atmosphere and C above, the sum of B and AC will equal the difference between C and BA.

If B and C are below atmosphere, the sum of B and AC will equal the sum of C and BA.

The writer's custom has been to have the readings inserted in this equation as soon as they have been taken. If the equation is not satisfied there is a leak, or something else out of order in the apparatus.

PRESSURE OF A JET DISCHARGING FROM AN ORIFICE INTO THE ATMOSPHERE

43 The only case discussed in this paper is that in which the final pressure is greater than about half the initial pressure, so that the theoretical orifice is wholly convergent. In such a case, *the static pressure of the jet discharged from an orifice is exactly equal to the pressure of the region into which it discharges.* This law applies

primarily to the discharge of an orifice into the atmosphere. However, it also applies to discharge of an orifice into an enclosed region, such as is shown in Fig. 14, provided there is ample room all around the orifice for a free discharge of jet and no restriction such as would be likely to cause a venturi-meter effect. The static pressure of the region into which a jet discharges must be measured in a number of places, such as A, B and C, Fig. 14, and seen to be exactly the same. If the orifice is supplied directly from the atmosphere, the

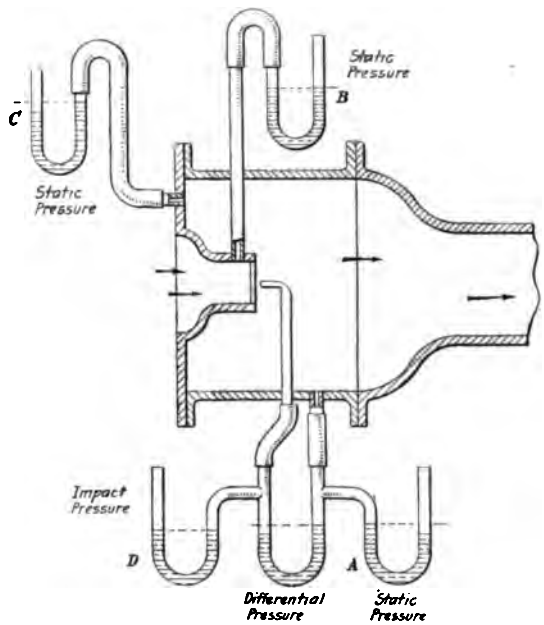


Fig. 14 Orifice Discharging into an Enclosed Region

The Static Pressure of Region into which Jet Discharges Must be Taken in a Number of Places, Such as A, B and C, and Seen to be Exactly the Same. If the Orifice is Supplied Directly from the Atmosphere, the Impact Pressure D will be zero.

impact pressure, D, will be zero. The accuracy of the above law is really the basis of all of the impact-tube work here discussed. So far as known, the law has never received any valid criticism. In one case a certain form of pitot tube, when inserted in a jet discharging into the atmosphere, gave readings below atmosphere at the static openings, and it was assumed that the static pressure of the jet was, therefore, below atmosphere. However, this particular form of pitot tube is now known to have static openings seriously in error. As an actual fact, readings of static pressure below atmosphere, when a pitot tube is inserted in a jet discharging into the atmosphere, are proof that the static pressure holes give erroneous readings.

44 If the orifice has a parallel portion of appreciable length at the end, the streams of the jet, as they discharge into the atmosphere, necessarily proceed in straight lines. This can also be seen by watching the jet discharging from a water or wet-steam orifice of the proper shape, or by feeling the surface of an air jet. The jet preserves a solid parallel column for quite a distance. If, now, a stream at the edge of the jet and a stream in the interior of the jet are pro-

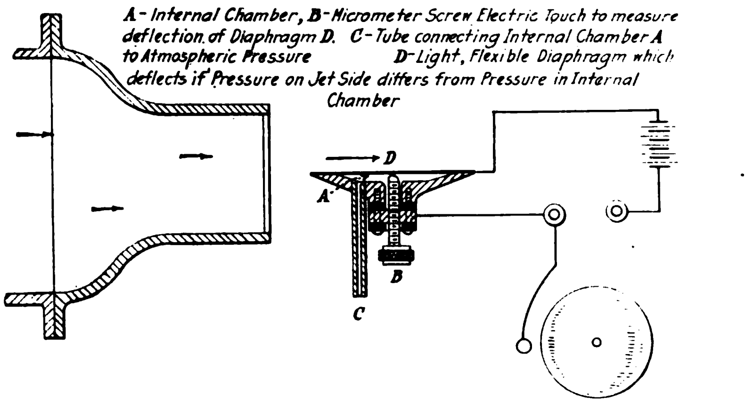


FIG. 15 APPARATUS TO SHOW EQUALITY OF STATIC PRESSURE IN JET AND ATMOSPHERE

ceeding in parallel directions, there can be no possibility of pressure difference between them. If there were any pressure difference, the flow would be in curved lines. A stream at the edge of the jet, proceeding in a straight line, is obviously at the pressure of the region. Hence, the same pressure exists throughout the cross-section of the jet.

45 In order to give additional verification of the above point, the writer made the experiment of Fig. 15. A diaphragm pressure gage was placed in a high-velocity jet with the diaphragm exactly parallel to the stream lines. This was accomplished by clamping the gage in a position perpendicular to the plane of the end of the orifice, as shown by an accurate square. The gage comprised a small chamber, one edge of which consisted of a very light, flexible dia-

phragm. A tube leading well outside the jet made the pressure in this chamber equal to the pressure of the atmosphere. A delicate micrometer screw, with an electric bell and battery, indicated the deflection of the diaphragm. The micrometer was screwed out until the bell just stopped ringing when the jet was not flowing. The jet was then turned on. Adjustment of the micrometer screw showed that the position of the diaphragm was unchanged. This experiment was made with a velocity of about 150 ft. per sec. and the deflection of the diaphragm was seen to be zero within a few hundred thousandths of an inch. The diaphragm was such that it would be deflected about 0.001 in. by a pressure of 0.005 in. of water. A

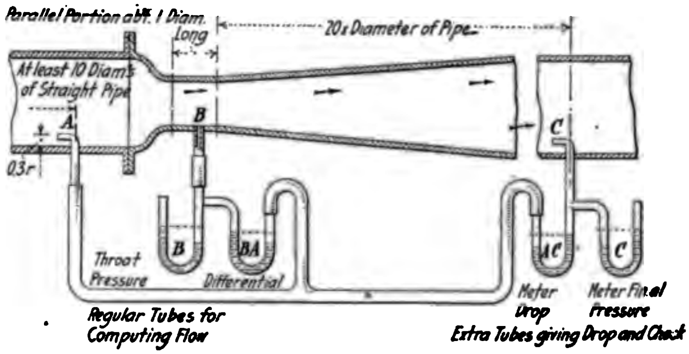


FIG. 16 VENTURI METER WITH IMPACT TUBE AND EXTRA TUBES FOR EXPERIMENTAL PURPOSES

similar result can be obtained if the side gage shown in Fig. 7 be set up in a jet, exactly parallel to the stream lines, and connected to a delicate pressure measuring instrument.

46 The static pressure of the jet being thus the pressure of the region into which it discharges it follows that the differential between impact and static for a jet discharging into the atmosphere is given by a gage which reads pressure above atmosphere.

ORIFICE CALIBRATION BY MEANS OF IMPACT TUBES

47 Many calibrations of orifices discharging high-velocity jets of air have been made by the writer with the apparatus of Fig. 17. This consists of a stationary impact tube placed in the center of the jet, slightly beyond the plane of the orifice end, and a very small movable impact tube which can make a traverse across the edge of

the jet covering a space of ¼ to ½ inch. The differential pressure between the two tubes is read by means of a vertical water U-tube, or some more delicate instrument, as circumstances warrant. The impact pressure is constant everywhere in the jet, except for a narrow ring around the edge which is traversed by the small impact tube. An impact tube about $\frac{1}{32}$ in. outside diameter has been used with a 4-in. orifice, traversing a space of about ¼ in., and proportionally with larger orifices. The results are plotted in Fig. 18. The ratio of the area of the small corner C to the area AB, of the large rectangle, gives the velocity loss. By subtracting this from unity,

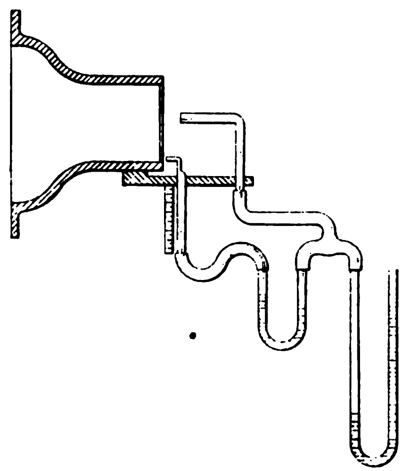


FIG. 17 ORIFICE CALIBRATION BY TRAVERSE OF AN EXPLORING IMPACT TUBE

the velocity coefficient of the orifice is obtained. Many such tests indicate a velocity coefficient of about 0.99. These experiments have been accurate to within ½ of a per cent, so that the coefficient may be somewhere between 0.985 and 0.995. It seems probable that the larger the orifice the greater the coefficient.

48 It should be noted that this method *gives an accurate calibration of any orifice without measurement of flow by means of tanks, weirs, gas holders, etc.*

49 Calibrations with water orifices and the same general system were made by John R. Freeman, and are described in Transactions, A.S.C.E.[1]

[1] No. 426, v. 21, Nov., 1889; No. 479, v. 24, June, 1891; and v. 13, p. 382.

EXPLORATION OF IRREGULAR JETS

50 The same general methods have been used by the writer to find velocities of various types of jets, such as those from the nozzles and stationary buckets of steam turbines. Small impact tubes were used which could be rotated and moved in two directions. At each position the tube was rotated until the maximum reading was found, and the angle noted. In this way, the stream lines were completely mapped out, both as to velocity and direction. In all

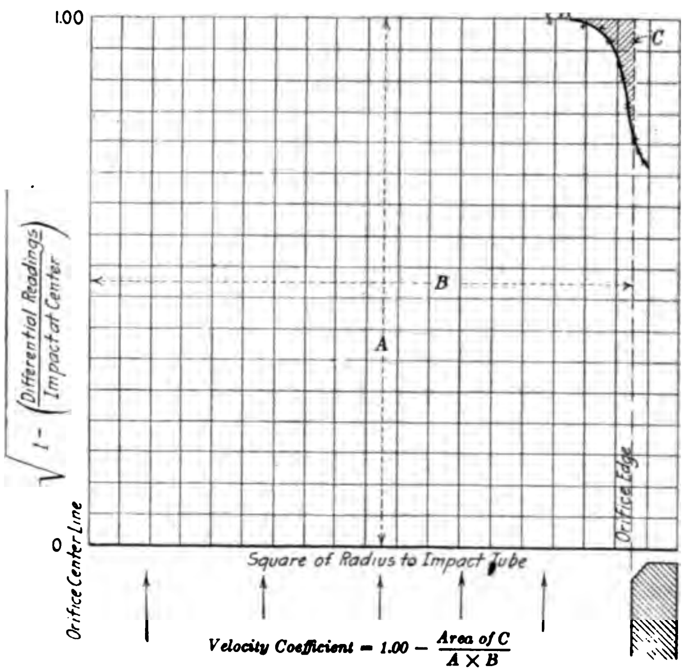


FIG. 18 CURVE OF ORIFICE CALIBRATION BY IMPACT-TUBE TRAVERSE

cases the work was done with jets discharging in straight lines into a large chamber which had a definite and uniform static pressure. Static pressure variation and curvilinear motion cannot be investigated easily in this way. The static pressure was found in a dead corner where the jets could not give error. The velocity was computed from the theoretical laws for adiabatic flow and the impact tube law already given.

51 The jets from a stationary set of turbine buckets were concentrated near the active faces, with gaps between. Hence a jet

from a moving turbine wheel would be pulsating, and therefore could not be explored by this method.

MEASUREMENT OF DISCHARGE OF HIGH-PRESSURE AIR BY MEANS OF AN ORIFICE AND IMPACT TUBE

52 The general arrangement is shown in Fig. 19. The discharge pipe of the machine which is being measured is connected

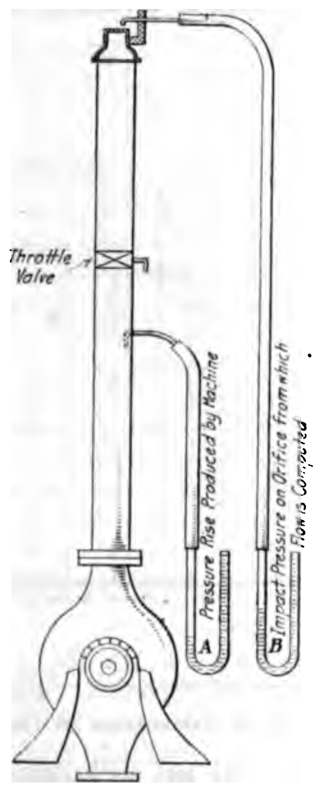


FIG. 19 MEASUREMENT OF QUANTITY OR FLOW BY MEANS OF ORIFICE AND IMPACT TUBE

up with a considerable length of straight pipe to the orifice. It is usually desirable to have a throttle valve in the pipe so that various amounts of flow giving various loads on the machine can be measured by means of a single orifice. It is necessary that there be at least 10 diameters of straight pipe between the throttle valve and the orifice.

If it is desired to measure the discharge pressure produced by the machine, it is necessary, in addition, to have about 10 diameters of straight pipe between the machine itself and the throttle valve. An impact tube inserted in this pipe gives the net pressure rise which the machine under test produces, while the impact tube on the orifice gives the volume or quantity passing through. The orifice should be selected by use of the formulæ, appearing later in this paper, so that it will just discharge the greatest quantity which the machine will produce with an orifice impact pressure equal to the total pressure produced by the machine when this quantity is flowing. Then the highest flow of the test will be obtained with the throttle valve wide open. For this flow, the readings of the impact tube in the pipe preceding the throttle valve, and that in the orifice jet, should be identical, except for slight friction loss in the pipe. This identity gives a check on the validity of the readings and apparatus. Various tests with successively lower amounts of flow are given by closing the throttle valve so as to decrease the orifice pressure.

53 The impact tube is to be stationary and in the center of the jet, slightly outside the plane of the orifice end. This distance should be about 1/5 of the orifice end diameter. The tube is in plain view throughout the test so that it can always be seen to be properly in line and free from obstructions. This is one of the advantages of this method.

54 The general method here described has been used on centrifugal pumps and fire streams. It is just as convenient with high velocity jets of air or other gases.

55 The flow is to be computed by the formulæ, appearing later in this paper, using a velocity coefficient of 0.99 for orifices described in this article. In case great exactness is desired, the coefficient must be found for the actual case by means of a traverse already described in Par. 47.

56 The parallel portion at the end of the orifice should have a diameter about one-third of the diameter of the straight pipe preceding the orifice. The orifice, of course, may be smaller than one-third pipe diameter without any effect. It is probable that the coefficient does not differ greatly from 0.99, even if the orifice is but one-half of pipe diameter. In all such cases, the effect of the varying velocity at different radii in the pipe is not transmitted through the orifice, and the discharge jet has the same velocity at all points, except at a narrow band close to the edge. For cases where the orifice diameter is greater than about one-half pipe diameter, it is

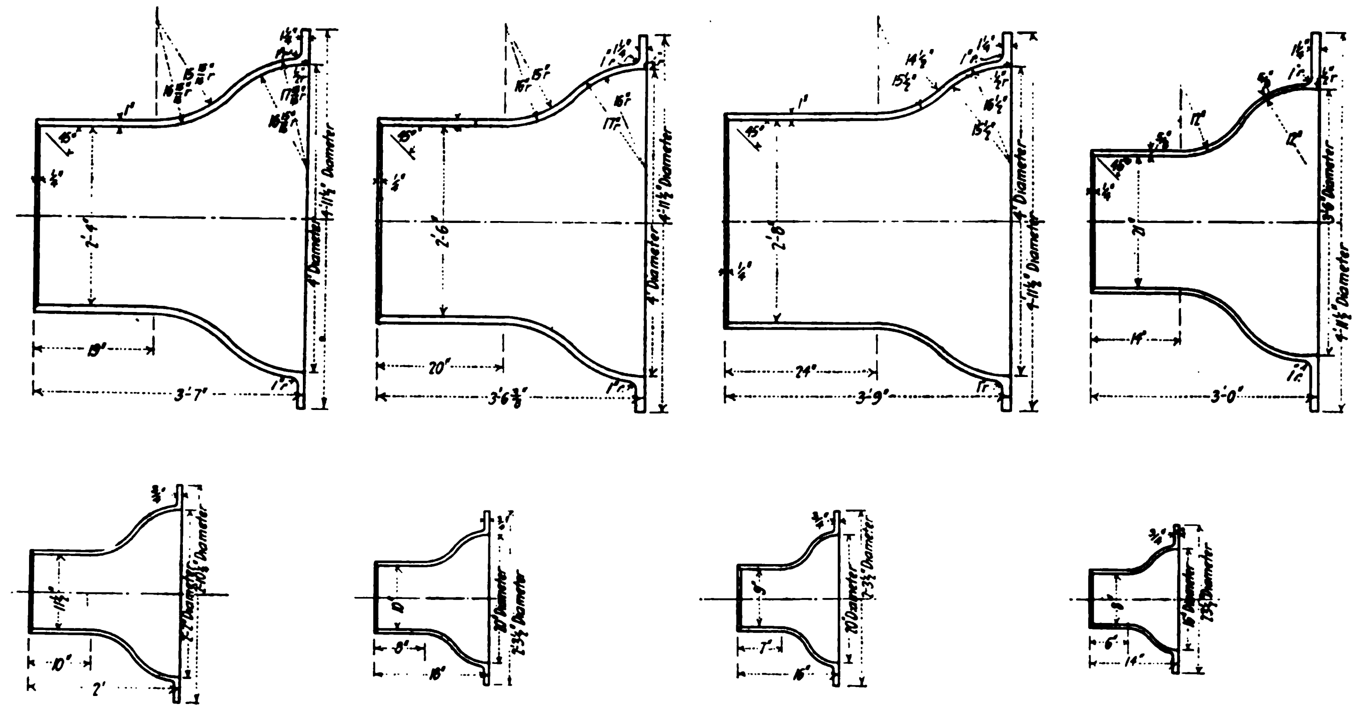


Fig. 20 Dimensions of Some Measuring Orifices

probable that the variation of pipe velocity has some effect on the discharge jet.

57 The impact tube, as here discussed, can only be used with continuous flow, such as is given by centrifugal machinery. Pulsating flow, such as is given by reciprocating machinery, can not be integrated by the impact tube.

58 In order to compute the quantity or flow, it is also necessary to have the temperature in the pipe preceding the jet. This must be secured by the insertion of a thermometer through the pipe wall. However, as already pointed out in Par. 14, a practically identical reading will be given by a thermometer inserted in the jet outside the orifice. In all cases, for accurate work, the bare bulb of the thermometer should be in the jet, and wells or metal casings should be avoided.

59 The above methods have been used for measurement of many million cubic feet of air discharged by various types and sizes of fan blowers and centrifugal air compressors in the Lynn Works of the General Electric Co., since 1904. Fig. 20 shows some of the orifices which have been used and Figs. 21 and 22 show the arrangement of the impact tube.

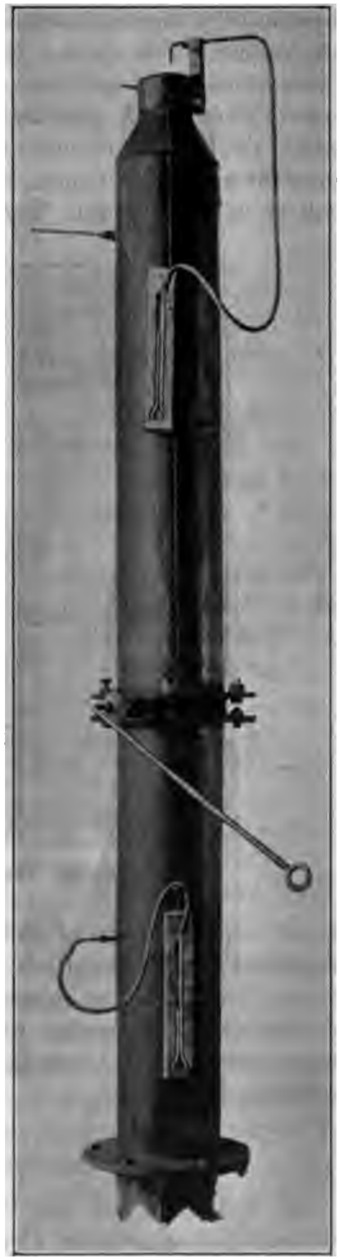


FIG. 21 ORIFICE, IMPACT TUBE AND DISCHARGE PRESSURE ARRANGEMENT

ADVANTAGES OF THE ORIFICE METHOD

60 If a machine whose flow is to be measured can be arranged so that its discharge can be wasted, the use of an orifice in the discharge pipe is the only method of flow measurement which should be considered. The reason is that nearly the entire pressure rise

produced by the machine can be utilized in causing discharge through the orifice. This gives a reading of such magnitude that accurate observations can be made. For instance, if the machine being tested gives an air pressure of 15 lb. per sq. in., the reading from which the flow is computed will be about 30 in. of mercury. If the machine gives an air pressure of about 1 lb. per sq. in., the reading will be 27 in. of water. Such magnitudes are easily measured and

FIG. 22 ORIFICE AND IMPACT TUBE

do not require the use of inclined gages, hook gages, or other means of getting small differentials.

61 The use of the impact tube gives the advantages of simplicity of formula in connection with initial velocity head, and ease of proper arrangement, over the use of a static hole in the pipe-wall preceding the orifice.

DISCUSSION OF FORMULÆ

62 Following are the theoretical formulæ for adiabatic flow from a region of pressure P_1 and temperature T_1, where the velocity is negligible, to a region of pressure P_2. These are the proper formulæ for use with impact tubes. A detailed discussion of them was given

by the writer in American Machinist.[1] Four types of formulæ are given.

63 *Exact Thermodynamic Formulæ.* For any case, the velocity of the jet is the spouting velocity theoretically produced by adiabatic expansion from P_1 to P_2.

For any perfect gas:

$$q = \frac{60\, T_0 \pi d^2 P_1}{4 \times 144\, P_0} \sqrt{\frac{2\, gJC_p}{T_1}} \sqrt{\left(\frac{P_2}{P_1}\right)^{2/K} - \left(\frac{P_2}{P_1}\right)^{1+1/K}}$$

$$Q = 2587\, d^2 P_1 \sqrt{\frac{C_p}{T_1}} \sqrt{\left(\frac{P_2}{P_1}\right)^{2/K} - \left(\frac{P_2}{P_1}\right)^{1+1/K}}$$

For air:

$$Q = \frac{1260.9\, d^2 P_1}{\sqrt{T_1}} \sqrt{\left(\frac{P_2}{P_1}\right)^{1.422} - \left(\frac{P_2}{P_1}\right)^{1.711}}$$

64 *Nearly Exact Formulæ.* For values of the differential up to the greatest possible value in the present discussion, which is about equal to the static pressure, very exact results are obtained by using the static pressure, P_2 and a corrective factor, in the incompressible fluid formula. The resulting formulæ are very much easier to use in computation work, and give almost exactly the same results as those above in the case of air and other gases for which C_p/C_v is about 1.41.

For any perfect gas:

$$Q = 187.28\, d^2 \sqrt{\frac{P_2 D}{T_1 y_s}} \sqrt{1 - \left(\frac{3}{2\,K} - 1\right) D/P_2}$$

For air:

$$q = 19.162\, d^2 T_0/P_0 \sqrt{P_2 D/T_1} \sqrt{1 - 0.0665\, D/P_2}$$

$$Q = 677.5\, d^2 \sqrt{P_2 D/T_1} \sqrt{1 - 0.0665\, D/P_2}$$

65 *Closely Approximate Formulæ for D Less Than* 10 *Per Cent of* P_2. For cases of air, and other gases where the specific heat ratio is in the neighborhood of 1.41, if the pressure which appears in the formula be made the final or static pressure of the jet, accurate results are obtained with a form of formula equivalent to the incompressible fluid form, with differentials up to about 10 per cent of the absolute static pressure.

For any perfect gas:

$$Q = 187.28\, d^2 \sqrt{\frac{P_2 D}{T_1 y_s}}$$

[1] September 20 and 27, 1906.

For air:

$$q = 19.162\, d^2T_0/P_0 \sqrt{P_2D/T_1}$$

$$Q = 677.5\, d^2 \sqrt{P_2D/T_1}$$

66 *Incompressible-Fluid Formulæ.* For water, or for air, gas or steam where the differential is less than 1 per cent of the static pressure,

$$Q = 31.495\, d^2/y_s \sqrt{Dy}$$

For air:

$$q = 19.162\, d^2T_0/P_0 \sqrt{PD/T_1}$$

$$Q = 677.5\, d^2 \sqrt{PD/T_1}$$

Quantity in cu. ft. per min. with density, y, same as at point of measurement $= 31.495\, d^2 \sqrt{D/y}$.

For the case of a venturi meter or orifice without an impact tube, but with static pressure difference only, the initial velocity has to be taken into account by dividing the results of the above formulæ by the following expressions.

Exact divisor (for Par. 63):

$$\sqrt{1 - (d/d_1)^4 (P_2/P_1)^{2/K}}$$

Nearly exact divisor (for Par. 64):

$$\sqrt{1 - (d/d_1)^4 (1 - 1.422\, D/P)}$$

Closely approximate divisor (for Pars. 65 and 66):

$$\sqrt{1 - (d/d_1)^4}.$$

NOTATION

67 All pressures and pressure differences are in pounds per square inch.

1 in. of water at ordinary atmospheric conditions
= 0.036 lb. per sq. in.

1 in. of mercury at ordinary atmospheric conditions
= 0.49 lb. per sq. in.

B = reading of the barometer as ordinarily given, in inches of mercury at 32 deg. fahr. $0.4912B$ = barometric pressure in pounds per square inch.

P = absolute pressure in pounds per square inch. This is the gage pressure or pressure above atmosphere, plus the barometric pressure.

P_1 = the higher absolute pressure or impact pressure, and P_2 = the lower absolute pressure or static pressure, or pressure of the region into which the jet is discharged, or barometric pressure, $0.4912\, B$.

D = differential pressure in pounds per square inch, or $P_1 - P_2$. This is the reading of gage connected between impact tube and static tube, or between impact tube and atmosphere, in case of a jet discharging into atmosphere.

T_1 = absolute fahrenheit temperature in pipe preceding the jet. This is ordinary fahrenheit temperature plus 459.5 deg., and is the temperature shown by a thermometer inserted anywhere in the jet, or in the pipe preceding it.

d = orifice diameter in inches. This is the diameter of the parallel portion at the end of an orifice, or venturi tube, or the diameter of the pipe, if pipe flow is being measured.

q = quantity discharged, cubic feet per minute, in terms of volume at any selected pressure, P_0 lb. per sq. in. absolute, and temperature T_0 absolute. These are usually the values at inlet of the machine in test and then q gives "cubic feet of inlet air."

Q = quantity discharged, in cubic feet per minute, in terms of volume at standard conditions, 14.7 lb. per sq. in. absolute, and 60 deg. fahr. This is obtained by putting P_0 = 14.7 and T_0 = 519.5 in formulæ for q.

y = density in pounds per cubic foot. y_s = value at standard conditions, 14.7 lb. per sq. in. pressure and 60 deg. fahr. temperature.

68 In making the computations, the following values of fundamental constants have been used.

J = mechanical equivalent of heat = 778.

g = gravity = 32.17

y_s = density of air at 14.7 lb. per sq. in absolute and 60 deg. fahr. = 0.0764 lb. per cu. ft.

$C_p/C_v = K$ = specific heat ratio. For air, $1/K = 0.711$, $C_p = 0.2375$.

These constants satisfy the fundamental relations for a perfect gas,

$$C_p - C_v = \frac{14.7 \times 144}{519.5\, J y_s}$$

LIMITATION OF FORMULÆ

69 In all cases, P_2/P_1 must be greater than the critical value. That is, the orifice theoretically required must be wholly convergent, and the pressure at the orifice end will influence the flow. Then, for any perfect gas P_2/P_1 must be greater than $\left(\frac{2}{K-1}\right)^{K/(K-1)}$. For air and any gas for which $1/K = 0.711$, P_2/P_1 must be greater than 0.5272. For steam P_2/P_1 must be greater than 0.58.

VELOCITY COEFFICIENTS

70 In using the formulæ for pipe flow, as in Fig. 8, the average value of $\sqrt{D}$ is to be found as in Fig. 9, and inserted directly in the

formulæ. In case of venturi meters or orifices, the observed impact and static pressures are to be inserted in the formulæ, and the quantity multiplied by a velocity coefficient of about 0.99 for orifices of the shape here shown.

USE OF FORMULÆ

71 The formulæ apply to cases of measurement of pipe flow by means of impact and accurate static pressure openings, or to venturi meters and orifices when initial or final impact pressure and final static pressure are taken, or to any other definite jet whose static and impact pressure can be found.

DISCUSSION

R. J. S. Pigott said that the curve of the nozzle should not have too short a radius, so as to induce the stream to leave the walls and form a *vena contracta* in the throat or beyond. It was not so much the effect upon the average velocity ultimately as the effect upon the impact, making vibrations in the jet similar to those of imperfectly designed steam nozzles.

He agreed to the value of the single static hole, rather than the pitot. The pitot tube offered two very pronounced disadvantages. It was difficult to be certain that the current was flowing past it in the right direction unless there was a device for accelerating the flow and smoothing out the eddies. A further disadvantage lay in the lack of standardization of the static openings. There were several designs, the tube with the pinholes in the side being in most favor at present and the Taylor tube practically discarded; but it seemed quite certain from the author's tests that the static hole in the side of the pipe would give accurate pressure results, provided it was not in the sweep of the curve.

The necessity of obtaining a straight section of pipe for making measurements of flow might be overcome by the use of guide vanes. Experience in testing blowers in the 59th Street Interborough station showed that a scheme similar to the spacer used in egg boxes would destroy the eddies in a pipe line. A pitot tube or impact orifice could be placed close to such a screen, and no appreciable difference would be noticed over the cross-section of the pipe.

He had used this scheme in connection with different pitot tubes for the measurement of steam. The use of such a screen would eliminate the necessity of having a long run of straight pipe up to the measuring device, whatever it might be — they were all sensitive to the same influences, and even the venturi meter would feel the influence of the eddy currents on the upstream side. If this device, or a straight pipe was used, the venturi meter would give the most consistent results in the measurement of the amount of water or steam used in a plant.

Next to the venturi meter for general reliability of reading would come the disk orifice, and last the various forms of pitot tube; and if pains were taken to have an absolutely satisfactory form — and pipes were led to the tube, consistent results might be secured. The fault was not in the pitot-tube device, but in the way it was handled.

The pitot tube suffered from a disadvantage for steam and water measurements. In a pipe line small pressure drops had to be dealt with, and the pitot tube had the smallest pressure drop of any of these devices, and consequently any leakage was very much more important in the case of the pitot-tube device than in the disk orifice or venturi meter, which dealt with larger pressure differences. If it were not for this, the pitot tube, with the static arrangement in the pipe, could compete in most cases with the venturi meter or the disk orifice.

In answer to a question by F. J. Richardson, he said that he had found no difference in the velocity pressures where the velocity was high or low in using the screen.

In reply to a further question by Mr. Richardson, he said that where he had no room to make a long radius elbow in a pipe to reduce eddies, he avoided the friction drop by putting in a series of fins or turbine blades to direct the flow.

Frederick N. Connet. Sometimes the currents are of a helical nature, that is, combining forward and rotary flow, in which case there will be a higher pressure at the upstream point of the venturi than that which would be due to the static pressure of the air, and in that case the method proposed by Mr. Pigott is entirely successful.

Another method that can sometimes be used, and is preferable to Mr. Pigott's because of the occasional danger of the breakage of the grid by the carrying of solid objects through the pipe, is to place a few radial fins within the pipe ahead of the meter tube. These need not extend inwardly more than 25 per cent of the diameter, and usually there need not be more than 4 or 6 such fins to entirely overcome the trouble.

Some years ago we had a similar case in a pair of venturi meters at one of the high-pressure pumping stations in a western city. The flow came horizontally, and then passed an elbow, and then passed another elbow, in a direction perpendicular to the first, so that there was positively rotary motion imparted to the water, and two venturi meters in series with each other did not give the same results. The first one had the rotary flow. A short section of pipe (about 3 ft.) was taken out and another pipe put in containing two cross-pieces or vanes of sheet iron, and this change made the two venturi meters read alike.

G. G. CREWSON described a form of device he had used on compressed air through a wide range of velocities. The peculiarity of this device was that the tube extending into the pipe was in the form of a vee, at a small angle, not over 30 degrees, facing the flow. With this he had obtained more uniform results over a long range of flow than with any other types of impact tubes he had tried. He attributed this to the fact that the tubes set up more or less disturbance in the flow of air themselves, and by making this long vee he could get the point of impact into the undisturbed flow as near as possible.

WM. H. KAVANAUGH gave an example from his own experience of a straightening device to take care of the swirl of water coming out of a standpipe used to water locomotives.

EDGAR BUCKINGHAM pointed out that the devices mentioned by Mr. Pigott were frequently employed in wind tunnels used for experiments on the aerodynamic properties of aeroplane models. The egg-container device was used in almost all wind tunnels for getting rid of eddies and making the current as nearly straight as possible. Regarding the turbine-blade device, it might be remarked that it was used in the Goettingen wind tunnel, where a fan was employed for pumping air around in a closed circuit, and the space available was rather restricted, so that sharp turns were unavoidable.

JOHN L. ALDEN (written). An impact tube with properly made static-pressure holes in the pipe wall will give as accurate a determination of velocity as the best pitot tube. In addition, the tube itself is much less delicate and is more easily manipulated because of the absence of the tiny static-pressure orifices. However, it is extremely doubtful whether the large static openings in the pipe as described by the author, are not a source of error. Holes ½ in. to 1 in. in diameter are very likely to be influenced by the velocity, and will probably give a combined reading of static and velocity pressure. If anything, the error is greater by using a large hole and getting velocity influence than by using a small hole and having a slight possibility of leakage. This leakage trouble is one not ordinarily encountered, I think, and it would seem that a lower static reading should be attributed to the measurement of the true pressure with the small hole rather than to leakage. The tests by W. C. Rowse at the University of Wisconsin, recorded in Vol. 35 of the Transactions, show pretty conclusively the effect of large static holes in recording velocities lower than the actual. The same conclusions apply in a considerable degree to static orifices in a pipe wall. The piezometer ring is unreliable, for the reason the author has given. It does not correctly average the static pressures around the pipe.

The impact-tube law as stated in the paper is merely a restatement of the law of conservation of energy. In other words, the kinetic energy of the jet may be transformed into potential energy in the impact tube with only the immeasurable loss in the tube itself. Such a conclusion should be expected, as the impact tube is primarily an instrument measuring energy.

Testing a fan by the use of a single impact tube in the discharge pipe, as shown in Fig. 10, does not give a true indication of the output of the fan. Two tubes must be used, or the static vacuum at the inlet must be measured, for otherwise the fan is not credited with the negative pressure which it is producing. This suction is used in creating flow through the inlet. Part of it is shown in the form of velocity and the rest is lost in overcoming the resistance of the inlet. Where the inlet and outlet areas are equal, the pressure not credited to the fan is the entry loss referred to. The method shown in Fig. 11 is the correct one for fans and blowers as well as centrifugal pumps.

The writer believes that the orifices shown in Fig. 20 would give very doubtful results if the velocity of approach were appreciable. From the standpoint of smooth stream flow these nozzles are very bad. At high entrance velocities, 3000 ft. per min. or greater, the contraction of the jet at the throat would probably be great enough to prevent filling of the mouth of the orifice. In other words, the condition of Fig. 12 would be greatly increased, and the expansion of the jet to the size of the orifice would not take place until the end of the orifice was passed. If this condition actually occurred, the area of the orifice could not be used in calculating the volume. Unless each orifice were calibrated for all high velocities, it would seem unsafe to use this form. If the approach were more gradual and the fillets were large to smooth out the stream lines, an orifice might be made which would cause no contraction whatever.

VICTOR R. GAGE (written). Work done on blower testing in the Sibley College laboratories verifies some of the conclusions drawn by the author. We have found that the impact tube gave reliable and consistent results, not subject to error from slight misalignment or from stream lines not parallel to the pipe walls. Difficulty has been encountered in obtaining a satisfactory method for determining the pressure head, *i.e.*, the so-called "static pressure" of moving fluids.

It is our custom to use as small an opening as is possible, without rounding the corners, but with all burrs removed, and with the pipe carefully smoothed, for measuring pressure head. Our laboratory experience has led us to this. In one case, using a large hole, the pressure head was of a considerably different value than when the manometer was connected to another and exactly similar hole about nine inches away. By greatly reducing the size of the holes the values became practically the same. Further investigations indicated the existence of eddies and standing waves. The conclusion was drawn that a large opening was subject to more danger of receiving some (positive or negative) impact from currents of fluid not parallel to the pipe wall, and that the skin-friction effect would protect a small hole to some extent. The smaller the hole, the more the protection. This conclusion is tentative. With ordinary or higher velocities the flow is always turbulent, and care must be exercised in obtaining measurements of pressure head, both as to the means used and the place where the instruments are located.

With air at very high velocities (about a mile a minute) it was once found that the pressure head was not constant across an 8-in. pipe, and was actually less than atmospheric pressure although the blower was discharging into atmosphere through the pipe. The discharging steam was convergent after leaving the pipe.

For several years the arrangement shown in Fig. 8 has been used in place of a pitot tube, as being more satisfactory, in the regular students' laboratory experiment upon testing of blowers.

In regard to the use of a thermometer, we have used one first bare and then protected, placed in the jet of steam in a throttling calorimeter to illustrate the direction of the changes in the adiabatic conversion of heat into kinetic energy, and back again to heat. The bare thermometer bulb only partially checks kinetic energy; there is a very perceptible temperature rise upon furnishing a more complete checking of the velocity, although the bare-bulb indication is considerably higher than the temperature corresponding to isentropic expansion.

The Author. Mr. Alden refers to Mr. Rowse's paper in Vol. 35, Trans.Am.Soc.M.E., as showing that large static holes give error in static pressure. In all cases the errors reported by Mr. Rowse in measurement of static pressure were in the static pressure in a pitot tube. The reason they were in error is due to the fact that the flow was not parallel to the surface. Mr. Gage also notes error due to large static holes when there were eddies. I believe, that with proper length of straight pipe, or other arrangement for insuring that flow along the surface is parallel to it, the size of the static hole is unimportant. However, there is no objection to a small hole if any one prefers it and has tight piping.

Mr. Alden has some fears regarding the shape of the orifices illustrated but actual experience with these orifices for many years shows that his fears are groundless and that the mouth is filled in all cases, even for extremely high velocities. The essential point of an orifice is a large fillet or gradual curve just preceding a parallel portion. A long gradual approach is not necessary. In steam-turbine nozzles, where very high velocities are used, this has also been found to be the case.

No. 1555

THE FLOW OF AIR AND STEAM THROUGH ORIFICES

By Herbert B. Reynolds, New York, N. Y.
Junior Member of the Society

Part I The Flow of Air through Orifices, including a Study of Pitot and Venturi Tubes for Measuring Compressed Air

INTRODUCTION

In connection with the running of air-compressor tests, and the checking of flow meters used on the lines of a large compressed-air system, the author was at a loss for simple yet accurate means of measuring compressed air. The most common method is by use of the orifice. However, very few data are available which give the coefficient of discharge through orifices when the pressure difference exceeds a few inches of water.

2 In the following experiments use has been made of the simplest form of orifice possible, that is, one in a thin plate inserted between two standard pipe flanges. The thin-plate orifice gives the lowest coefficient of discharge of any form. Orifices of the converging type give much higher coefficients, however there is no advantage in the higher coefficient when the purpose of the orifice is simply to measure the air. The greatest advantage in the use of the thin-plate orifice is the ease with which it can be made and duplicated.

3 From the data of the tests an empirical equation has been derived for computing the flow of air through a thin-plate orifice, under any pressure from 15 up to 100 lb. gage. This equation contains four factors upon which the flow of air depends — the pressures on the two sides of the orifice, the temperature of the air entering it, and its diameter. Each one of these measurements can be taken

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers. Received Honorable Mention, Junior Prize Contest, 1916.

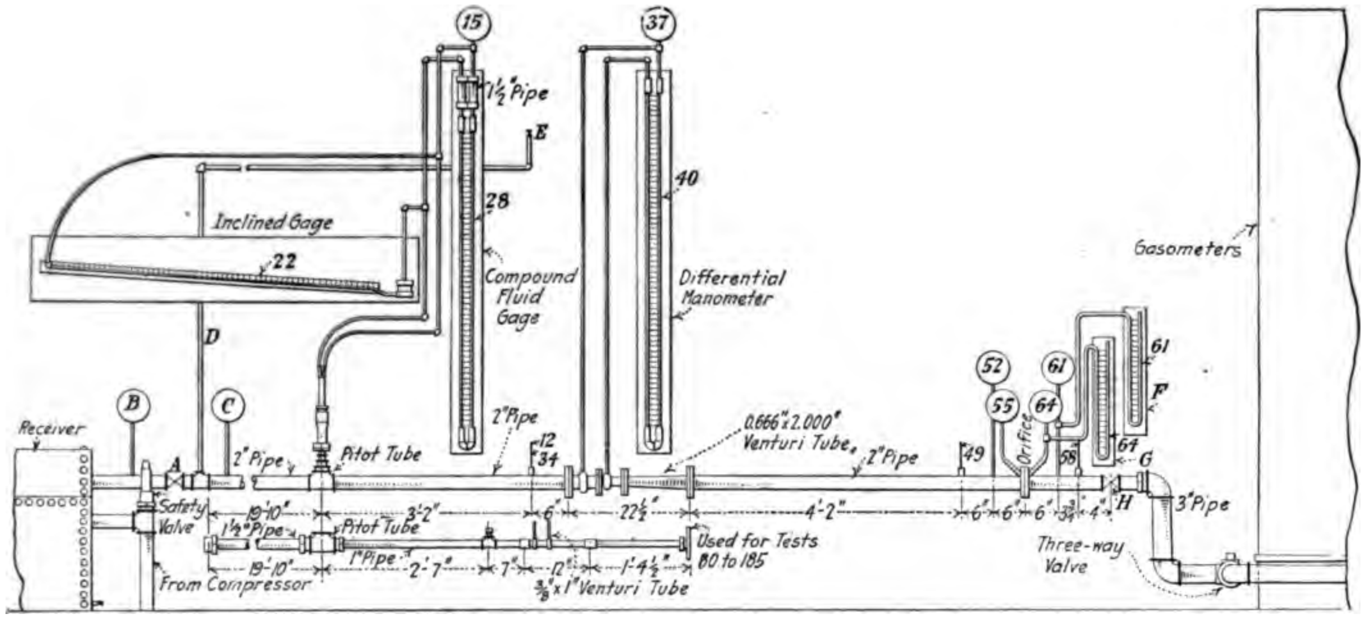


Fig. 1 Layout of Apparatus for Air Tests

very accurately with simple instruments, requiring no special or elaborate apparatus.

4 Often it is desirable to measure the flow of air in a pipe line without causing any drop in pressure, as when an orifice is used. For such needs data have been collected giving the coefficients of two venturi tubes and of one Taylor pitot tube.

5 The apparatus for the experiments was set up in the mechanical laboratories of Sibley College, Cornell University, in the spring of 1913. The author is indebted to Prof. R. C. Carpenter, Mem. Am.Soc.M.E., and to Prof. W. M. Sawdon, Mem.Am.Soc.M.E., for their advice during the course of the work.

FIG. 2 GENERAL VIEW OF APPARATUS USED DURING AIR TESTS

DESCRIPTION OF THE APPARATUS

6 The compressed air was furnished by a two-stage steam-driven compressor of about 150 cu. ft. capacity. A slight additional capacity was obtained by using a Westinghouse locomotive compressor. These two compressors discharged into a common receiver of about twelve cubic feet capacity. The rest of the layout of the apparatus can be understood easily from Figs. 1 and 2. The air passed from the receiver through a 2-in. pipe line to a valve A, Fig. 1,

that was used to regulate the pressure on the apparatus, a gage each side of it showing the pressures. To obtain a still closer adjustment of the pressure a small pipe was tapped into the line, D, Fig. 1, at the end of which a small valve E was attached.

7 With the apparatus arranged for tests Nos. 1 to 79, the air, after passing the throttle valve A, flowed through a 2-in. pipe at the end of which a standard Taylor pitot tube was inserted through a pipe tee. The tube extended up into the pipe, so that the tee did not interfere with the motion of the air until after the pitot tube had been

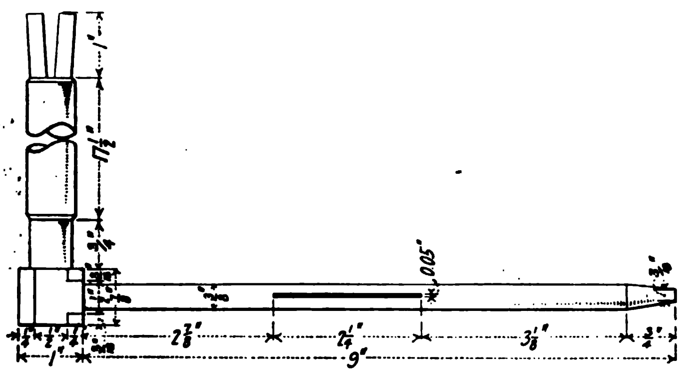


FIG. 3 TAYLOR PITOT TUBE

passed. Fig. 3 gives the dimensions of the pitot tube used. The stem of the tube passed through a stuffing box which prevented leakage. This stuffing box also had a ball-and-socket joint, so that the tube could be adjusted in its position in the pipe line. With the stuffing box a traverse of the pipe could have been made, but as time did not permit, the tube was kept always in the center of the pipe.

8 To determine the velocity head from the pitot tube, two forms of manometers were used. One was the familiar form of compound fluid manometer with water as the heavier and olive oil as the lighter fluid. As constructed, and by using the above fluids, the readings were multiplied about ten times. In addition an inclined gage of the ordinary draft-reading type was used that could be set at different angles, to give any multiplier. In its lowest position the readings were multiplied about forty times.

9 After the air passed the pitot tube it flowed through a short 2-in. pipe and then through a venturi tube. A cross-section of the tube used for the first tests is shown in Fig. 4, manufactured by the Builders Iron Foundry. Just before the air passed into the venturi

tube its temperature was taken by a thermometer in a well tapped into the pipe. An ordinary differential manometer determined the difference of pressure between the mouth and throat of the venturi tube. A pressure gage was attached, as shown, from which the total pressure of the air at the mouth was obtained.

10 After flowing through the venturi tube, the air continued through a 2-in. pipe and then through a square-cornered orifice in a thin plate inserted in the line between two standard pipe flanges. The pressure was obtained on each side of the orifice in two places. One gage was tapped into the side of the pipe 6 in. from the orifice, while the other gage was tapped into the flange on an angle, so that the pressure was obtained right at the face of the orifice. Fig. 5 shows the details of these connections. Fig. 6 shows the different orifices that were used during the tests. These were made by drilling a brass plate and not rounding the corners except to remove the burr formed by the drilling. After leaving the orifice the air passed through the valve *H* which was used for adjusting the final pressure.

11 Thence the air flowed through a 3-in. pipe to a three-way cock, by which it could be discharged into either of two gasometer tanks, each of about one hundred cubic feet volume. Thus an absolute measure of the volume of air flowing was obtained, the rise of each tank being determined by a scale reading to tenths of inches. The pressure of the air in the tanks, which varied from 2 to 4 in. of water, was obtained from a water manometer on top of each tank. The air temperature in the gasometer tanks was determined by thermometers inserted through their tops.

12 As the tests were very short and the air in the tanks was kept in motion by the inflowing air, it is believed that representative temperature readings were obtained. Through large quick-opening valves on top of the tanks they were emptied after the readings were taken. It was at first thought desirable to provide the inside tanks with counterweights to reduce the pressure inside of them, and thus prevent the final pressure on the discharge side of the orifice from exceeding 14.7 lb. per sq. in. abs., but the apparatus being at an elevation of about eight hundred feet above sea level, where the atmospheric pressure rarely exceeds 14.5 lb., it was found that counterweights were unnecessary.

13 The arrangement of the apparatus for tests Nos. 80 to 185 was somewhat different. After passing through the throttle valve *A* the air passed through a 1½-in. pipe and then past the same Taylor pitot tube as in the first tests. This time the pitot tube was in the

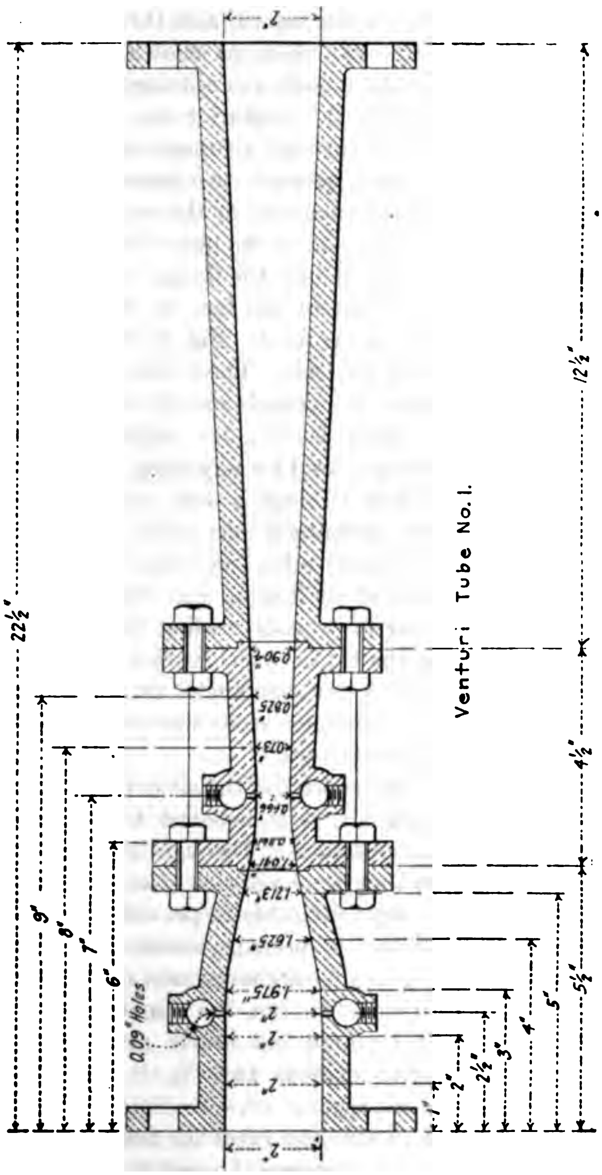


FIG. 4 2 × 0.666-IN. VENTURI TUBE

center of the 1½-in. pipe. The same manometers were used as before for obtaining the velocity head.

14 Beyond the pitot tube the air passed a thermometer registering its temperature, and then through a 1 × ⅝-in. venturi tube illustrated in Fig. 7. This tube is in one piece and not of exactly the

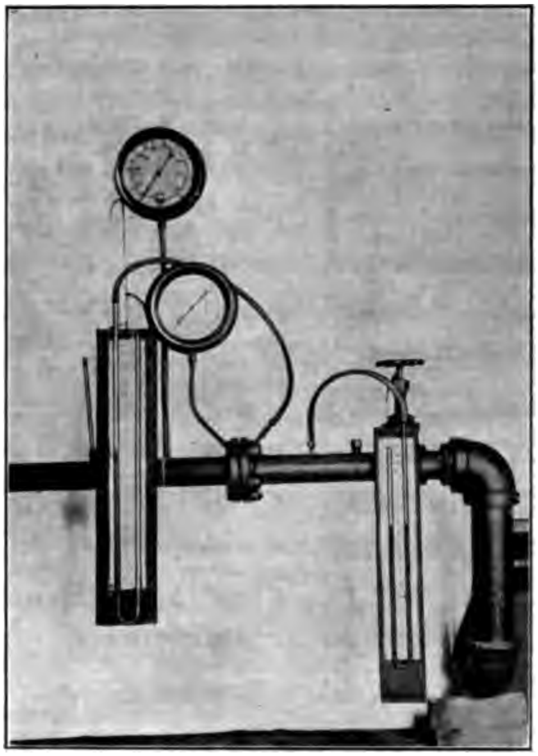


Fig. 5 Arrangement of Gages at Orifice

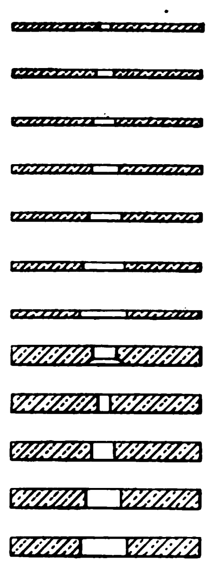
same form as the larger tube used before. From this point on, the layout of the apparatus was the same as for the other tests.

DESCRIPTION OF THE TESTS

15 After the apparatus was set up and all piping completed it was tested under pressure for leaks and all that could be found were stopped.

16 As it was impossible to make the gasometer tanks absolutely air-tight, a leakage test was run on each tank to determine the neces-

sary correction for leakage. These tests were made by filling the tanks with air and then noting the time of fall. It took 45 hr. for the air to leak out of one tank, while the other was still more than half full. The pressure inside the tanks was from 2 to 4 in. of water above the atmosphere, it being at the higher value when the inside tanks were in the highest position, and at the lower value when the



Size	Used on Tests Nos.
$\frac{1}{8}'' \times \frac{1}{16}''$	1 to 11
$\frac{3}{16}'' \times \frac{1}{16}''$	12 to 22
$\frac{1}{4}'' \times \frac{1}{16}''$	23 to 34, 113 to 157
$\frac{5}{16}'' \times \frac{1}{16}''$	35 to 44
$\frac{3}{8}'' \times \frac{1}{16}''$	45 to 53
$\frac{7}{16}'' \times \frac{1}{16}''$	62 to 68
$\frac{1}{2}'' \times \frac{1}{16}''$	74 to 79
$\frac{1}{4}'' \times \frac{1}{8}''$	80 to 90, 102 to 112
$\frac{1}{8}'' \times \frac{3}{16}''$	158 to 168
$\frac{1}{4}'' \times \frac{3}{16}''$	91 to 101
$\frac{3}{8}'' \times \frac{3}{16}''$	169 to 177
$\frac{1}{2}'' \times \frac{3}{16}''$	178 to 185

Fig. 6 Orifices Used During Air Tests

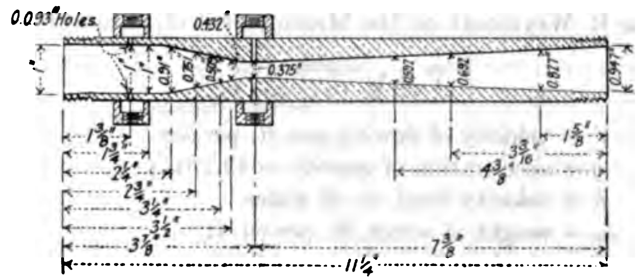
tanks were only a short distance from the lowest position. As the average time of filling each tank during the tests was only about one minute, it will be seen that the leakage correction was very small.

17 The actual volume of each gasometer was computed from the average diameter found by measuring the diameter of the inside tank at a great many points (four diameters at every foot throughout the length).

18 The compound fluid manometer and the inclined manometer used in connection with the pitot tube were calibrated by connecting to a U-tube manometer and taking readings at several points. All pressure gages were calibrated by a dead-weight gage tester. All thermometers were calibrated by reference to a standard thermometer.

19 The tests were run as follows: An orifice was selected and placed in the line; both gasometer tanks were emptied, the exhaust valve on one tank was left open while that on the other was closed, the three-way valve was then turned so that the air could enter the gasometer that was open to the atmosphere; the throttle valve *A*, Fig. 1, was then opened until the gage *C* indicated a pressure slightly in excess of that desired; the pressure was then adjusted by the valve *E* and gage 37; then the three-way valve was quickly turned so that the air discharged into the gasometer that was closed. When this gasometer was full the three-way valve was turned back to its original position so that the air flowed into the other gasometer, which in the meantime had been closed.

20 The rise of the first tank and the pressure and temperature of the air in it were then recorded while the second tank was filling.



Venturi Tube No. 2.

FIG. 7 1 × ⅜-IN. VENTURI TUBE

After the readings had been taken the exhaust valve was opened allowing the tank to empty. After a length of time equal to that taken to fill the first gasometer, the three-way valve was quickly turned, so that the air could flow to the atmosphere through the first gasometer. The rise, temperature, and pressure were then recorded for the second gasometer. The pressure in the pipe line was maintained constant during the entire operation. The various gages and thermometers along the pipe line were read every 5 min. during the long tests, and as often as possible during the short tests.

21 Tests were made on each orifice at pressures of 10, 15, 20, 30, etc., up to 100 lb. per sq. in. gage. During most of the tests the final pressure on the discharge side of the orifice was kept constant at 14.7 lb. per sq. in. abs. However, a few tests were run with final pressures as high as 90 lb. per sq. in. gage. If during any test the

rise of the two gasometer tanks did not check each other within one inch, the test was repeated.

22 To test the pitot and venturi tubes for large volumes and low pressures, a few tests were run without any orifice in the line. The flow during these tests was adjusted by the back-pressure valve H.

23 During most of the tests water was the fluid used in the venturi manometer, but during some the readings were so high that it was necessary to use mercury.

COMPUTATIONS

24 All of the computations are simple and need no explanation. However, the equations used for the pitot tube, venturi tube, and the orifice will be given.

25 The equation for the pitot tube was taken from a paper by Thomas R. Weymouth on the Measurement of Natural Gas:[1]

$$V = \sqrt{\frac{2\,gh\delta_w}{12\,\delta_a}\,\frac{14.7\,T}{492\,PG}}$$

where V = velocity of flowing gas, ft. per sec.
g = acceleration of gravity = 32.2 ft. per sec.
h = velocity head, in. of water
δ_w = weight of water, lb. per cu. ft.
T = absolute temperature of flowing gas, deg. fahr.
δ_a = weight of 1 cu. ft. of air at 32 deg. fahr. and 14.7 lb. per sq. in. abs., = 0.08073 lb.
P = absolute pressure of flowing gas, lb. per sq. in.
G = specific gravity of flowing gas, air = 1.000.

26 The equation which was used for the venturi was taken from the same paper by Weymouth and is as follows:

$$Q = \frac{24.4\,A_2T_s}{P_s}\sqrt{\frac{n}{(n-1)\,G}}\,\frac{P_1}{\sqrt{T_1}}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}}$$

where Q = quantity of air per minute at 32 deg. and 14.7 lb. abs., cu. ft.
A_2 = area of throat, sq. in.
A_1 = area of entrance, sq. in.
T_s = absolute temperature of standard air = 492 deg. fahr.

[1] Trans. Am.Soc.M.E., vol. 34, p. 1639.

P_s = absolute pressure of standard air = 14.7 lb. per sq. in.
n = a constant, for air = 1.402
G = specific gravity of gas, air = 1.000
P_1 = pressure of air at entrance, lb. per sq. in. abs.
P_2 = pressure of air in throat, lb. per sq. in. abs.
T_1 = temperature of air at entrance, deg. fahr. abs.

27 The equation which was used for the orifice was taken from Hirshfeld and Barnard's Elements of Heat Power Engineering, and is as follows:

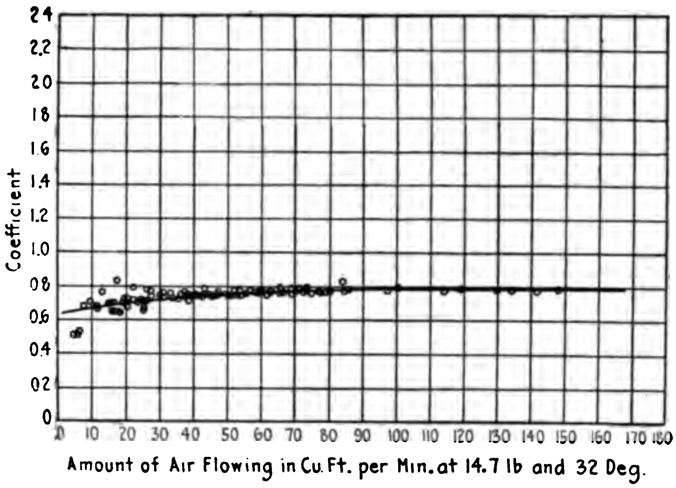


FIG. 8 COEFFICIENT OF TAYLOR PITOT TUBE IN A 1½-IN. PIPE

$$Q = \frac{1522\, P_1 A}{\sqrt{T_1}} \sqrt{\left(\frac{P_2}{P_1}\right)^{1.42} - \left(\frac{P_2}{P_1}\right)^{1.71}}$$

where Q = quantity of air per minute at 32 deg. and 14.7 lb. abs., cu. ft.
P_1 = initial pressure, lb. per sq. in. abs.
P_2 = final pressure, lb. per sq. in. abs. When P_2 is less than 0.527 P_1 substitute 0.527 P_1 for P_2.
A = area of orifice, sq. in.
T_1 = initial temperature of air, deg. fahr. abs.

28 Using the foregoing three theoretical equations, coefficients were computed for the pitot tube, venturi tube, and the orifices. The actual volume of the air flowing during the tests was computed from the known dimensions of the gasometers. Corrections were made for temperature, pressure, leakage, zero reading of scale, etc.

PITOT AND VENTURI TUBES

29 Fig. 8 gives a curve of coefficients for the Taylor pitot tube when in the center of a 1¼-in. pipe. The coefficients were computed using the readings from the inclined gage, as it was impossible to obtain satisfactory readings from the compound fluid gage. This curve shows that below 30 cu. ft. per min. the results are not very reliable and that the coefficient increases with increase in velocity until at about 90 cu. ft. per min., after which it is constant.

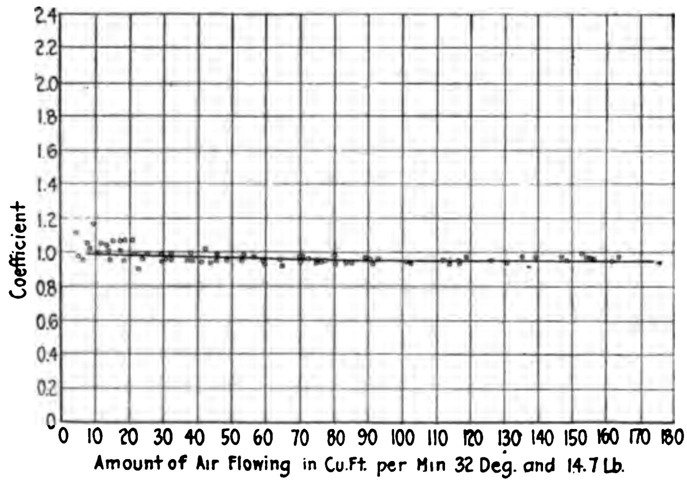


FIG. 9 COEFFICIENT OF THE 2 × 0.665-IN. VENTURI TUBE

30 A similar curve, not shown, was plotted of coefficients for the same tube when in the center of a 2-in. pipe, the velocity head being obtained with the inclined gage as before. This figure showed that the pitot tube is uncertain when used in a 2-in. pipe if the amount of air flowing is less than 40 or 50 cu. ft. per min., while the coefficient becomes fairly steady when the discharge is greater than 80 cu. ft. per min.

31 The curve in Fig. 9 of coefficients for the 2 × 0.665-in. venturi tube shows that the average is about 0.95. It also shows that this sized venturi tube is not accurate when the flow is less than 25 cu. ft. of free air per minute, and that the coefficient decreases slightly as the flow increases.

32 The curve of coefficients for the 1 × ⅜-in. venturi tube is shown in Fig. 10. This curve differs slightly from that for the larger

tube, and the average coefficient is higher, being about 0.99 over most of the range of the tests. The reason for this difference in the curves and the coefficients is that the converging section of the larger tube continues up to where the throat pressure is taken. (See Figs. 4 and 7.) This tends to contract the stream at the throat, and so to give a lower pressure reading here than should be. This causes the reading to be high. In the smaller tube the converging section stops before the point where the pressure is taken so that the stream completely fills the throat at that point. The contraction effect is greater at the high velocities, which explains the decrease in the coefficient at the high rates of flow.

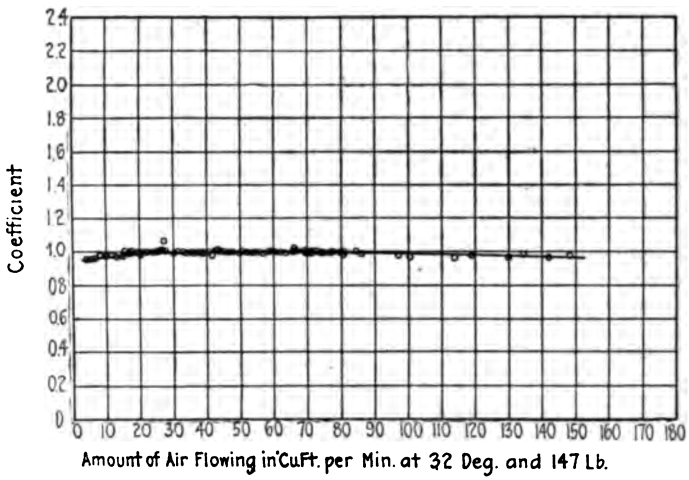


Fig. 10 Coefficient of the 1 × $\frac{3}{8}$-in. Venturi Tube

ORIFICES

33 A number of tests were run on various-sized orifices from $\frac{1}{8}$ to $\frac{1}{2}$ in., increasing in size by $\frac{1}{16}$ in. These orifices were all formed in $\frac{1}{16}$-in. brass plate. The actual results of these tests are shown in Fig. 11.

34 As has been mentioned the pressure was taken near the orifice at two points. However, it was impossible to detect any difference between the pressures taken at these two points with the gages used, which could be read to the nearest pound, the smallest division on the scale being 5 lb.

35 It was impossible to keep the temperature of the air constant

during the tests, so the actual discharge has been corrected to 32 deg. fahr. by multiplying the actual discharge by the factor $\sqrt{\frac{T}{492}}$, in which T is the absolute temperature of the air flowing.

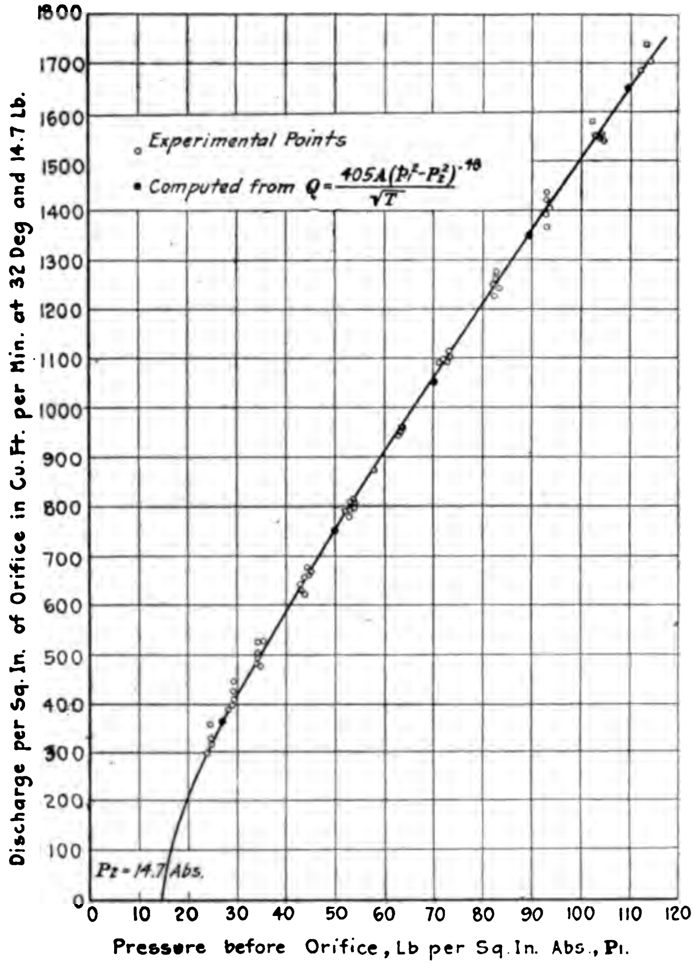


FIG. 11 INITIAL PRESSURE-QUANTITY CURVE. FINAL PRESSURE = 14.7 LB.

36 Judging from the results the size of the orifice has no effect upon the coefficient of discharge.

37 The open circles in Fig. 11 are the experimental points obtained from all the orifices in the $\frac{1}{16}$-in. plates The solid circles will be mentioned later.

38 To compare orifices of the same size but in different thicknesses of plate, tests were run on three orifices ¼ in. in diameter. The first was in $\frac{1}{16}$-in. plate, the second in ⅛-in. plate, and the third was in

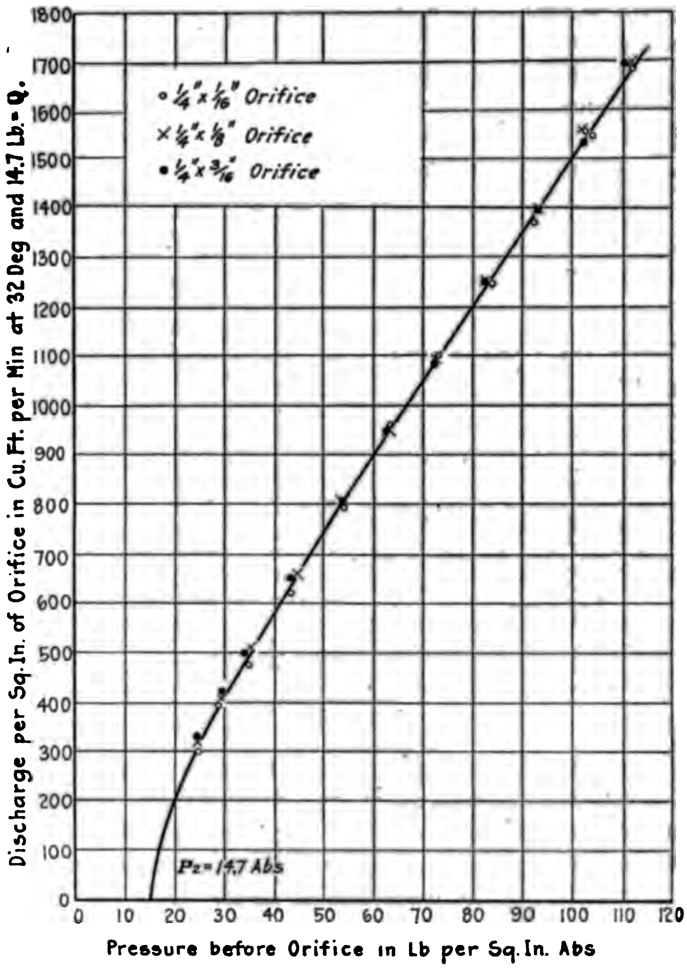


FIG. 12 INITIAL PRESSURE-QUANTITY CURVE. ORIFICES OF DIFFERENT THICKNESSES

$\frac{3}{16}$-in. plate. The curve and points in Fig. 12 show that very nearly the same result was obtained in each case.

39 The curve at the extreme right in Fig. 13 shows how the actual discharge varies with the ratio of pressure at the orifice, when the final pressure is constant at 14.7 lb. per sq. in. abs.

40 A number of tests were run on the $\frac{1}{4} \times \frac{1}{16}$-in. orifice with various back pressures from atmospheric pressure up to about ninety pounds gage. The results of these tests are shown in Fig. 14. The curves in Fig. 15 are cross-plotted from Fig. 14 for various constant quantities.

41 The curves in Fig. 16 are also cross-plotted from Fig. 14. To make an interesting comparison the ordinates in this case were multiplied by the constant 0.661. By doing this the intercept of any one curve on the ordinate will be at the same distance from the origin as the intercept of the same curve on the abscissa. The same result

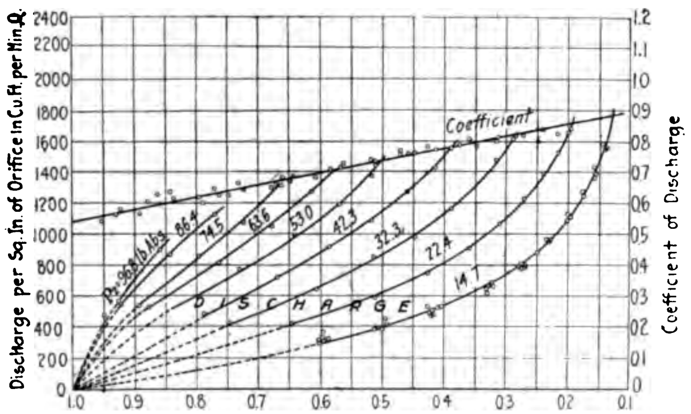


Fig. 13 Coefficient of Discharge, and Discharge for Orifices in $\frac{1}{16}$-in. Plate

could have been accomplished by preparing special cross-section paper, having the divisions along the ordinates equal to 0.661 of the divisions along the abscissæ. The object of plotting these curves this way was to compare the curves obtained with arcs of circles. The dotted curves in the figure are arcs of circles, while the full curves are the results of the actual experiments.

42 Fig. 13 also shows the coefficient of discharge and the discharge plotted against the ratio of pressures at the orifice, for the tests when the final pressure was greater than atmospheric pressure.

FLOW OF AIR THROUGH A THIN-PLATE ORIFICE

43 For a given orifice, and a given temperature, the flow of air involves three variables — the quantity discharged, the initial pressure, and the final pressure. Evidently then the geometric figure that

illustrates the relation of the three variables is a curved surface involving all three dimensions. Examination of the various curves

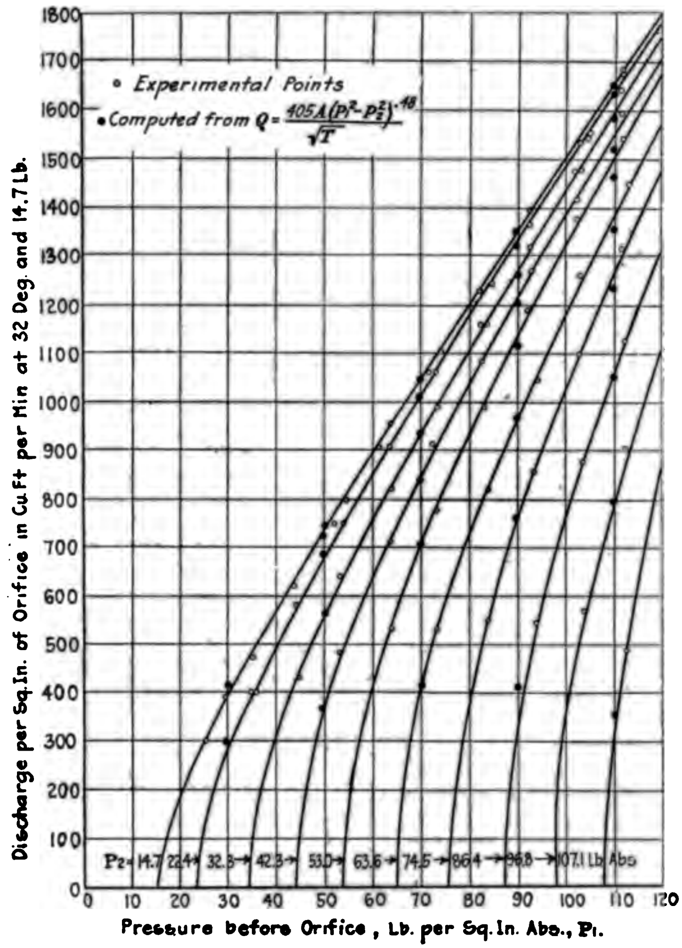


FIG. 14 INITIAL PRESSURE-QUANTITY CURVES. VARIOUS FINAL PRESSURES

discussed above shows them to be conic sections. The equation of a cone whose axis coincides with the z-axis, is:

$$m^2 (x^2 + y^2) = (z - c)^2$$

where m is the slope of the surface of the cone with the z axis and c is the intercept on the z-axis. For a special case when the apex of the cone is at the origin, $c = 0$, and when the surface of the cone is at an angle of 45 deg. with the axis, $m = 1$. In our case $c = 0$, $m = 1$, $x = P_2$, $y = Q$, and $z = P_1$; then, by substitution,

$$Q^2 = K'(P_1^2 - P_2^2)$$

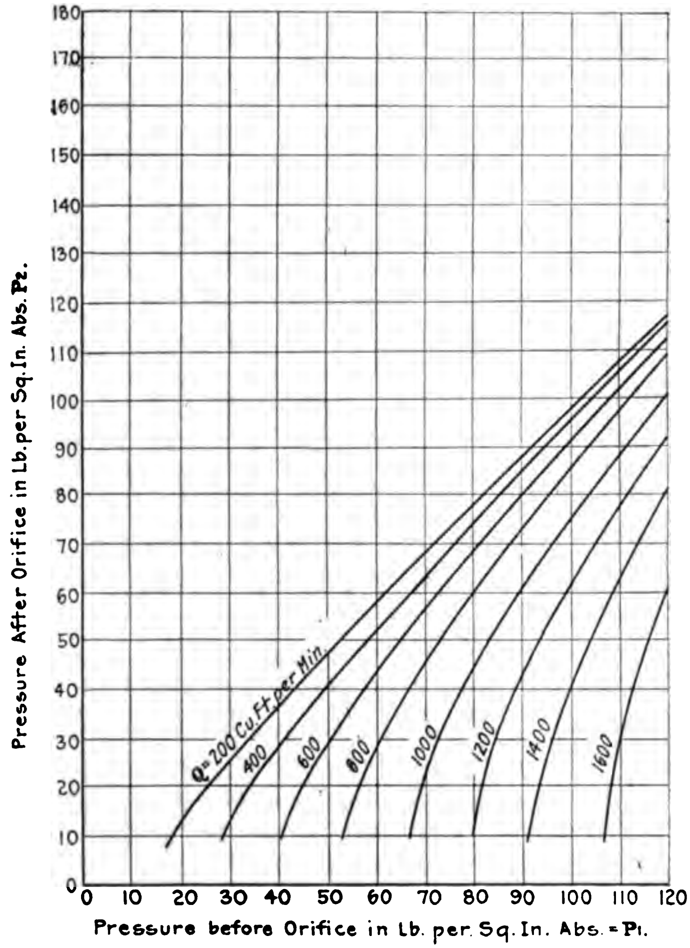


FIG. 15 INITIAL PRESSURE-FINAL PRESSURE CURVES. VARIOUS CONSTANT QUANTITIES

or, after inserting the temperature factor, the general form of the equation is

$$Q = \frac{KA\,(P_1^2 - P_2^2)^n}{\sqrt{T}}$$

44 To obtain the numerical values of K and n, values of $(P_1^2 - P_2^2)$ were computed from the results of the tests and plotted against the quantity Q in Fig. 17. From this logarithmic curve it

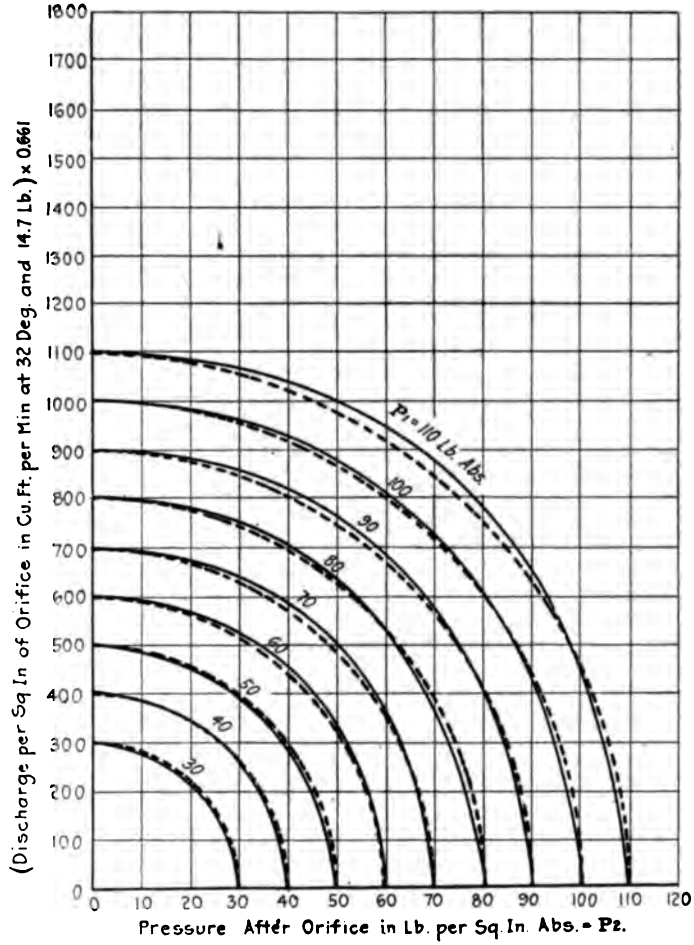


FIG. 16 FINAL PRESSURE-QUANTITY CURVES. VARIOUS CONSTANT INITIAL PRESSURES

was found that $K = 405$ and $n = 0.48$. Thus the final equation is

$$Q = \frac{405\, A\, (P_1^2 - P_2^2)^{0.48}}{\sqrt{T}}$$

where Q = quantity of air discharged per min. at 32 deg. fahr. and 14.7 lb. per sq. in. abs., cu. ft.

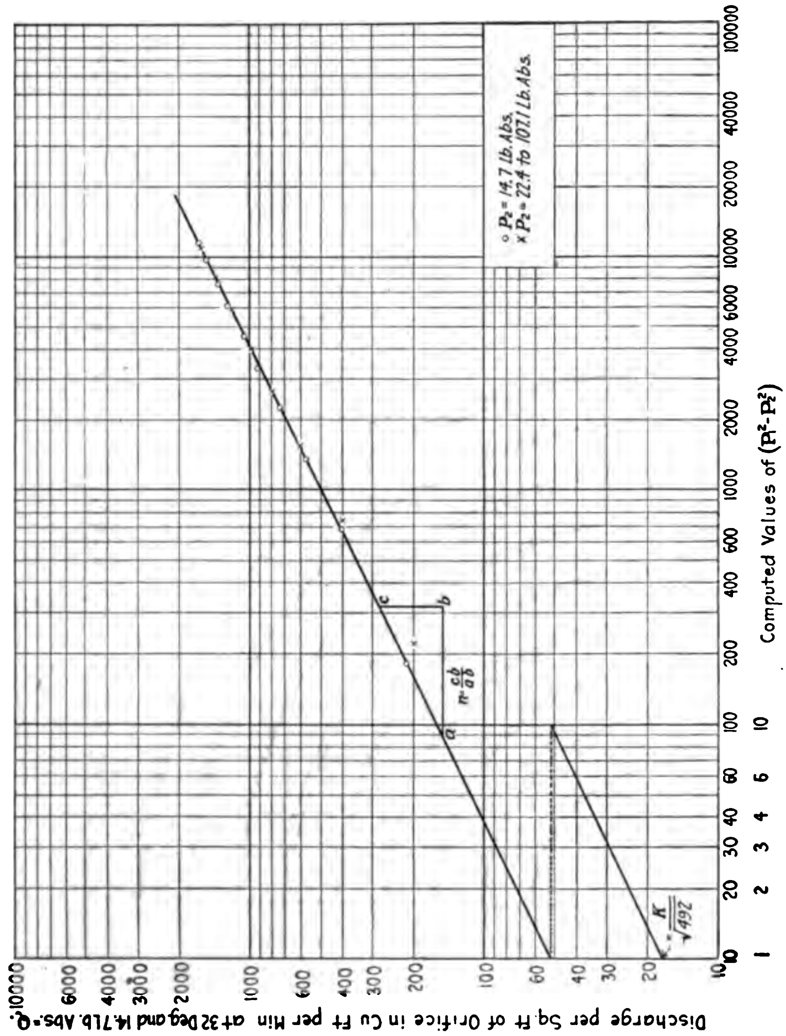


FIG. 17 QUANTITY-$(P_1^2 - P_2^2)$ CURVES

A = area of the orifice, sq. in.
P_1 = initial pressure before orifice, lb. per sq. in. abs.
P_2 = final pressure after orifice, lb. per sq. in. abs.
T = absolute temperature of air entering orifice, deg. fahr.

45 To check the accuracy of this equation values were computed for various pressures. These are plotted in Figs. 11 and 14 and are shown by the solid circles. It will be seen that these points coincide closely with the experimental curves.

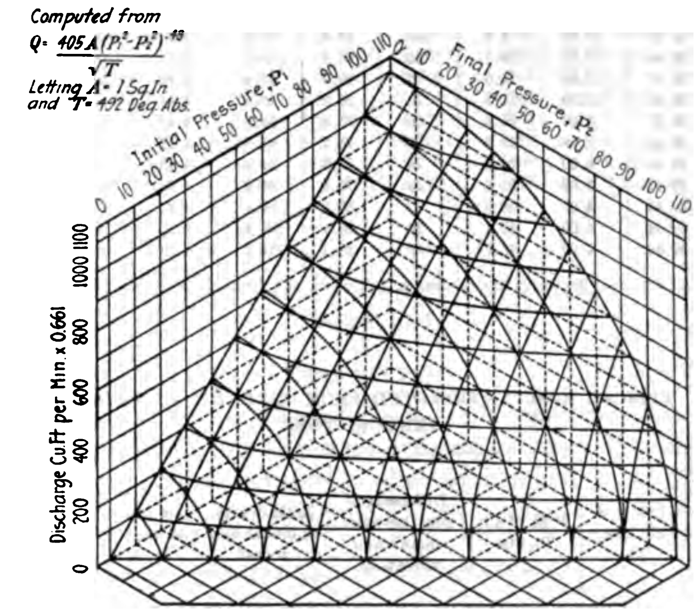


Fig. 18 Axonometric Chart for the Flow of Air through Thin-Plate Square-Corner Orifices

46 To show the geometric shape of the curves when plotted in three dimensions, the axonometric chart shown in Fig. 18 was computed from the above equation.

TYPICAL RESULTS SELECTED FROM TESTS ON THE TAYLOR PITOT TUBE IN A 2-IN. PIPE

No.	General Data				Pitot Tube				
	Barometer, in. Hg at 32 deg. fahr.	Length of test in minutes	Cu. ft. of air per min. at 32 deg., 14.7 lb. abs.	Temp. of room, deg. fahr.	Temp. of air, deg. fahr.	Pres. of air, lb. abs.	Actual velocity in pipe, ft. per sec.	Velocity head, in. water	Coefficient of tube
12	29.44	20.00	8.6	65	64.4	25.4	3.82	0.017	0.584
13	29.44	16.00	10.9	66	65.1	29.9	4.12	0.021	0.608
14	29.47	12.00	14.0	70	69.9	35.4	4.48	0.013	0.923
15	29.47	10.00	17.8	70	70.0	45.4	4.48	0.021	0.818
16	29.47	8.00	21.5	70	70.6	54.8	4.49	0.018	0.957
17	29.44	6.00	25.3	70	70.7	64.5	4.49	0.021	0.963
18	29.44	6.00	29.1	70	71.5	74.4	4.48	0.030	0.868
19	29.44	5.92	32.5	71	73.2	84.4	4.44	0.024	1.018
20	29.44	4.00	38.0	57	56.7	94.4	4.47	0.028	1.037
21	29.44	4.00	42.1	59	58.1	103.4	4.55	0.022	1.225
22	29.44	4.00	45.7	60	61.1	114.1	4.51	0.060	0.775

TYPICAL RESULTS SELECTED FROM TESTS ON A 2 × 0.666-IN. VENTURI TUBE

No.	Temp. of air, deg. fahr.	Pres. of air, lb. abs.	Difference of pres. between mouth and throat, lb.	Ratio of pres. at mouth and throat	Cu. ft. of air per min. at 32 deg., 14.7 lb., computed	Coefficient of tube
12	64.4	24.42	0.01980	0.99919	8.4	1.017
13	65.1	29.52	0.02670	0.99909	10.9	0.999
14	69.9	34.73	0.03680	0.99894	13.9	1.009
15	70.0	44.43	0.04690	0.99894	17.7	1.002
16	70.6	54.43	0.0567	0.99895	21.6	0.997
17	70.7	64.23	0.0664	0.99896	25.5	0.993
18	71.5	73.92	0.0761	0.99897	29.2	0.998
19	73.2	83.82	0.0867	0.99897	33.0	0.986
20	56.7	94.12	0.0978	0.99896	38.0	0.998
21	58.1	104.22	0.1118	0.99893	42.6	1.011
22	61.0	114.13	0.1230	0.99892	46.6	0.982

TYPICAL RESULTS SELECTED FROM TESTS ON AIR ORIFICES

No.	Size of orifice, inches	Temp. of air entering orifice, deg. fahr.	Initial pres. of air, lb. abs.	Final pres. of air, lb. abs.	Ratio of pres. at orifice	Actual discharge per sq. in. of orifice, cu. ft. per min. at 32 deg. and 14.7 lb.	Coefficient of discharge	Discharge per sq. in. of orifice corrected to 32 deg. fahr.
12	$\frac{3}{16}$ by $\frac{1}{16}$	64.3	24.7	14.7	0.598	311	0.713	321
13	$\frac{3}{16}$ by $\frac{1}{16}$	65.0	29.5	14.6	0.498	394	0.774	426
14	$\frac{3}{16}$ by $\frac{1}{16}$	69.3	36.0	14.6	0.411	507	0.822	526
15	$\frac{3}{16}$ by $\frac{1}{16}$	69.7	45.4	14.6	0.322	644	0.820	668
16	$\frac{3}{16}$ by $\frac{1}{16}$	70.1	54.4	14.6	0.268	778	0.826	807
17	$\frac{3}{16}$ by $\frac{1}{16}$	70.3	63.9	14.6	0.230	916	0.834	951
18	$\frac{3}{16}$ by $\frac{1}{16}$	71.0	73.9	14.6	0.199	1052	0.830	1092
19	$\frac{3}{16}$ by $\frac{1}{16}$	73.0	82.9	14.6	0.176	1178	0.827	1227
20	$\frac{3}{16}$ by $\frac{1}{16}$	57.0	93.9	14.6	0.155	1373	0.830	1406
21	$\frac{3}{16}$ by $\frac{1}{16}$	58.0	104.7	14.6	0.139	1522	0.829	1559
22	$\frac{3}{16}$ by $\frac{1}{16}$	60.4	114.0	14.6	0.127	1658	0.830	1704

Part II The Flow of Steam through Orifices at Low Pressures

INTRODUCTION

47 Many investigations have been made relative to the flow of steam through orifices at high pressures, but very few data are available for such flow under pressures slightly over and under atmospheric. The purpose of this investigation is to collect data of the latter kind.

DESCRIPTION OF APPARATUS

48 Fig. 19 shows the arrangement of the apparatus as set up in the mechanical laboratories of Sibley College, Cornell University. Steam was taken from a steam main through about thirty feet of 3-in. pipe, thence through the valve *B*, the separator, the valve *C*, and the orifice, which was placed between two standard pipe flanges as shown. After passing through the orifice the steam continued through the valve *D* into the surface condenser shown at the right lower corner in Fig. 19. The general appearance of the apparatus is shown in Fig. 20.

49 The condensate was pumped from the condenser into weighing tanks by either of the two wet-vacuum pumps shown. The small pump was used while the small orifices were being tested; the large one during the tests on the larger orifices.

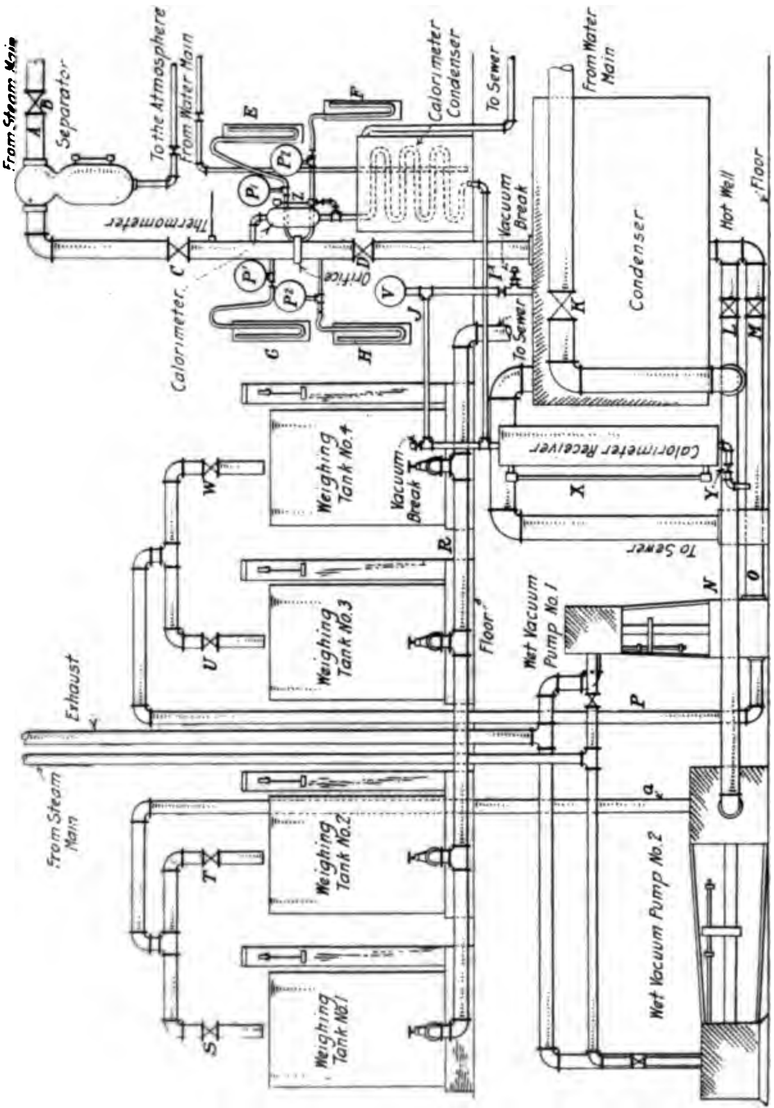


FIG. 19 LAYOUT OF APPARATUS FOR STEAM TESTS

50 To make sure no superheat was in the steam before it passed through the orifice, a thermometer was inserted into the steam pipe as shown.

51 The quality of the steam was determined by a separating calorimeter. The exhaust from the calorimeter was condensed in a small surface condenser and the condensate passed into a receiver provided with a gage glass X, to show the quantity of condensate. Since, during some of the tests, the pressure above the orifice was below atmospheric, the receiver was connected to the large condenser

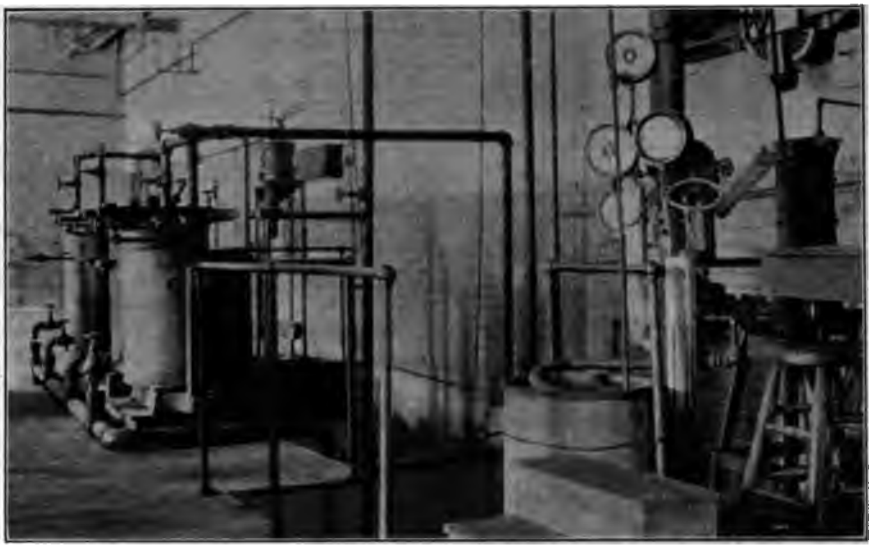


Fig. 20 General View of Apparatus Used on Steam-Orifice Tests

by the small pipe J. Thus a flow of steam could always be maintained through the calorimeter.

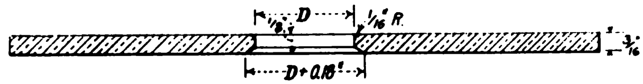
52 Pressure connections were inserted at two points both above and below the orifice as shown. One was inserted through the side of the pipe 6 in. above the orifice, while another was inserted through the flange on an angle so that the pressure was obtained right at the face of the orifice. Similar connections were placed below the orifice.

53 Mercury manometers were first used for obtaining the pressures; but, so much trouble resulted from water collecting in the connections and over the mercury, that spring gages were substituted for the manometers. These could be read easily to 0.1 in. on the vacuum side and 0.1 lb. on the pressure side. Compound gages

were used above the orifice, while simple vacuum gages were used below it. Because spring gages do not give accurate readings near atmospheric pressure, mercury manometers were used when the pressure was near or at that of the atmosphere.

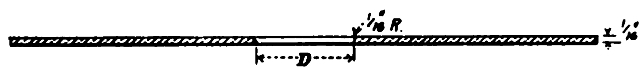
54 Seven different sizes of orifices were used, ranging from 1/2 to 2 in. diameter, and two different thicknesses of plate as shown in Fig. 21, and the corners of the orifices were rounded off with a 1/16-in. radius.

Nominal Size	Actual Diameter, In.	Area, Sq. In.	Used on Tests Nos.
	Orifice in 3/16-in. Brass Plate		



Nominal Size	Actual Diameter, In.	Area, Sq. In.	Used on Tests Nos.
1/2" × 3/16"	0.4985	0.1952	1 to 29
3/4" × 3/16"	0.7500	0.4418	30 to 57
1" × 3/16"	0.9995	0.7846	58 to 86
1 1/4" × 3/16"	1.2460	1.2194	87 to 117
1 1/2" × 3/16"	1.4970	1.7601	118 to 151
1 3/4" × 3/16"	1.7510	2.4080	242 to 272
2" × 3/16"	2.0010	3.1447	273 to 302

Orifice in 1/16-in. Brass Plate



Nominal Size	Actual Diameter, In.	Area, Sq. In.	Used on Tests Nos.
1/2" × 1/16"	0.5000	0.1963	152 to 181
3/4" × 1/16"	0.7515	0.4435	397 to 428
1" × 1/16"	1.0025	0.7893	365 to 396
1 1/4" × 1/16"	1.2490	1.2252	212 to 241
1 1/2" × 1/16"	1.5005	1.7683	182 to 211
1 3/4" × 1/16"	1.7530	2.4136	333 to 364
2" × 1/16"	2.0000	3.1416	303 to 332

FIG. 21 ORIFICES USED DURING TESTS

DESCRIPTION OF THE TESTS

55 Before the tests were started all gages were calibrated, all weighing scales checked, and the calorimeter receiver calibrated. The gages and scales were also checked from time to time throughout the tests.

56 An orifice was selected and placed in the line. Steam was then turned on through the valve B so that the pressure above the orifice was slightly above that desired. The exact pressure was then

obtained by adjusting the drain valve on the separator. The valve *C* was not used much because, when the steam was throttled at this point, superheated steam was obtained at the orifice. The desired back pressure on the orifice was obtained through the valve *D*.

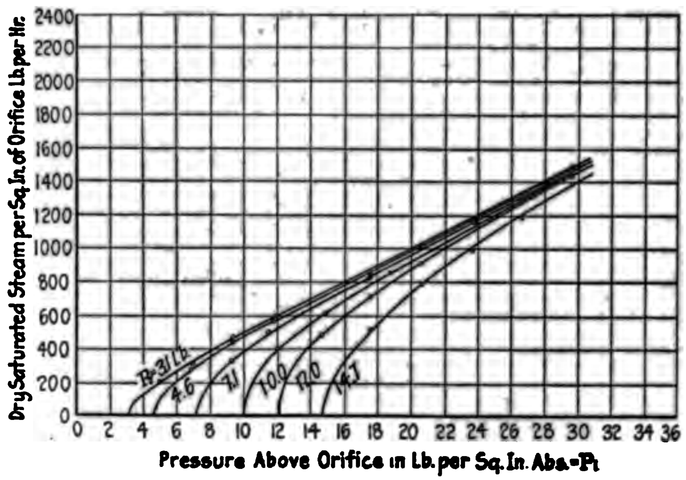


FIG. 22 ABSOLUTE PRESSURE-QUANTITY CURVES. $1\frac{3}{4} \times \frac{1}{16}$-IN. ORIFICE

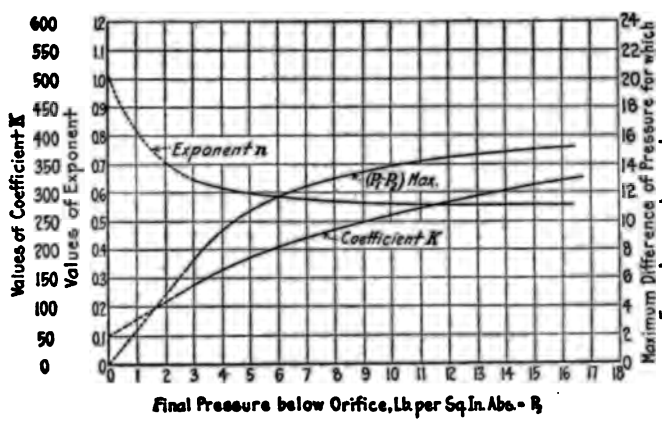


FIG. 23 VALUES OF K, n AND $(P_1 - P_2)$ MAX. FOR USE IN $W = KA\,(P_1 - P_2)^n$

57 Tests were run with varying initial pressure and with various constant final pressures. Calorimeter readings were taken at the beginning and at the end of each test. All other readings were taken every 5 min. The weight of steam discharged was obtained by the

usual two-tank method of filling one weighing tank while the other is discharging, and so on alternately.

COMPUTATIONS

58 The computations are all simple and require no explanation except the one for the quantity of dry steam. The correction for the

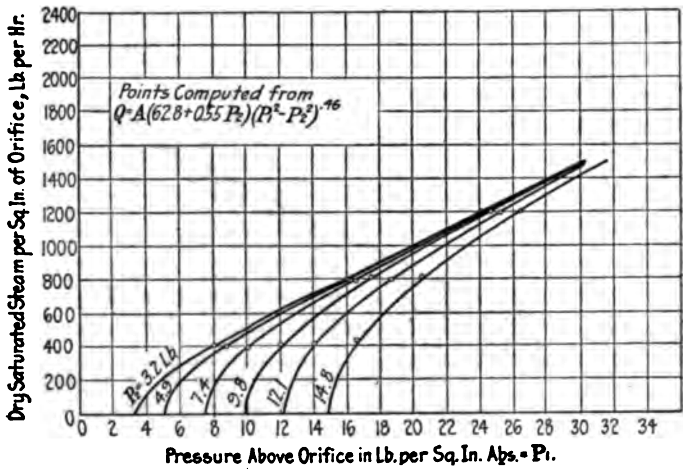


FIG. 24 INITIAL PRESSURE-QUANTITY CURVES. AVERAGE OF ALL ORIFICES

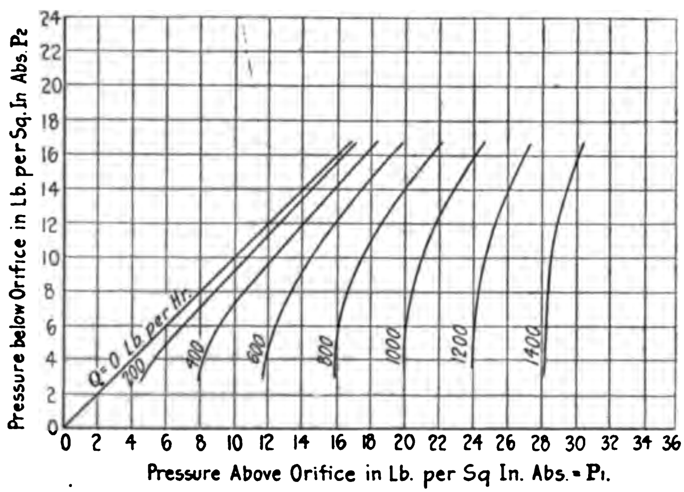


FIG. 25 INITIAL PRESSURE-FINAL PRESSURE CURVES. AVERAGE OF ALL ORIFICES

moisture in the steam is based on the assumption that the quantity of steam flowing increases directly with the moisture, that is, the total water pumped from the condenser is multiplied by the percentage of dry steam to obtain the correct quantity of dry steam passing through the orifice. This is not exactly correct, as the flow of steam

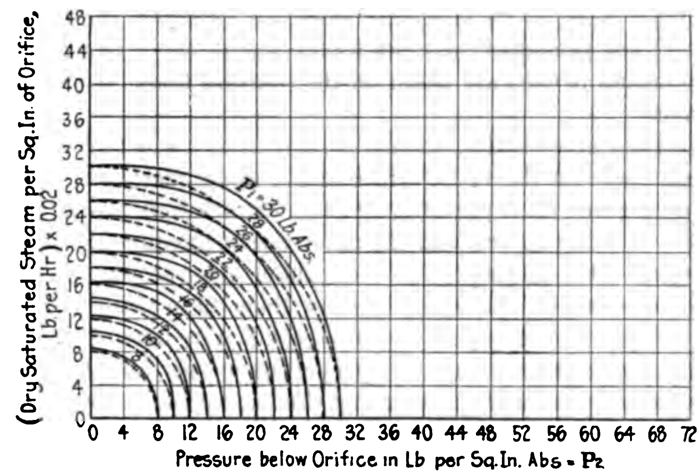


FIG. 26 FINAL PRESSURE-QUANTITY CURVES. AVERAGE OF ALL ORIFICES

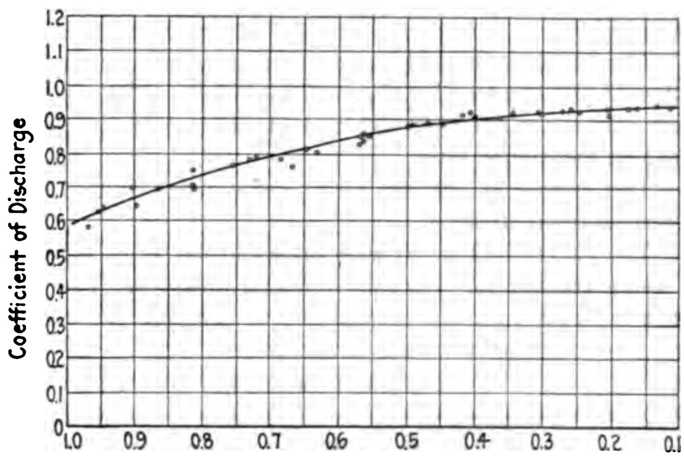


FIG. 27 COEFFICIENT OF DISCHARGE. AVERAGE OF ALL ORIFICES

through an orifice does not increase exactly as the quantity of moisture in the steam; however, since the average quantity of moisture in the steam during the tests is only about one per cent, the error is very small.

CURVES

59 Examining the numerical results in the tables will show that the pressures taken 6 in. before and 6 in. after the orifice do not agree with those taken at its face. As would be expected, the difference between these pressures increases as the size of the orifice increases. All the curves plotted are based on the pressure taken at the face of the orifice.

60 The curves in Fig. 22 are plotted from the actual results of the tests made on the $1\frac{3}{4}$-in. orifice in the $\frac{1}{16}$-in. plate. Each curve is for a constant back pressure. Similar curves were plotted for all the tests and also curves with the difference of pressure at the orifice as abscissæ — the latter sort on logarithmic cross-section paper, but it was not considered necessary to show them all in this report.

61 Values of the slope n and intercept K were taken from all the logarithmic curves (from the lower part of the curves), and the averages of these are plotted in Fig. 23 against the back pressures or final pressures as the abscissæ. The curve marked $(P_1 - P_2)_{max}$ refers to points of inflection of the logarithmic curves. By substituting values of K and n taken from these curves in the equation $W = KA\,(P_1 - P_2)^n$, the flow of steam through the form of orifice described can be computed.

62 The curves in Fig. 24 are the averages of all curves similar to Fig. 22, that were plotted for the various orifices. The points shown on these curves are not from experiments, but were computed from an equation mentioned later.

63 The curves in Fig. 25 were cross-plotted from Fig. 24 for various constant quantities.

64 The curves in Fig. 26 were cross-plotted from Fig. 24, each curve being plotted for a constant initial pressure. The quantity here was multiplied by a constant 0.02 to compare the curves with the arcs of circles shown by the dotted lines.

65 The coefficient of discharge was computed from the curves in Fig. 24, and is plotted against the ratio of pressure in Fig. 27.

66 The curves in Figs. 24, 25, and 26 will be seen to be conic sections, so an equation has been obtained for computing the flow of steam through the form of orifice used. The general equation of the cone is

$$m^2\,(x^2 + y^2) = (z - c)^2$$

When the apex of the cone is at the origin, $c = 0$, and when the side of the cone is at an angle of 45 deg. with the axis, $m = 1$. In our case $x = P_2$, $y = Q$, and $z = P_1$, whence

$$Q^2 = K'\,(P_1^2 - P_2^2)$$

or the general form of the equation desired is

$$Q = KA\,(P_1^2 - P_2^2)^n$$

67 Values of $(P_1^2 - P_2^2)$ and Q were computed from the average curves in Fig. 24 and were plotted in Fig. 28. From this logarithmic curve a value of 0.46 for n was decided upon. However, it was found

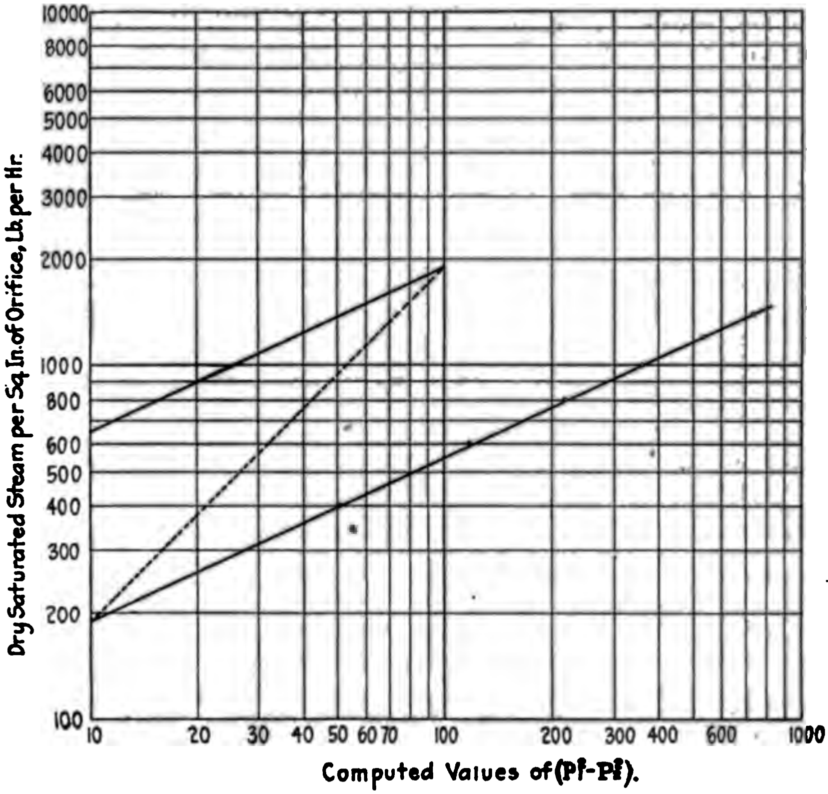


FIG. 28 $(P_1^2 - P_2^2)$-QUANTITY CURVES. AVERAGE OF ALL ORIFICES

that the value of K varied slightly with the final pressure on the orifice. Values of K were computed and plotted against the final pressure P_2. From this curve it was found that K varied as a straight line, and so a simple relation between K and P_2 was obtained. This relation expressed in the form of an equation is

$$K = (62.8 + 0.55\,P_2)$$

Thus we have for the final equation,

$$Q = A\,(62.8 + 0.55\,P_2)\,(P_1^2 - P_2^2)^{0.46}$$

where Q = quantity of dry saturated steam per hour, lb.
A = area of the orifice, sq. in.
P_1 = initial pressure on orifice, lb. per sq. in. abs.
P_2 = final pressure, lb. per sq. in. abs.

68 The above equation was checked by computing values of the discharge at various pressures and plotting them in Fig. 24. As will be seen, these computed points closely agree with the experimental curves.

TYPICAL RESULTS SELECTED FROM TESTS ON STEAM ORIFICES

No.	Size of orifice, in.	Length of test in min.	Barometer, in. of mercury at 32 deg. fahr.	Initial pres. 6 in. from orifice, lb. abs.	Initial pres. at orifice, lb. abs.	Final pres. 6 in. from orifice, lb. abs.	Final pres. at orifice, lb. abs.	Quality of steam, per cent dry	Dry steam per sq. in. of orifice, lb. per hr.
333	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	17.4	17.6	14.7	14.7	99.4	516
334	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	20.4	20.6	15.0	14.7	99.3	793
335	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	23.5	23.7	15.3	14.7	99.7	995
336	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	26.3	26.6	15.7	14.7	99.7	1182
337	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	29.1	29.6	15.8	14.7	99.1	1382
338	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	14.5	14.7	12.5	12.0	99.2	490
339	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	17.5	17.6	12.7	12.0	99.8	717
340	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	20.3	20.6	12.8	12.0	99.5	907
341	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	24.3	24.7	13.3	12.0	99.6	1182
342	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.18	29.2	29.6	14.7	12.0	99.2	1438
343	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.70	11.9	12.0	10.2	10.0	99.3	399
344	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.70	14.7	14.9	10.5	10.0	99.0	618
345	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.70	18.6	18.8	11.0	10.0	99.5	859
346	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.70	23.3	23.9	11.4	10.0	99.4	1142
347	$1\frac{3}{8}$ by $\frac{1}{16}$	15	29.70	29.4	29.8	12.1	10.0	99.3	1473
348	$1\frac{3}{8}$ by $\frac{1}{16}$	20	28.94	8.9	9.3	7.4	7.1	100.0	330
349	$1\frac{3}{8}$ by $\frac{1}{16}$	20	28.94	11.4	11.6	7.6	7.1	99.3	505
350	$1\frac{3}{8}$ by $\frac{1}{16}$	20	28.94	16.9	17.5	8.0	7.1	99.3	826
351	$1\frac{3}{8}$ by $\frac{1}{16}$	15	28.94	22.9	23.6	8.8	7.1	99.7	1160
352	$1\frac{3}{8}$ by $\frac{1}{16}$	20	28.94	28.7	29.5	10.1	7.1	99.3	1457
353	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.13	6.5	6.9	5.0	4.6	99.4	280
354	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.13	8.9	9.4	5.2	4.6	99.3	439
355	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.13	11.4	11.8	5.4	4.6	99.4	572
356	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.13	16.9	17.6	6.0	4.6	99.5	823
357	$1\frac{3}{8}$ by $\frac{1}{16}$	15	29.13	22.9	23.7	7.8	4.6	99.5	1158
358	$1\frac{3}{8}$ by $\frac{1}{16}$	15	29.13	28.8	29.6	7.6	4.6	98.2	1493
359	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.13	4.9	5.0	3.4	3.1	98.4	206
360	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.33	6.6	7.0	3.6	3.1	96.9	311
361	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.33	9.1	9.5	3.9	3.1	98.7	469
362	$1\frac{3}{8}$ by $\frac{1}{16}$	20	29.33	11.5	11.9	4.2	3.1	100.0	594
363	$1\frac{3}{8}$ by $\frac{1}{16}$	15	29.33	20.2	20.7	5.6	3.1	99.5	1015
364	$1\frac{3}{8}$ by $\frac{1}{16}$	15	29.33	29.2	29.7	6.4	3.1	9.96	1465

DISCUSSION

R. J. S. Pigott extended the thanks of the Research Committee to the author for the data in his paper, all of which he regarded as very valuable. He said that the paper would add to the data as to which was the best measuring device; but it was certain that both the disk orifice and the venturi meter offered pronounced advantages over the plain impact tube for ordinary measurements about the power plant. In connection with the practical use of the devices described, he said his own experience favored the venturi tube, provided serious eddy currents were not allowed to enter the upstream side of the tube.

The disk orifice had one advantage which might offset, in the majority of cases, its chances of somewhat smaller accuracy. Suppose a meter to be installed in a plant for the maximum load to be obtained eventually: for the venturi meter there would be spent $200 or $300 for a 10- or 12-in. line, and the readings, for the first few months, or a year, would be very low and below the accurate range of the recording instrument. The disk orifice for this purpose was particularly convenient, in that a suitable orifice for high reading at low rates of flow may be obtained. Later, as the flow increases, the disk orifice may be changed in an hour or less, when the line is shut down, and high accuracy be obtained at all times on the same recorder.

E. D. Thurston asked the author if he had had any experience in using venturi meters in parallel, not necessarily putting two on at a time, but a bank of venturi meters between two headers for use on different machines as the demand for air varied. He himself had once used four tubes in a bank and found that with three shut off there was a record on their manometers indicating a slight pressure at the throats.

Frederick N. Connet explained this by saying that if there was a dead end beyond the venturi meter, a dead end at pulsating pressure, there was a constant vibration of the air back and forward from the throat, and a slight differential would be obtained, although there was no flow.

He described some experiments made at Providence in which the air was measured in a simpler way than that described by the author. The weight of air in a tank of 104 cu. ft. capacity was determined from its pressure and temperature. For accurate pressure measurements a device similar to a gage-testing machine was used. There was an electrical connection on the table of the gage tester so that if the plunger descended about an eighth of an inch, contact was made and a bell rung. By this means the decrements of pressure of the air in its passage through the venturi meter could be measured with very much greater accuracy than by watching a large test gage, and the time intervals between each drop of 5 lb. pressure could be measured by a stop watch. The tests were made through a 2-in. venturi meter, and the duration of run was long enough to obtain a state of uniformity. The weight of air in the tank at each interval of 5 lb. drop in pressure was known with considerable accuracy.

A. G. Christie said that he had checked the results in the paper against a number of careful tests made several years ago on some Taylor tubes. The coefficient for the Taylor tube placed at the center of a 12-in. pipe was practically 0.80, and in Fig. 8, as closely as he could estimate, the author's value for a 2-in. pipe was 0.77. The similarity in the value of the two tests on entirely different sizes of pipe struck him as remarkable. In Mr. Rowse's paper in the Transtions of the Society for 1913, the coefficient for a Taylor tube placed at the center of the pipe, and using a hole on the side of the pipe for the static, was the square root of 0.80, which gave him a velocity coefficient of 0.90 instead of 0.77 for the 2-in. tube.

Another point in the paper which impressed him was the statement that below a certain number of cubic feet discharge the readings of the pitot tube were unreliable. He had checked that over, and estimated that the velocity at which the readings became unreliable was practically 30 ft. per sec. He had also checked the same thing in Mr. Rowse's work, and found he was getting results with considerable accuracy as low as 20 ft. per sec.

E. G. Bailey spoke of the swirling effect in a tube, and said that he did not think it was generally appreciated how much this effect amounted to. Cases had been found where, in the center of the pipe, at certain velocities, the velocity was only 15 per cent of what it would be expected to be with a true elliptical flow, showing the centrifugal force due to the helical motion in the pipe to be enormous. When this obtained in the throat of the venturi meter, there was nothing to stop the helical motion; in fact, its angular velocity was accelerated, giving an impinging force against the pressure openings at the throat of the tube.

The sudden restriction caused by the thin-plate orifice was very effective in killing the swirling of the flowing fluid and producing a dependable pressure difference even when the orifice was placed between flanges only a short distance beyond a bend, and at the inlet of a valve or fitting.

The thin-plate orifice was in reality a venturi, the orifice plate merely acting as a forming tool while the fluid itself formed its own venturi shape beyond the orifice plate. When the pressure connections were properly located the venturi formula applies and the orifice had the additional advantage of the downstream or throat connection being away from the active stream, and thereby obtained a true pressure and overcame the difficulty encountered in venturi tube. The highest velocity of flowing fluid at the throat of a venturi tube made it extremely difficult to secure the true pressure at this point without some impact or suction effect due to slight variations in the edges of the holes, either when new or after accumulating a little scale or sediment.

Sanford A. Moss (written). The author has executed some difficult experimental work in a very creditable manner. The coefficients for venturi meters at high densities are valuable pieces of contributory evidence to the general principle that an orifice or venturi coefficient is nearly unity.

The coefficient which the author gives for the Taylor pitot tube is the product of the ratio of average to central velocity and of the static-hole constant of the Taylor tube, which is not unity.

Some remarks may be made regarding the formulæ used. The venturi formula given is the corect general formula both for the venturi and the plate orifice. It should never be used for computation work, however, as simplified formulæ give equal accuracy with vastly less computation.

For the case of the author's venturis the differentials are exceedingly small, and the proper formula for cubic feet of standard air at 32 degrees is

$$\frac{491.5 \times 677.5\, d^2 \sqrt{[P_1 (P_1 - P_2) / T_1]}}{519.5 \times \sqrt{[1 - (A_2/A_1)^2]}}$$

For the case of the table given at the bottom of p. 820 this becomes

$$286.1 \sqrt{[P_1 (P_1 - P_2)/T_1]}$$

This formula is only to be used for the case of small differentials such as in the author's experiments.

The following table gives a comparison. There is a constant difference of a fraction of a per cent, due to difference of fundamental constants.

From table on p. 820...	8.40	10.90	13.90	17.70	21.60	25.50	29.20	33.00	38.00	42.60	46.60
From formula.........	8.69	11.09	14.05	17.95	21.82	25.66	29.44	33.38	38.19	42.91	46.95

The formula used by the author for the thin-plate orifice is theoretically correct only for pressure ratios greater than 0.52. For less pressure ratios the flow is dependent on the initial pressure only. I understand that the author has made his computations on this basis. However, his actual results for small pressure ratios show that the flow does vary with the final pressure. This means that the coefficient of a thin-plate orifice varies with the pressure. The exact variation is given for the first time by the author, and he gives a very interesting empirical law taking account of both the theoretical and the coefficient variation and giving the net flow.

The conclusion which I think should be drawn from the data presented is that the venturi meter is much preferable to the thin-plate orifice for the following reasons:

The venturi coefficient is well established by much previous data and by the author himself as nearly unity, while the coefficient of the thin-plate orifice is variable and the net flow is a complication of this factor and the theoretical flow. The author's empirical formula is very ingenious, but it would have to be established for many other cases before it would be proper to use in general.

The corner of the thin-plate orifice has an important influence and even the author used two different forms — square and with $\frac{1}{16}$-in. radius. I believe a venturi with a well-rounded approach and some length of parallel portion can be more easily made and duplicated than a true thin-plate orifice.

The author also mentions, in the steam case, an uncertainty regarding point of measurement of pressure.

He complicates the venturi case by use of the full theoretical formula, whereas much simpler and equally accurate formulæ are available. He uses a rather large venturi throat, giving small differentials which are comparatively hard to measure. One of the great advantages of the venturi is the fact that large differentials can be used by proper selection of the throat diameter.

Hence I feel that the paper gives valuable evidence showing that the venturi meter should not be displaced by the thin-plate orifice for the case of large pressures.

G. B. UPTON (written). Inspection of the paper as to conditions of measurement shows (1) that pressures were taken in the dead eddies close to the orifice plate, above and below; (2) that orifices were small compared to pipe diameter, the largest ratio being with a ½-in.-diameter orifice in a 2-in. pipe; (3) and that pressure drops were large, generally all or nearly all of the excess of pressure P_1 above atmosphere being used to cause flow. The author did not touch, nor intend to touch, on the experimental case of a thin-plate orifice, large compared to pipe and pressure drop small compared to initial pressure — that is, the conditions which would make the thin-plate orifice a competitor of the venturi tube as a measuring device.

The usual formula for flow of gases or vapors through a venturi is

$$W = CF_2\left(\delta_0 \frac{T_0 2gy}{P_0(y-1)}\right)^{\frac{1}{2}}\left(\frac{P_1}{T_1^{\frac{1}{2}}}\right)\left\{\frac{(1-x)^{2/y}-(1-x)^{(y+1)/y}}{1-a(1-x)^{2/y}}\right\}^{\frac{1}{2}} \quad [1]$$

In this equation the units are in the foot-pound-second system. W is the flow in pounds per second; C the "coefficient" of the venturi; F_2 the area of the throat of the venturi; δ the density in pounds per cubic foot; y the ratio of specific heats at constant pressure and constant volume for the gas or vapor; x the ratio of the "venturi head" (or drop in pressure from entrance to throat of the venturi) to the driving pressure at entrance, or $x = \Delta P/P_1 = (P_1 - P_2)/P_1$; and a the square of the area ratio F_2/F_1. Subscripts 0 refer to standard conditions for the gas; subscripts 1 and 2 to entrance and throat of the venturi, respectively.

In an article by the discusser in the *Sibley Journal of Engineering* for December, 1914, pp. 90–95, it was shown that the venturi formula [1] could be replaced by

$$W = CF_2\left\{\frac{2g\,\delta_0 T_0}{P_0(1-a)}\right\}^{\frac{1}{2}}\left\{\frac{P_1\cdot\Delta P}{T_1}\right\}^{\frac{1}{2}}\left\{1-\frac{3+a}{4y(1-a)}\cdot\frac{\Delta P}{P_1}\right\} \quad [2]$$

or

$$W = CF_2\left\{\frac{2g\,\delta_0 T_0}{P_0(1-a)}\right\}^{\frac{1}{2}}\left\{\frac{P_1\cdot\Delta P}{T_1}\right\}^{\frac{1}{2}}\left\{1-\frac{(3+a)}{2y(1-a)}\cdot\frac{\Delta P}{P_1}\right\}^{\frac{1}{2}} \quad [3]$$

The accuracy of this simplified formula was shown to be within one per cent, even for the largest values of $\Delta P/P_1$, if we remember the "critical" value of P_2 with regard to P_1, for gas expansion, and the variation of nozzle coefficients with flow when ΔP is large compared with P_1.

To compare equation [3] with that of the author, we must change to the same units. W of [3] changes to Q of his formula by multiplication by $60/\delta_0$. P and ΔP values of [3] change to corresponding values of the author's formula by multiplication by 144. F_2 of [3] must be multiplied by 144 to reduce to inches. C may be taken as unity. δ_0 for air = 0.0807; T_0 = 492 deg. fahr.; P_0 = 14.7 and $\sqrt{2g}$ = 8.02. Hence the reduced formula obtained from [3] is

$$Q = \frac{816\,F_2}{\sqrt{1-a}}\left\{\frac{P_1 \cdot \Delta P}{T_1}\right\}^{\frac{1}{2}}\left\{1 - \frac{(3+a)}{2\,y\,(1-a)} \cdot \frac{\Delta P}{P_1}\right\}^{\frac{1}{2}} \quad . \quad . \qquad [4]$$

The orifices used by the author ranged from $\frac{1}{8}$ in. to $\frac{1}{2}$ in. in diameter, in 2-in. pipe. Taking a $\frac{3}{16}$-in. orifice in 2-in. pipe as

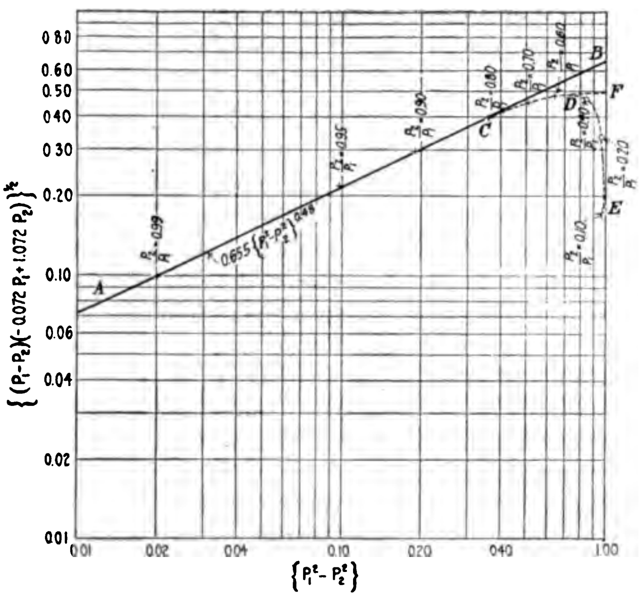


FIG. 25 COMPARISON OF REYNOLDS AND VENTURI FORMULAE

typical, the ratio of orifice area to pipe area is $(\frac{3}{32})^2$. There is a contraction of area of the flow beyond the thin-plate orifice on the low-pressure side, with a "contraction coefficient" (K) which most probably lies near 0.60, and must be between 0.60 and 1.00. The reason that we cannot take K as 0.60 is that when the pressures are measured, as the author did, close up to the orifice plate, the pressure drop is slightly less than if it were taken exactly at the section of minimum area of flow below the orifice. This smaller pressure drop we compensate for in our calculations by using a value of K larger than its true value, which is somewhere near 0.60. The effective value of K may be 0.70 or 0.80.

It is the contracted flow beyond the thin-plate orifice which really corresponds to the throat of a venturi. The area ratio from pipe size to throat size is $K\left(\frac{3}{32}\right)^2$; and assuming $K = 0.75$, the factor a in the venturi formula is $0.75^2\left(\frac{3}{32}\right)^4 = 0.000043$. This is so near to zero as to be negligible. The value of y for air is 1.40, under ordinary pressures and temperatures. On account of the contraction of flow beyond the orifice the value of F_2 in the venturi is KA, A being the area of the orifice. With these values inserted in formula [4] it becomes

$$\left.\begin{aligned} Q &= 816\,KA\left\{\frac{P_1\cdot\Delta P}{T_1}\right\}^{\frac{1}{2}}\left\{1-\frac{3.00}{2\times 1.40}\cdot\frac{\Delta P}{P_1}\right\}^{\frac{1}{2}} \quad \text{or} \\ Q &= \frac{816\,KA}{\sqrt{T_1}}\left\{P_1\,(P_1-P_2)\left(1-1.072\,\frac{(P_1-P_2)}{P_1}\right)\right\}^{\frac{1}{2}} \quad \text{or} \\ Q &= \frac{816\,KA}{\sqrt{T_1}}\left\{(P_1-P_2)\,(-\,0.072\,P_1+1.072\,P_2)\right\}^{\frac{1}{2}} \end{aligned}\right\} \cdot \quad [5]$$

The expression $\{(P_1 - P_2)\,(-\,0.072\,P_1 + 1.072\,P_2)\}^{\frac{1}{2}}$ is obviously not convertible by algebraic transformation into anything much like the author's expression $(P_1^2 - P_2^2)^{0.48}$, which may be written for comparison as $\{(P_1 - P_2)\,(P_1 + P_2)\}^{\frac{1}{2}}$. Dropping the attempt to make a direct conversion of one formula into the other, we may test to see whether there is some constant ratio between the two expressions $\{(P_1 - P_2)\,(-\,0.072\,P_1 + 1.072\,P_2)\}^{\frac{1}{2}}$ and $(P_1^2 - P_2^2)^{0.48}$. Fig. 25 and the following table show the comparison and its results.

COMPARISON OF REYNOLDS AND VENTURI FORMULÆ

Argument, P_2 (P_1 taken as unity)	Values of expressions		Ratio	Remarks
	$[(P_1-P_2)(-0.072\,P_1+1.072\,P_2)]^{\frac{1}{2}}$	$(P_1^2-P_2^2)^{0.48}$		
0.99	0.09945	0.1526	0.652	Above critical pressure
0.95	0.2175	0.3271	0.665	
0.90	0.2971	0.4506	0.659	
0.80	0.3964	0.6122	0.647	
0.70	0.4512	0.7238	0.623	
0.60	0.4780	0.8072	0.592	
0.40	0.4627 (0.490)	0.9187	0.504 (0.533)	Below critical pressure
0.20	0.3375 (0.495)	0.9806	0.344 (0.499)	
0.10	0.1780 (0.500)	0.9951	0.179 (0.503)	

The figures in the table are plotted in Fig. 25. The line AB has the equation 0.655 $(P_1^2 - P_2^2)^{0.48}$. Up to the point C it fits nicely the values calculated from the modified venturi equation. The venturi values then go down CDE; but the venturi equation is here wrong because the values of P_2 are below the critical. The actual values would follow out the general direction CDF rather than CDE. If we remember that the contraction coefficient of the orifice is a function of P_1 and $(P_1 - P_2)$, it will be evident that the expression 0.655 $(P_1^2 - P_2^2)^{0.48}$ may be a very good fit to experimental values of flow throughout the entire range of pressure and pressure-drop values.

Putting the expression 0.655 $(P_1^2 - P_2^2)^{0.48}$ in place of $\{(P_1 - P_2)(-0.072 P_1 + 1.072 P_2)\}^{\frac{1}{2}}$ in equation [5], there results

$$Q = \frac{0.655 \times 816\,KA}{\sqrt{T_1}} \left\{ P_1^2 - P_2^2 \right\}^{0.48} = \frac{534\,KA}{\sqrt{T_1}} \left\{ P_1^2 - P_2^2 \right\}^{0.48} \quad \cdot \ [6]$$

The author's expression makes the coefficient of A to be 405. For this to be so, it is necessary that K have the value $405/534 = 0.758$, which, as has been pointed out above, is both possible and reasonable. The conclusion is that the author's empirical formula agrees as well as can be expected with the mathematical calculations of the flow of air under the circumstances of his measurements.

The Author. In regard to the relative merits of the venturi tube and the thin-plate orifice for measuring compressed air and steam, I wish to say that the venturi tube considered from the theoretical point of view is a more desirable instrument. However, when the cost and other practical considerations are taken into account, the thin-plate orifice has a great many advantages over the venturi tube, as is pointed out by Mr. Pigott.

In reply to Mr. Thurston's question as to the use of venturi tubes in parallel, I have no doubt that the apparent difference of pressure noticed by him when some of the tubes are shut off is due to the pulsating pressure, as explained by Mr. Connet.

The method of measuring the air described by Mr. Connet is probably as accurate as the method used in these tests. However, I think that the gasometer method is freer from uncertainties than any other method, as it is a direct measure of the actual volume of air.

Referring to Mr. Christie's interesting remarks about the close agreement of the coefficient of the Taylor pitot tube when used in a 12-in. pipe with the coefficient which I found for a 2-in. pipe, I find that, by referring to my original curves — which are easier to interpolate — the coefficients are about 0.78 for both the 2-in. and the 1½-in. pipes.

I think that everyone will agree with me that the pinhole type of pitot tube is much to be preferred to the Taylor tube, as the coefficient of the pinhole type can be considered unity for all practical purposes if a traverse is made in the pipe or duct. The data for the Taylor tube were obtained in this case because it was desirable to calibrate this form of tube in connection with some other work.

I do not agree with Dr. Moss when he says that the formulæ which I have used should never be used for computations. One object of these experiments was to compare the actual performance with the theoretical performance, and thus I do not think that short-cut methods should be used on the theoretical side in order to save work any more than they should be used in determining the actual performance, namely, in the actual measurement of the air. A great deal of work and expense could have been saved if some simpler method had been used in measuring the air. However, the uncertainties would have been increased. On the other hand, a simpler formula might have been used in computing the theoretical flow, and giving the same degree of accuracy as pointed out by Dr. Moss, but at the same time involving uncertainties. Therefore I think in this case, with the object of the computations in view, the unquestionable correct theoretical formula should be used in comparing the theoretical flow with the actual flow.

I have made the above remarks about the approximate formula granting that it gives practically the same results as the theoretical formula. However, in investigating the formula as given by Dr. Moss, which becomes

$$Q = 286.1 \sqrt{P_1 (P_1 - P_2)/T_1}$$

for the 2 × 0.666-in. venturi tube, I find that for the lower pressure ratios it gives results which are quite far from the correct results. For example, the percentage of error varies from 0.64 per cent when the pressure ratio is 0.99892, to 34.62 per cent when the pressure ratio is 0.5821. A simplified formula which may be used with accuracy is given by Professor Upton in his discussion.

As pointed out by Dr. Moss, the equation which I have used for the orifice is theoretically correct only when the ratio of pressures is greater than 0.527. However, this equation may be used when the ratio of pressures is less than 0.527, if the value 0.527 P_1 is substituted for P_2 in the formula. This was done whenever the ratio of pressures was less than 0.527.

Dr. Moss says that the size of the venturi throats which I have used resulted in very small differentials which are very hard to measure. Expressed in pounds per square inch, these differentials appear to be very small. However, it should be kept in mind that these pressure differences were measured in inches of water, which resulted in readings of considerable magnitude in most of the tests. In fact, in some of the tests the differentials were so large that it was necessary to use mercury in the manometer in place of water. By referring to the complete results of the tests it is found that the ratio of pressures varied from 0.99982 to 0.5821 for the 2×0.666-in. tube, and from 0.9952 to 0.616 for the $1 \times \frac{3}{8}$-in. tube.

No. 1556

SPONTANEOUS IGNITION STUDIED BY MEANS OF PHOTOGRAPHIC PLATES

By Frederick J. Hoxie, Boston, Mass.
Member of the Society

Spontaneous ignition is a much more common cause of fires than is generally supposed. Many familiar substances can ignite spontaneously at slightly elevated temperatures or if acted upon by strong sunlight. Fires from this source are in the class most dangerous to life and property, because they start at the least expected time and place, and when no one is near to extinguish them or to give the alarm. The study of this class of fires, therefore, offers an attractive field of usefulness for the fire-protection engineer; but the pursuit of such studies is beset by difficulties arising from a lack of exact knowledge of the chemistry of oxidation and lack of convenient methods of studying the early stages of combustion.

2 The ignition of a handful of excelsior with a match is familiar to all: the match raises the temperature of a small amount of the wood above its ignition point, which is about 500 deg. fahr.; the combustion of this wood then raises the temperature of the surrounding particles above their ignition point, and the flame spreads rapidly through the excelsior. If the excelsior is wet, or is pressed close against a cold surface, the velocity of combustion may not be sufficient to keep the surrounding temperature up to the ignition point, in which case it will smolder, and its flame or glowing coal will "go out." The effect of the flame from the match is to render the oxygen of the air, which at lower temperatures is inert, sufficiently active to combine with the wood.

OXYGEN CAN BE MADE ACTIVE BY SUNLIGHT AND ELECTRIC CHARGE AS WELL AS BY HEAT

3 This quickening of oxidation may be accomplished, although much less thoroughly, by sunlight or other light of short wave lengths,

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by an electric charge, or by radioactive materials. Oxygen in this condition is said to be activated, ionized, or, when in this state as a result of just emerging from a chemical combination, nascent.

4 On the other hand, the wood instead of the oxygen may become activated. If the wood is partially rotted or charred, or is in intimate association with oxygen-carrying materials, it may be caused to ignite after a considerable time at a surrounding temperature below 300 deg. fahr., or very much below the temperature at which it can be ignited under normal conditions.

5 The sight of dry cotton or of the fine shavings of soft wood known in the trade as "excelsior" in contact with the oxygen of the atmosphere in sunlight with no apparent oxidation or combustion, is too familiar to attract attention. If such material should burst into flame spontaneously at the ordinary temperature of the air, this action would be regarded as little short of a miracle. Nevertheless, this cotton and excelsior are all the time very slowly burning or being oxidized at the ordinary atmospheric temperature, the velocity of this oxidation or slow combustion depending on the temperature, moisture, light and mineral or organic substances associated with the cotton or wood.

6 Cotton cloth which has been exposed for several months to sunlight and atmospheric moisture to the extent of so-called "dampness," exhibits on close inspection somewhat of the same charred appearance that it has after a few seconds' contact with a hot iron. The addition of certain mineral substances, known to chemists as catalyzers, such as iron and manganese, which can act as carriers of oxygen, accelerates this action. This slow combustion liberates the same amount of heat per pound burned that would be liberated by the more familiar combustion with flame, but the heat is given off over such long periods of time that the resulting rise in temperature is not appreciable.

VELOCITY OF COMBUSTION INCREASES RAPIDLY WITH TEMPERATURE

7 If, however, the slowly burning substance is completely inclosed by material of high heat-insulating power, the heat generated by the combustion will be made to escape more slowly and the temperature will be increased proportionally. The velocity of combustion increases with the temperature and more rapidly than the first power of the temperature, as shown by the curves in Fig. 1 giving the rate of oxygen absorption, which is the first stage in combustion. On

the other hand, the escape of heat will be more rapid with higher temperature and the same heat insulation; so that in most cases in common experience the velocity of combustion, the rise in temperature and the radiation of heat soon come to equilibrium with each other.

8 In many cases where the burning substance is enclosed or shielded from atmospheric circulation, the exhaustion of the oxygen of the air in contact with it limits the oxidation. If, however, the heat insulation is sufficiently good and the necessary air supply is available, and if the temperature is sufficient to cause the combustion to proceed at a rate which increases more rapidly than the increase in the rate at which the heat is being radiated, ignition will result.

CURVE OF VELOCITY OF OXIDATION SIGNIFICANT

9 In this connection the form of the velocity-of-combustion curve gives the most significant characteristics of substances subject to spontaneous ignition. In studying this hazard it is important to know both the velocity of combustion of a given substance at the ordinary temperature of the air and *also the temperature at which this velocity commences to increase rapidly, as indicated by the hump on the velocity curve;* such, for example, as is found at 100 deg. fahr. for linseed oil and at 200 deg. fahr. for cottonseed oil. See Fig. 1.

10 To obtain practically this curve of the velocity of combustion for different substances at the ordinary temperature of the air, is a difficult matter. To determine it by the rate of absorption of oxygen as shown in Fig. 1 is a troublesome and uncertain process, owing to the fact that there are doubtless many intermediate reactions between the combination of oxygen and wood or cotton or other hydrocarbons and the end products, which are carbonic acid and water. Therefore, any means of determining this curve or of *comparing the velocity of oxidation of substances of which the characteristics are unknown with that of substances of which the characteristics are known,* will be a valuable help in studying the mysterious mechanism of spontaneous ignition.

SUBSTANCES UNDERGOING AUTOMATIC OXIDATION GIVE OFF HYDROGEN PEROXIDE

11 Schoenbein showed that many substances undergoing automatic oxidation give off hydrogen peroxide.[1] Moritz Traube later

[1] Gesammelte Abhandlungen von Moritz Traube (Berlin, 1899), p. 400.

pointed out that many of these substances *would not oxidize automatically except in the presence of water*,[1] and propounded the theory, which bears his name, that the hydrogen peroxide is formed by the combination of the oxidizable substance with the oxygen of water, the hydrogen of the water combining with free oxygen to form hydrogen peroxide. This cycle would make moisture a necessary part of an automatic oxidation, an equal number of molecules of water and oxygen being required.

12 The hydrogen peroxide being unstable at high temperatures and high concentrations, it would be expected to decompose soon into water and oxygen, thereby lessening the quantity of water necessary for the progress of the reaction. However, when the reaction is proceeding very slowly, at the ordinary temperature of the air, hydrogen peroxide would probably be formed at the same rate as the oxidation.

13 Many examples can be cited of extremely slow oxidation and of oxidation once started which has practically stopped from the absence of water; noticeable among which are the linen cloths wrapped about the mummies of Egypt, which have oxidized but little in four thousand years, and also the wooden mummy cases.

HYDROGEN PEROXIDE CAN MAKE A DEVELOPABLE IMAGE ON A PHOTOGRAPHIC PLATE

14 The hydrogen peroxide given off by substances undergoing automatic oxidation can make a developable image on a photographic plate in the dark, and the intensity of this image appears to be about proportional to the velocity of oxidation, as would be expected from Moritz Traube's theory, if each molecule of oxygen absorbed sets free a molecule of hydrogen peroxide and if each molecule of hydrogen peroxide has an equal action on the sensitive plate.

15 The intensity of the impressions made on photographic plates by exposure to close contact with many different substances, among which are woods of various kinds, rosin, coal and zinc, has been investigated by William J. Russell.[2] Oils subject to spontaneous ignition are found particularly active.

[1] Gesammelte Abhandlungen von Moritz Traube (Berlin, 1899), p. 403.

[2] Proc. Royal Soc., vol. 78 (1908), p. 385; vol. 80 (1908), p. 376. Philosophical Transactions, vol. 197 (1905), p. 281. British Journal of Photography, Nov. 15, 1908, p. 866.

INTENSITY OF IMAGE PROPORTIONAL TO RATE OF OXIDATION

16 In order to determine whether the intensity of the image on the photographic plate is proportional to the velocity of oxidation, I have taken samples of raw linseed, refined cottonseed, and lard oils and have measured the rate of absorption of oxygen by means of the decrease in the volume of air in contact with the oil for short periods at temperatures from 60 to 250 deg. fahr. The results of my experi-

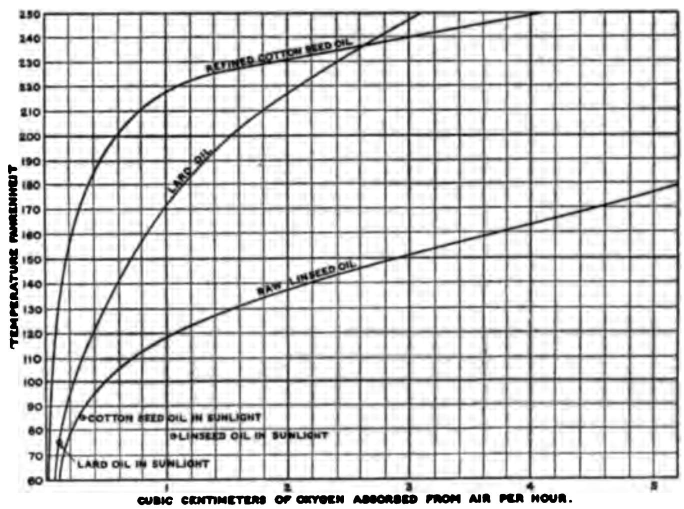


FIG. 1 INCREASE IN OXYGEN ABSORPTION OF OILS WITH INCREASE IN TEMPERATURE

ments are shown in Fig. 1. In Fig. 2 are shown the impressions made on photographic plates by placing the oils near them in the dark.

17 The photographic plate undoubtedly indicates only the first stage in the automatic oxidation reaction, namely, the absorption of oxygen by the unsaturated oils and the formation of hydrogen peroxide.[1] It is apparent from the lack of parallelism between the curves in Fig. 1 for lard oil, linseed and cottonseed oil that the rate of absorption at low temperature is not proportioned to the so-called iodine values[2] for these oils. The lard oil, with an iodine value less

[1] Kritische Studien über die Vorgång der Autoxydation, C. Engler. J. Weissberg, 1904.

[2] The iodine number of an oil is the percentage of iodine which it can absorb. It represents the amount of oxygen which can be absorbed, while the photographic image represents the rate at which the oxygen is absorbed, which is the important factor in spontaneous ignition.

than the cottonseed, absorbed considerably more rapidly up to about 230 deg. fahr. Also the impression made by the lard oil on the plate is stronger than that by the cottonseed oil, which has more than double the iodine value. The photographic plate can only be used at low temperatures, owing to its sensitiveness to heat.

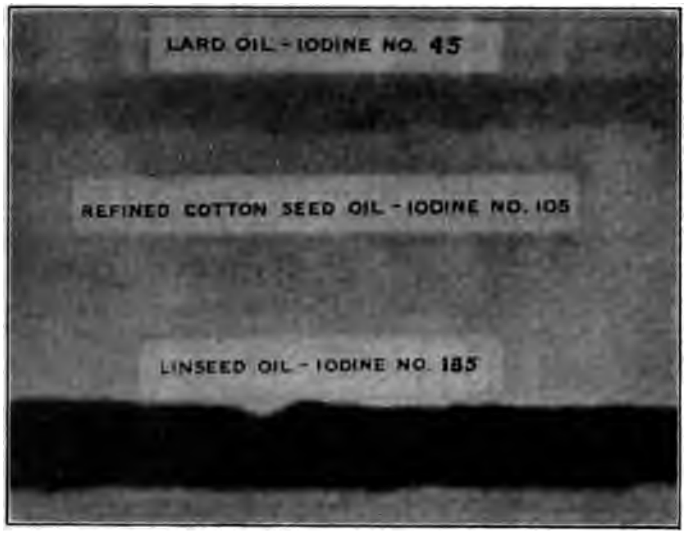


FIG. 2 IMAGES MADE ON PHOTOGRAPHIC PLATES BY OILS OF FIG. 1

HUMP IN OXIDATION CURVE AN IMPORTANT FEATURE

18 A feature worthy of careful notice with reference to the spontaneous-ignition hazard is the decided hump in the absorption curves of linseed and cottonseed oil at 100 deg. and 200 deg. fahr., respectively; also the fact that lard oil at ordinary temperatures absorbs oxygen more rapidly and acts on the photographic plate more strongly than cottonseed oil. From the absorption curve of cottonseed oil it would be expected that the spontaneous-ignition danger from this oil would not be great at temperatures below 150 deg. fahr., and that linseed oil could be used for almost any purpose with comparative safety against spontaneous ignition at temperatures below 80 deg. fahr., if kept out of the sunlight; also that lard oil would be more hazardous than cottonseed oil at room temperatures.

SUNLIGHT INCREASES THE RATE OF OXIDATION

19 A potent factor in increasing the rate of oxidation at room temperature is light, not necessarily ultra-violet light, but ordinary sunlight which has passed through several thicknesses of lead glass so that most of the light of shorter wave length has been removed. This at 80 to 85 deg. fahr. increased the rate of absorption of oxygen by cottonseed and linseed oils by about ten times, while with lard oil it apparently had less effect.

20 Oils painted on glass and maintained at 120 deg. fahr. for several days while being exposed to a photographic plate showed a stronger activity than other similar samples maintained at 65 deg. fahr., in about the proportion indicated by the oxygen-absorption curves.

COLLOIDS AND CATALYZERS FUNDAMENTAL IN PLANT AND ANIMAL PHYSIOLOGY

21 Automatic oxidation and reduction at low temperatures are fundamental to the physiological reactions which go on in plants and animals, the mechanism by which these reactions are carried out being a combination of colloids and catalyzers. Gelatinous substances, starch and cellulose act as the colloids, and iron and manganese compounds are chief among the catalyzers or carriers of oxygen. Dony-Henreault[1] has shown that many of the oxidizing reactions given by organic compounds known as oxidases can be duplicated by a combination of gum arabic, which is a colloid, and minute quantities of manganese.

22 That sunlight is not only an important factor in increasing the rate of oxidation and photographic activity of the oils, but also the photographic activity of wood, is shown by the impression made by a piece of birch on a photographic plate (see Fig. 3). This piece of red birch with heart and sap wood was exposed to the action of sunlight for an hour, with a cross-shaped piece of tinfoil screening part of it from the light. The tinfoil was then removed and a rapid photographic plate placed against the wood and both left for a week in total darkness. At the end of this time the plate was developed, giving an image as shown in Fig. 3.

23 Subsequently I extracted what was apparently one or more of the materials causing this photographic activity from wood shav-

[1] Bull. Acad. Royal Belg., Class de Science, 1907, p. 536; 1908, pp. 105–163; 1909, pp. 342–409.

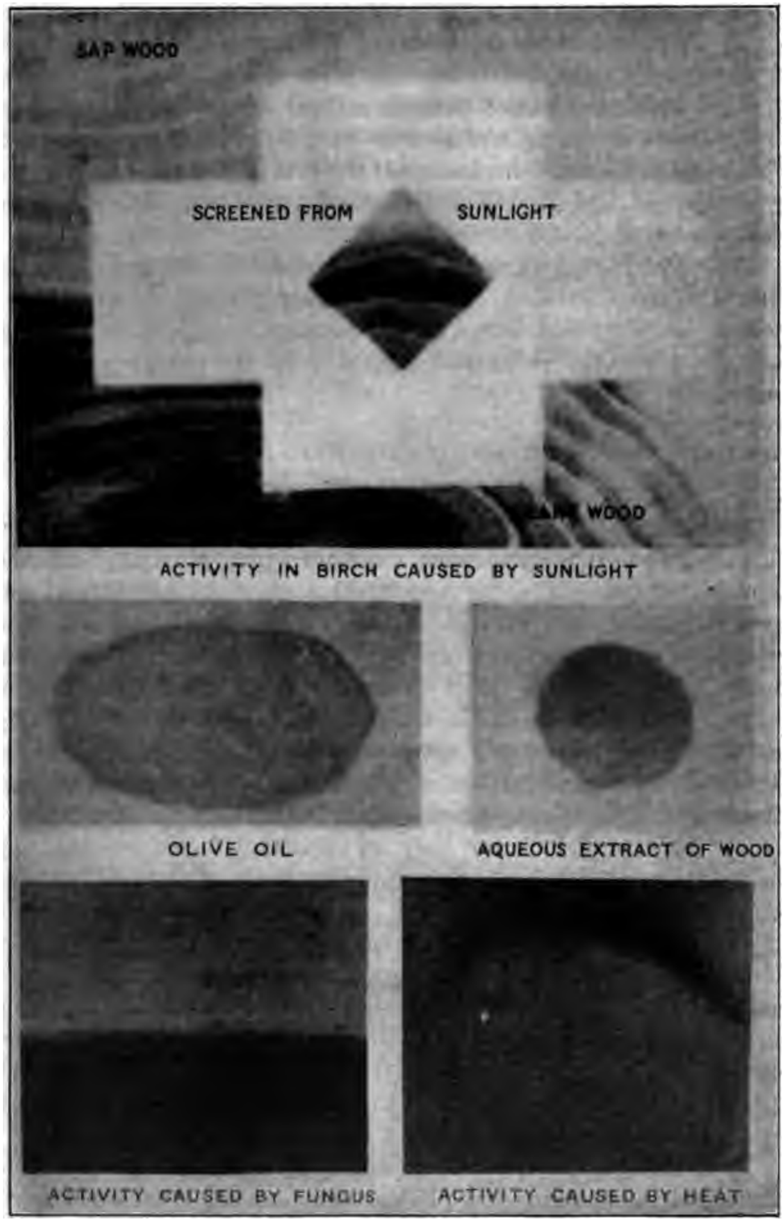


Fig. 3 Images Resulting from Photographic Activity Due to Various Causes

ings, by soaking them in water. It was then precipitated by adding strong alcohol to the water, filtered out, dissolved in a small amount of water and painted on a piece of plate glass, which is the method employed by Bertrand[1] for extracting oxidases from wood and plants. The impression which this made on a photographic plate had an intensity about equal to that made by olive oil, as shown in Fig. 3.

MODERATE HEATING INCREASES PHOTOGRAPHIC ACTIVITY, WHILE STRONG HEATING DESTROYS IT

24 The lower right-hand part of Fig. 3 was obtained from a piece of birch which had been moderately heated by being placed in contact with a hot electric stove for a few seconds. The wood, after cooling, was placed in contact with a photographic plate in the dark and developed at the end of a week, the image indicating that it had been rendered most active by a temperature in the neighborhood of 150 deg fahr. This may be a first reaction in the combustion of wood, which forms hydrogen peroxide more rapidly than it is decomposed, while at a slightly higher temperature the rate of decomposition is greater than that of formation.

25 In the hard pines there are at least three different substances which can cause this activity, namely, turpentine, resin, and the water-soluble substance mentioned above.

IT IS POSSIBLE TO PRINT A PHOTOGRAPH ON NEWS PAPER

26 It is possible to make a fairly clear photograph upon ordinary unprinted news paper containing considerable ground wood, by rinsing it in hot water and drying it in a dark room. This destroys the peroxide already formed by exposure to light, which would otherwise cause it to act like photographic paper that has been fogged by exposure to light. The paper is then exposed to sunlight behind a very strong negative for half an hour, and developed with ferrous sulphate and tannic acid — separately or together. The resulting picture is in tannate-of-iron ink. A picture can be similarly printed on the heart wood of several varieties of trees and developed with ferrous sulphate.

27 The chemical cause of this is doubtless the same as that of the photographic activity. In the oxidation brought about by the sun-

[1] Sur le Latex de l'arbre laque. Comptes Rendus, Académie des Sciences, vol. 118 (1894), p. 1215; vol. 120 (1895), p. 266; vol. 121 (1895), p. 166; vol. 122 (1896), p. 1132; vol. 124 (1897), p. 1032; vol. 124 (1897), p. 1355.

light, hydrogen peroxide or other form of active oxygen is formed, and this oxidizes the ferrous salt to a ferric salt which combines with the tannin already in the wood or with that added in the developer in the case of the news paper, forming the blue-black tannate of iron.

SPONTANEOUS IGNITION OF CHARCOAL

28 Spontaneous ignition of wood is doubtless intimately associated with spontaneous ignition of charcoal, and this phenomenon has a much wider field of interest than would appear at first sight. In dwellings and factories numerous fires originate in the woodwork near steam pipes or other heating appliances which are maintained at temperatures far below the ignition point of wood, and are due to spontaneous ignition of charcoal formed by slow combustion over long periods of time.

ALL CHARCOAL IS NOT ALIKE

29 All charcoal is not alike, the difference appearing to be somewhat dependent on the completeness of the charring process, and apparently, to some extent, on the variation in the chemical components of the charcoal. The images shown in Fig. 4 were made upon a Standard "Orthonon" photographic plate by several samples of charcoal from a lot, which, when freshly burned had caused six fires in rapid succession from spontaneous ignition.

30 Freshly burned charcoal is known to be highly susceptible to spontaneous ignition, and the records of the Bureau of Explosives of the American Railway Association indicate that charcoal burned in the spring is more subject to this action than that burned at other seasons of the year.

31 This charcoal ignited when kept for two hours in an oven at 250 to 275 deg. fahr. That which made the strongest impression on the plate ignited in 5 min. or less when put on an electric stove and heated to 500 deg. fahr., while that which made little or no impression on the plate did not ignite at this temperature in 15 min. Charcoal which has been strongly heated and quickly cooled again just before being put in contact with the plate, shows greater activity than that which has not been submitted to this action. Moist specimens are less active than dry ones, owing to the desensitizing action of the moisture on the sensitive coating. The specimens which are most active are generally of a brown color, sometimes as a result of incom-

plete charring and sometimes, apparently, from some difference in the chemical composition of the charcoal.

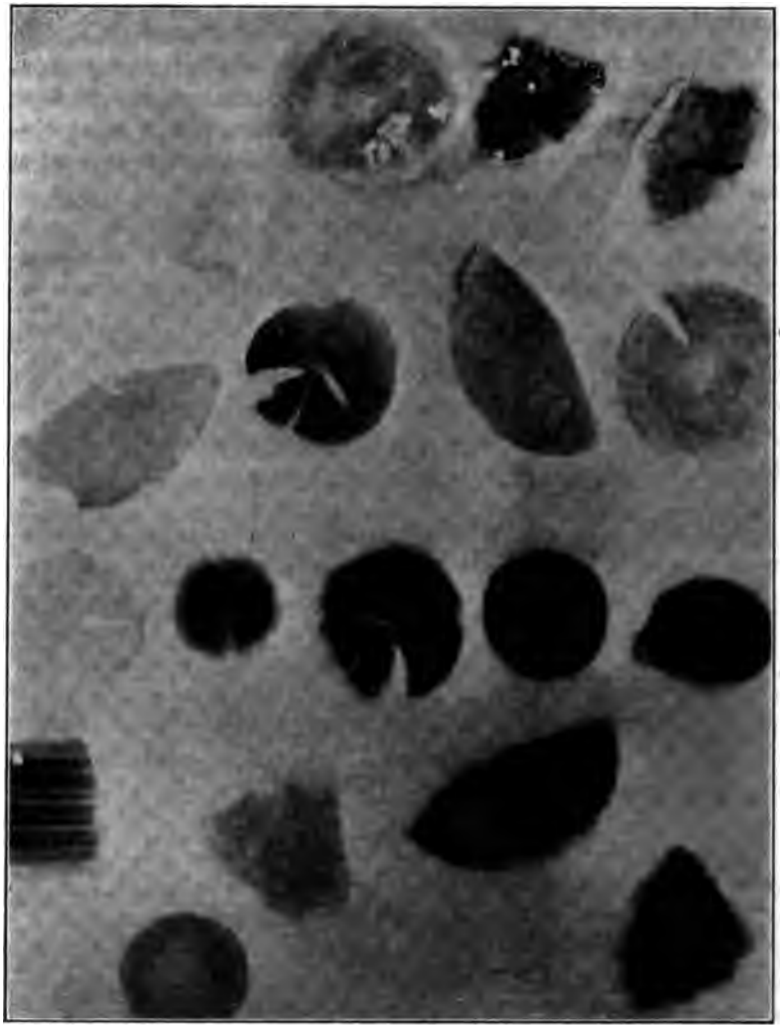


FIG. 4 IMAGES MADE BY DIFFERENT SPECIMENS OF CHARCOAL ON A PHOTOGRAPHIC PLATE

32 The foregoing statement of experiments and observations on photographic plates as an index of tendency towards spontaneous ignition is but a beginning, but it indicates a line of research that can

be followed profitably in studying the early stages of spontaneous oxidation.

33 Many substances which are subject to spontaneous oxidation, such as jute and hemp, may be seriously damaged commercially long before ignition takes place, and in some of them ignition rarely, if ever, occurs. Therefore, the interest arising from commercial considerations should make careful investigation worth while in cases where interest from the standpoint of fire protection would not warrant the expense.

DISCUSSION

IRA H. WOOLSON asked the author if he had discovered any difference in the activity of charcoal due to the variety of wood from which it was made; whether wood that contained resin, for instance, had any different activity upon the plate, as compared with the wood that did not contain resin.

C. P. BEISTLE[1] stated that charcoal made from beech, birch or maple was more liable to spontaneous ignition than that made from other woods. He thought the real cause of the risk in charcoal was due to the process rather than to the wood used. Most charcoal nowadays was made as a by-product of making wood alcohol and acetate of lime, and almost invariably retained a good deal of volatile matter. Formerly, charcoal was burned anywhere from seven to fourteen days, and was very hard, very dense, and almost free from spontaneous heating.

LEONARD WALDO stated that he had had a case come up of the ignition of flash powders, and was interested to find some simple way by which the rate of oxidation of magnesium, for instance, could be determined. He desired to learn whether oxidation of powdered metals, either with or without the presence of an oxidizing substance, would be shown by Orthonon plates, by simply putting the metal in contact with the plates and letting it stay there awhile exposed to the air; also whether there were any publications that gave extended lists of temperatures of ignition. If the gradual oxidation of metal could be shown, either in steels or lighter metals, by simply putting them on a sensitized plate, and developing the plate later, this would

[1] Bureau of Explosives, Am. Ry. Assn., 75 Church St., New York.

be a most important contribution to the technique of recording slow oxidation.

Christopher H. Bierbaum said that one question raised in the paper was whether the oxidation was actually in direct proportion to the actinic effect upon the plate. The chemistry of the photographic plate was this, that we started with bromide of silver. The bromine, being a strong oxidizer, produced an effect upon this salt, in the presence of the actinic ray, so that it would readily change from a bromide to an oxide during the developing. Was it not possible that it was all due to the ionic potential of the substance brought in contact with the plate? In other words, the question was whether it was actually due to actinic rays or vibrations in the hypothetic ether, or to the presence of an ionic emanation that produced this chemical change.

Charles T. Plunkett said that in the interest of the cotton manufacturers he would like to ask the author whether he had discovered that there was a material change in cotton fiber by the slow oxidation mentioned in the paper, which it was said is going on at all times, sometimes being on the outside of the bale, and having a moisture and temperature differing from that in the interior of the bale. And whether it had been found true that after a long time the staple was weakened on account of the oxidation which had been going on, possibly for years, before the bale had been opened for use.

Cotton eighteen or twenty years old was now coming on the market because of the present high price. The question was whether there had been a very serious deterioration in the interior portions of the bale, by reason of combustion that had been going on slowly, but over a long period of years; whether it was really deteriorated so that it would be unfit for use, or suitable for the uses for which such cotton would be apt to be used. He also desired information as to cotton fabric piled in storehouses in bales. Many times discoloration was found on the part toward the windows. Was it true that there would be a deterioration of the strength of that fabric, and would the color indicate that the fabric itself had been injured by its storage, where it had received the rays of light?

Milton F. Jones[1] (written). The subject of the early stages of oxidation of inflammable substances which, should the reaction

[1] National Fire Protection Association, Boston, Mass.

continue, may terminate in spontaneous ignition, is a matter of importance regarding which very little is known.

If the author's assumption that the effect upon the photographic plate is due entirely to liberated hydrogen peroxide, is correct, then we have perhaps the first tangible evidence of the beginning of the oxidation process, in a form which would allow comparisons to be made.

That the density of the photographic image does not harmonize with the iodine number of the oil, is not remarkable. The iodine number of an oil is usually employed as a means of identification. It is based upon the capacity of the oil in question to combine with one of the halogens, that is, iodine. It is doubtful, however, whether we have the right to assume that the oil will combine with an equivalent amount of oxygen. Chlorine unites directly with some elements with which it is difficult for oxygen to unite.

Because the drying oils have as a rule the higher iodine numbers and are prone to spontaneous ignition, it has been suggested that the iodine number might prove a guide in determining the tendency of an oil in this direction. Boiled linseed oil possesses greater drying properties than raw oil, and is usually considered more hazardous, yet it seems to have been conclusively demonstrated that it has the lower iodine number. The ordinary red oil, which is a crude oleic acid obtained from tallow — an animal oil, is probably as hazardous as regards spontaneous ignition as linseed oil. Cottonseed oil is not considered a drying oil.

The statement of the author that all charcoal is not alike will hardly be questioned; it is one of the most perplexing factors in the investigation of charcoal fires. He finds that the density of the photographic image corresponds with the activity of the sample of charcoal, which is interesting. The conditions as to charcoal are different from those of oil. In the latter case, having determined that an oil is hazardous, it can be safeguarded more or less. With the charcoal, however, although many samples may show an inactive condition, one active piece favorably located in a pile may under certain conditions determine the destruction of the mass.

As the author states, the results are only a beginning, and subsequent research will determine the value of the process, whether of scientific interest only, or whether applicable to industrial or fire-prevention uses.

The Author, in replying first to Mr. Woolson's question, stated that he had found certain varieties of wood were more active than others, but that some other specimens of the same variety were not active, so that he was not inclined to generalize very strongly on that point.

In reply to Dr. Waldo, he said that he had never heard of any experiments being made on magnesium; zinc, however, had been shown to be active. In regard to the ignition points of different substances, there was quite a lot of information available. To say, for example, that the ignition point of wood was 500 deg., was only generally correct. The time of exposure to the temperature, as well as the condition of the wood, would vary the results.

Replying to Mr. Bierbaum, he said that the action was purely a chemical one. Apparently most any rapid plate could be used, but a slow plate could not be used successfully. Some exposures had been made on Lumière X-ray plates, a very rapid plate, and some were made on the Standard Orthonon, also a rapid plate, but in no sense were they photographs. The first impression was that this was a radio-active phenomenon, that it was an emanation from the oxidizing substance, very much like the X-ray, but this had been found to be incorrect. One experiment tried was to place a thin microscope cover glass over the object which was being studied, with the result that the plate was unaffected under the glass except near the edges, where there was an apparent diffusion of the active gas. He thought the active agent present was hydrogen dioxide. It was well known that hydrogen dioxide, when put in contact with a photographic plate, would make a developable image. The active agent might possibly be some other form of active oxygen, but ozone, without the presence of water, did not give this action.

Replying to Mr. Plunkett's first question, he said there was little doubt but what the strength of the staple had suffered. A common sight in almost any cotton mill, where pieces of sheeting were used for window curtains, was that where the moisture touched a curtain it turned brown, and had every appearance of being as thoroughly burned as if it had been in contact with hot iron; this was doubtless due to the oxidation which had taken place because of the combination of moisture and light. He had often found cotton and linen, linen particularly, which had been subjected to a comparatively low temperature, after a long period of years, that had every appearance of being thoroughly charred.

As to the second question, he was inclined to believe that the staple would be found to be weaker, and, under certain conditions, considerably so.

Replying to inquiries as to how the substances were exposed, he said that in the case of articles of wood they were placed in direct contact with the plate, but the oils were kept a little distance away, to avoid getting oil spots on the plate. The nearer the object to the plate the stronger the action was, and therefore it was desirable to get it as close as possible.

No. 1557

HEAT TRANSMISSION THROUGH VARIOUS TYPES OF SASH

EXPERIMENTS TO DETERMINE THE RELATIVE HEAT TRANSMISSION THROUGH VARIOUS TYPES OF SASH AND THE CAUSES AND PREVENTION OF INTERNAL CONDENSATION IN DOUBLE-GLAZED SASH

By Arthur N. Sheldon, Providence, R. I.
Member of the Society

It is the purpose of this paper to describe some experiments which were made to determine the rate of heat transmission through various types of sash. In the industrial plant of today, where so much emphasis is placed upon maximum daylight, the exterior wall surface consists principally of windows, and the heat loss through them is a very important matter. Indeed, there may be a question whether in certain plants window area is not carried to excess, that is, to such a point that the increase of daylight is so slight as not to be warranted by the increased heat loss. In the absence of any authentic data on the relative heat loss through single- and double-glazed wood, steel, and hollow metal sash, these experiments were conducted to enable us to logically design and proportion the fenestration of industrial buildings. The experiments were conducted by W. S. Brown, under the writer's direction.

DESCRIPTION OF APPARATUS

2 *Test Box.* The sash tested were of such size as completely to fill an opening on the face of a test box 4 ft. 2¾ in. wide by 7 ft. 9½ in. high, the test box being 1 ft. 2 in. deep. The box was constructed of 2-in. tongued-and-grooved white pine, well seasoned, covered with 2-in. cork boards, the joints in the latter being made airtight with brine cement. To reduce to a minimum the leakage around the sash at its line of contact with the box, a continuous strip of wool felt 1 in. thick was placed around the edge, and the sash screwed firmly against it at short intervals. The box rested on two gypsum blocks, as shown in Fig. 1.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

3 *Source of Heat.* The test box was set up on end (Fig. 1), and an electric heater, supplied with direct current at 250 volts, was placed in the bottom. The heater consisted of six sets of nickel-steel resistance wires controlled by an external switchboard. The sets were arranged for control in three groups of two each. The wires in each set, varying in number from 14 to 18, were strung horizontally across

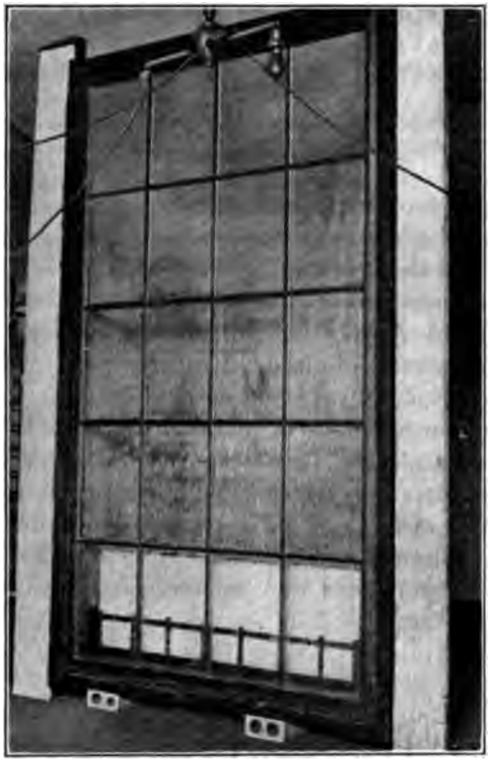


FIG. 1 SASH ON FRONT OF TEST BOX SHOWING GYPSUM-BLOCK SUPPORTS, BAFFLE BOARDS AND HEATER

the full length of the heater. Beginning at the bottom, the first two coils were designed to dissipate approximately 0.25 kw. each of electrical energy; the next two, 0.50 kw. each; and the uppermost two, 0.75 kw. each. The switching arrangement was so designed that any of the coils could be connected in multiple, and by means of triple-pole switches any two sets forming one group could be connected in series, thus giving a total capacity of 3 kw., obtainable in successive steps of $\frac{1}{8}$ kw. each. This was found sufficient for the purpose, and furnished ample capacity for heating the test box to the desired temperature in a short time. For arrangement of heater and switchboard see Figs. 2 and 3. To prevent direct radiation of heat through the lower panes of glass, $\frac{1}{4}$-in. white asbestos-board plates 15 in. high were fastened against the framework of the heater (Fig. 2).

4 *Air temperatures inside the box* were observed by means of twelve thermometers, arranged as shown in Fig. 4, and numbered from 1 to 12. The thermometers were placed so as to obtain a fair average of the internal air temperatures. In the vicinity of the heater, where the greatest variation of temperature naturally occurred, a double

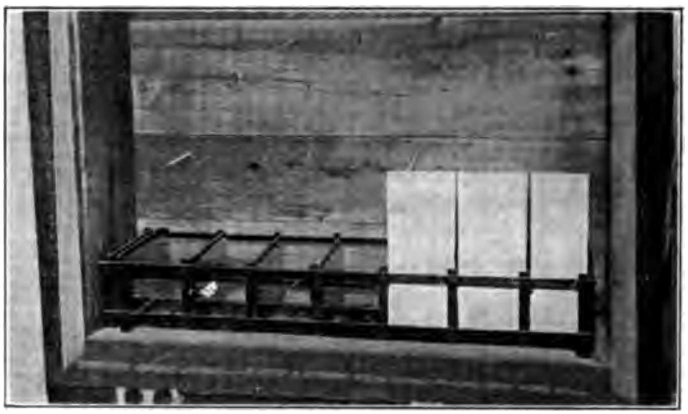


FIG. 2 INTERIOR OF TEST BOX, SHOWING HEATER WITH FOUR ASBESTOS PLATES REMOVED, AND ALSO THERMOMETERS

spacing was used, there being two thermometers above and two below the heater. In the tests of wood and steel sash, it should be noted that the thermometers above the heater were placed opposite the center of the various panes. The total average temperature was obtained by averaging the individual average temperatures behind each horizontal row of panes. From the many tests conducted, some of which were repeated, it is believed that a fair average temperature was obtained in this manner. The Fahrenheit long-scale chemical thermometers used protruded about 7 in. into the interior of the test box through holes bored through the back of the box and its covering. All thermometers were tested for accuracy before and after the experiments.

5 *Air temperatures outside the box* were observed by means of an

accurate wall thermometer hung midway between two fans (Par. 7) which directed currents of air across the face of the box. This thermometer registered the maximum and minimum temperatures and was used to determine the limits of temperature which had occurred in the room during each test.

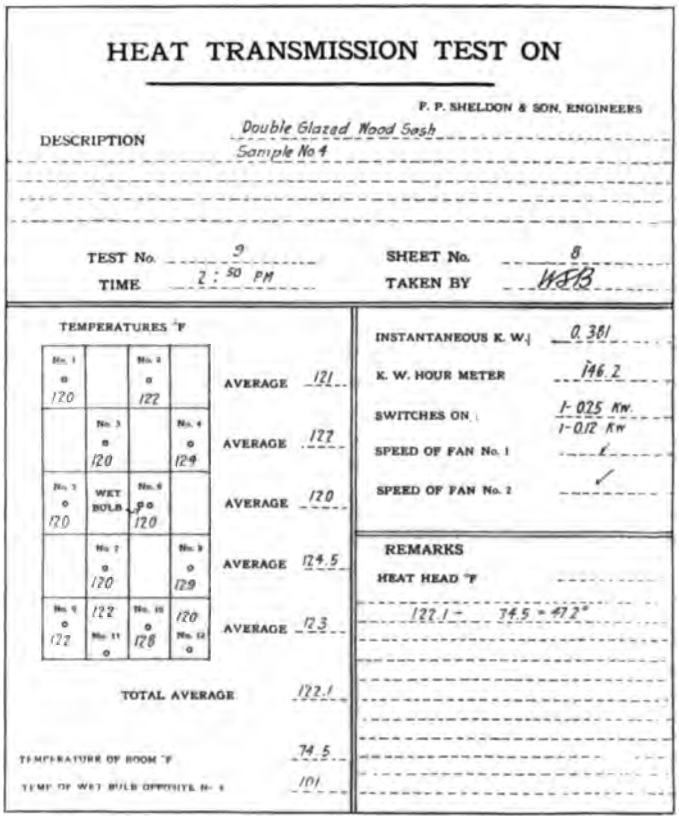
6 *Temperature control of the air* surrounding the test box was obtained by means of a thermostatically controlled steam radiator.

FIG. 3 GENERAL ARRANGEMENT OF APPARATUS, SHOWING FANS, SWITCHBOARDS, METERS, ETC.

Constant room temperature was desired because any appreciable variation of this temperature would materially affect the accuracy of the results on account of the temperature lag of the materials of which the box was constructed.

7 *Circulation of air* over the surface of sash to be tested was provided by means of two 12-in. fans, placed one above the other (Fig. 3), 4 ft. from the vertical center line of the sash. The fans directed air currents against the sash at an angle of 30 deg., and pro-

vided a positive circulation over it. It did not seem advisable to attempt to duplicate the actual conditions of wind on a building wall. Although Weather Bureau reports show that the average wind velocity in New England during the heating season is 11½ mi. per hr., the average conditions of direction would be difficult to reproduce, as usually no more than two sides of a building are exposed to wind



HEAT TRANSMISSION TEST ON

F. P. SHELDON & SON, ENGINEERS

DESCRIPTION Double Glazed Wood Sash
Sample No 4

TEST No. 9 SHEET No. 8
TIME 2:50 PM TAKEN BY WSB

TEMPERATURES °F

No. 1 o 120		No. 2 o 122		AVERAGE	121
	No. 3 o 120		No. 4 o 124	AVERAGE	122
No. 5 o 120	WET BULB	No. 6 o 120		AVERAGE	120
	No. 7 o 120		No. 8 o 129	AVERAGE	124.5
No. 9 o 122	122 No. 11 o	No. 10 o 128	120 No. 12 o	AVERAGE	123

TOTAL AVERAGE 122.1

TEMPERATURE OF ROOM °F 74.5

TEMP OF WET BULB OPPOSITE No. 6 101

INSTANTANEOUS K. W. 0.381
K. W. HOUR METER 146.2
SWITCHES ON 1-025 Kw.
1-0.12 Kw
SPEED OF FAN No. 1 ✓
SPEED OF FAN No. 2 ✓

REMARKS
HEAT HEAD °F
122.1 − 74.5 = 47.2°

FIG. 4 SAMPLE HOURLY RECORD SHEET

of this average velocity. The main purpose of the fans, therefore, was to provide a positive circulation of air over the sash, and thereby to secure constant conditions in this respect throughout the entire series of tests. Incidentally, a number of anemometer readings showed that the average velocity of the air current directed against the sash was approximately 7 mi. per hr. Considering that the average velocity of 11½ mi. per hr. is effective on two sides of a building only, it is believed that the conditions obtained in the test are not far from the average.

8 *Protection of the back and sides* of the box from air currents created by the fans was secured by vertical baffle boards, shown plainly in Fig. 1, placed on either side of the box. This was necessary because, as described later, the heat-transmission constants for the box alone were obtained under conditions of still air.

9 *Measurements of electrical energy* converted into heat were obtained simultaneously by means of two meters. The first meter was a standardized indicating wattmeter, guaranteed to be accurate within ½ of one per cent, and compensated for temperature changes. The scale was graduated to hundredths of a kilowatt and readings could be estimated to thousandths. The other meter was a standard service watt-hour meter, installed and adjusted by the service company furnishing the power.

10 *The humidity of the air* inside the test box was kept as constant as possible throughout the series of tests by means of water-saturated felt, and was measured by means of a wet- and dry-bulb hygrometer. The humidity remained constant during the period of each individual test.

METHOD OF CONDUCTING TESTS ON HEAT TRANSMISSION

11 The Government Climatological Reports show an average temperature of 36 deg. fahr. during the heating season in New England. Therefore, it was thought advisable to make a series of tests based on a heat head[1] of about 40 deg. fahr., corresponding to a room temperature of 76 deg. fahr. For the purpose of greater accuracy, however, and also to determine the relative heat transmission during colder weather, there was conducted another series of tests based on a heat head of about 70 deg. fahr.

12 In order to determine what portion of the total heat measured was conducted through the sash in a given test, it was necessary to subtract from the total the amount lost through the back, sides, top, and bottom surfaces of the box. This latter amount was ascertained by means of "blank-run" tests, the front of the box being covered in the same manner (Fig. 5) and with the same materials as the other surfaces. Furthermore, the error due to a possible change in conductivity of the box itself during the experiments was determined by tests made at heat heads of 40 and 70 deg. fahr., both before and after the series of sash tests. It was found that a slight increase in con-

[1] Temperature difference between outside and inside air.

ductivity occurred, due, evidently, to shrinkage and warping of the boards.

13 *Blank-Run Tests.* These were run very carefully over an extended period of time (at least 18 hr.) to make sure that all conditions such as temperature of the room, temperature in the box, and relative humidity had become constant. After taking several pre-

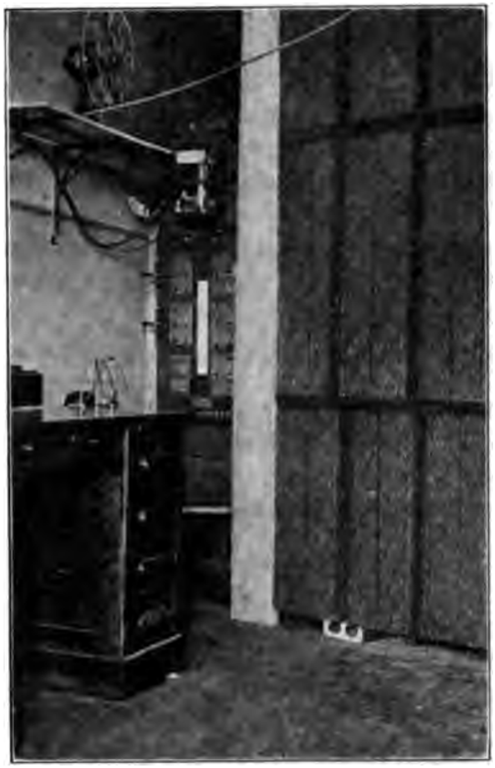


FIG. 5 FRONT OF TEST BOX COVERED FOR "BLANK RUN"

liminary readings to demonstrate this fact, regular readings, including all temperatures inside and outside of box, relative humidity, instantaneous kilowatts, kilowatt-hours, were recorded hourly for at least eight hours. In general, during the period of tests, the temperature of the room varied from 1 to not over 2 deg. fahr., the average temperature inside the box varying about 1 deg. fahr., and the heat head varying about 1 deg. fahr. Fig. 4 shows a sample hourly record sheet. During blank-run tests the fans were not in operation, as the object was to obtain the rate of transmission through the box and covering under the conditions of the main tests; that is, still air for the back, sides, top and bottom.

14 *Sash tests* were conducted in the same manner, except that the fans were put in operation and readings taken to be sure that they were up to speed. In each separate test the watt-hour meter was calibrated for the load by timing it for ten minutes with the instantaneous meter and counting the revolutions of the disk. This, compared with the average of about thirty readings of the instantaneous wattmeter taken over the same period, gave the required correction to apply to the service meter. When the correction factors found in this way were applied to the readings of the watt-hour meter, the results agreed very closely with the average of the hourly indicating wattmeter readings, the voltage being nearly constant. During all tests a careful record was kept of panes of glass on which condensation occurred.

RESULTS AND COMPUTATIONS OF TESTS ON HEAT TRANSMISSION

15 The heat transmission through the various types of sash has been worked out and expressed on the basis of B.t.u. transmitted per 24 hr. per deg. fahr. per sq. ft. of opening, and is designated in the formula following as H, whence

$$H = (L - l)/ad \qquad [1]$$

where

L = total heat loss through test box and sash in B.t.u. per 24 hr., as computed from wattmeter measurements

l = heat loss in B.t.u. per 24 hr. through top, bottom, back and sides of test box, as determined from blank run and corrected for the actual heat head during the sash test

a = area of opening filled by sash = 33 sq. ft.

d = temperature difference or average heat head recorded throughout test, deg. fahr.

L is found from the formula

$$L = (w \times 3412 \times 24)/t \qquad [2]$$

where

w = electrical energy dissipated in heater as measured in kilowatts at average temperature d

t = the number of hours duration of test

3412 = the equivalent of 1 kw. in B.t.u.

In the same manner l is found from the "blank-run" tests by the formula

$$l = R\,(w \times 3412 \times 24)/t \qquad [3]$$

where

R = the ratio of the area of top, bottom, back and sides of the box exposed during the sash tests, to the total area of the box, or that exposed during the blank run. As noted above, l should be taken at the same temperature as L, by interpolation from the known results if necessary.

The other terms have the same meaning as for equation [2].

16 Following is a sample computation of H for a double-glazed wood sash. From a summary of the test sheets, it was found that

$$d = 47.3 \text{ deg. fahr.} \qquad w = 1.50 \text{ kw.} \qquad t = 4 \text{ hr.}$$

During this period the temperature of the room varied 0.5 deg. fahr. and the average temperature of the box varied 0.4 deg. fahr., the average relative humidity being 49 per cent. From curves previously worked up from the blank-run tests, $l = 8600$ at a temperature difference of 47.3 deg. fahr., correction being made for a very slight change of conductivity in the test box itself during the series of tests. From equation [2]

$$L = (1.50 \times 3412 \times 24)/4 = 30{,}700 \text{ B.t.u.}$$

From equation [1]

$$H = (30{,}700 - 8600)/(33 \times 47.3) = 14.2 \text{ B.t.u.}$$

transmitted per 24 hr. per deg. fahr. per square foot of opening.

DESCRIPTIONS OF THE SASH TESTED

17 Tests were conducted on the following seven samples of sash, each of suitable dimensions to fit the test-box opening. All sash were without ventilators. In what follows, the word "pane" refers to a single sheet of glass, whereas the word "light" refers to a section of the sash or, in the case of double-glazed sash, two panes and the air space.

18 *Sample No. 1* was a standard make of single-glazed, solid, rolled-steel sash consisting of 20 lights, each approximately 12 in. by 18 in., arranged 4 wide and 5 high. The panes were ¼-in. rough wire glass and the putty seemed to be intact. The sash-bar section was 1⅜ in. deep, the glass being bedded on the front side, the putty at the back having a good ½-in. body. Glazing was done from the inside, as is customary with solid steel sash, on account of the section of the bar. The exposed glass area was 28 sq. ft., or 85 per cent of the total.

19 *Sample No. 2* was a double-glazed, solid, rolled-steel sash of the same make, size, and arrangement of lights as Sample No. 1. The outside panes were $\frac{1}{4}$-in. rough wire glass, the inside panes being single-thick plain glass, separated from the former by a galvanized-iron-channel separator, making a $\frac{3}{4}$-in. air space. The sash bar was $2\frac{3}{8}$ in. deep, the outer pane being well bedded on the outside and the inner pane put on with a good $\frac{1}{4}$-in. body of putty. All putty seemed intact. The glazing of both panes as before was done from the inside on account of the section of this type of sash bar.

20 *Sample No. 3* was a double-glazed, solid, rolled-steel sash of the same make, size, and arrangement of lights as Sample No. 1. The outside panes were $\frac{1}{8}$-in. factory ribbed glass (ribs inside); the inside panes were single-thick plain glass, separated from the former by a galvanized-iron-channel separator, making a $\frac{3}{16}$-in. air space. The sash bar was $1\frac{3}{8}$ in. deep, the outer pane being bedded on the outside and the inner pane put on with a good $\frac{1}{2}$-in. body of putty. All putty seemed intact. The glazing of both panes, as before, was done from the inside on account of the section of the sash bar.

21 *Sample No. 4* was an ordinary double-glazed wood sash, of same size and arrangement of lights as Sample No. 1. The outside panes were $\frac{1}{4}$-in. rough wire glass, the inside panes being double-thick plain glass, separated from the former by a $\frac{5}{8}$-in. air space. The sash was made of $1\frac{3}{4}$-in. stock, muntin bars being $\frac{5}{16}$ in. The outer panes had a $\frac{5}{16}$-in. body of putty, not bedded, and the inner panes $\frac{1}{2}$-in. body of putty, bedded. All putty seemed intact. Glazing was done, as is usual on this type of sash, from both sides.

22 *Sample No. 5* was a single-glazed wood sash of same size and arrangement of lights as Sample No. 1 and was obtained from No. 4 by taking out the inside panes of plain glass, thus leaving a sash-bar section practically the same as an ordinary single-glazed sash of this size. Glazing was done, as is usual on this type of sash, from the outside.

23 *Sample No. 6* was a double-glazed hollow metal sash. On account of the larger muntin bars required by this type of sash, it was decided to make the lights larger in order to obtain more light and more closely to approximate the probable actual design of a sash of this size. The sash was accordingly made three lights wide and four high, each light being approximately $14\frac{1}{2}$ in. by 21 in. The outside panes were $\frac{1}{4}$-in. rough wire glass, the inside panes being single-thick plain glass, separated from the former by $\frac{3}{4}$-in. air space. The exposed glass area was 23.5 sq. ft., or 71.5 per cent of the total. The

sash bar was $2\frac{1}{8}$ in. deep, formed of No. 26 gage galvanized iron. Muntin bars showed an external width of $1\frac{3}{4}$ in. The outer panes were bedded on the outside and the inner panes were bedded both sides.

24 *Sample No. 7* was a single-glazed hollow metal sash of the same arrangement of lights as sample No. 6, being obtained from it by removing the inside panes of plain glass, together with the galvanized-iron formed sections which held them in place. This left the muntin bars $1\frac{1}{2}$ in. deep.

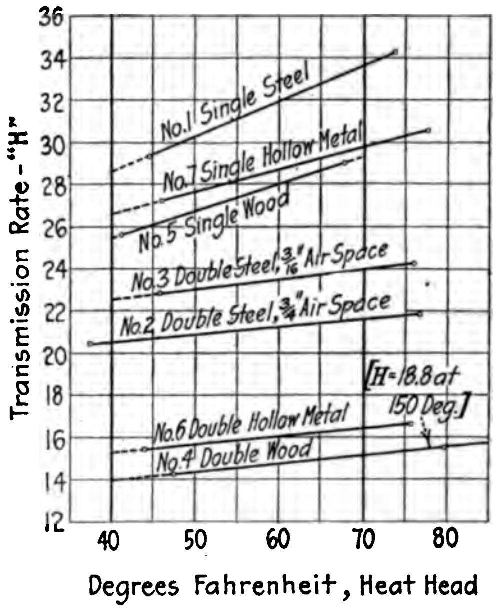


FIG. 6 DIAGRAM OF VARIOUS TRANSMISSION RATES, *H*, FROM TESTS

RESULTS OF TESTS

25 A résumé of the comparative heat-transmission rates, that is, H, of formula [**1**], as deduced from the tests, is given in Tables 1 and 2.

26 The results of the tests have been plotted in the form of curves (Fig. 6). It will be noted that the rate of transmission increases with the heat head and that there is a greater increase, or steeper curve, for the single- than for the double-glazed sash.

27 In order to ascertain the cause of the much greater heat-transmission rate through steel as compared with the wood and the hollow metal sash, temperatures of the glass and sash bars were taken during each test. It was hoped that by this means the paths of heat

transmission would be indicated more clearly. For this purpose a thermometer was inserted in a chamber in a 2-in. cork board, which, in turn, was fitted tightly against the surface whose temperature was to be tested. Although the temperatures thus recorded must have been several degrees lower than actual, it is believed that the comparison of temperatures was fairly well obtained.

28 These tests showed that for a given temperature difference, the solid steel sash bars were hotter than the wooden bars, and by their greater conductivity constituted a direct path for the

TABLE 1 HEAT-TRANSMISSION RATES AT APPROXIMATELY 40 DEG. FAHR. TEMPERATURE DIFFERENCE

Sample	Actual Heat Head, Deg. Fahr.	H Computed at Actual Heat Head	H Corrected to 40 Deg. Fahr. Heat Head	Relative Humidity	Condensation
No. 1 Single-glazed solid steel	44.5	29.3	28.5	36	None
No. 2 Double-glazed solid steel — $\frac{3}{4}$-in. air space	37.4	20.4	20.5	34	None
No. 3 Double-glazed solid steel — $\frac{3}{16}$-in. air space	45.7	22.8	22.5	43	Very slight on air-space side of one outside pane and box side of one inside pane
No. 4 Double-glazed wood — $\frac{3}{4}$-in. air space	47.3	14.2	13.9	49	None
No. 5 Single-glazed wood	41.2	25.6	25.5	44	Hardly noticeable
No. 6 Double-glazed hollow metal — $\frac{3}{4}$-in. air space	43.9	15.4	15.2	54	On air-space side of one outside pane
No. 7 Single-glazed hollow metal	46.0	27.1	26.4	43	All

transmission of heat to the outside air. The hollow metal sash bars were cooler than the solid steel bars but warmer than the wood bars. Indirectly, a further heat loss through the glass itself, in the case of double-glazed sash, can be laid to the solid steel section. Since the amount of heat lost through a given pane of glass depends directly upon the difference in temperature between the two sides, it is evident that in a *double-glazed sash* the rate of heat transmission through the lights must increase with the temperature difference between the air space and the outside air. The tests showed that the temperature difference between the air in the $\frac{3}{4}$-in. air space of the steel sash and the outside air was at least 20 per cent higher than the corresponding temperature difference in a wood sash, evidently due to the effect on the enclosed air of the warmer steel sash bars.

TYPICAL PROBLEM IN RELATIVE COSTS

29 In order to emphasize the practical application of these experiments, a typical problem is given below. In the design of two

TABLE 2 HEAT-TRANSMISSION RATES AT APPROXIMATELY 70 DEG. FAHR. TEMPERATURE DIFFERENCE

Sample	Actual Heat Head, Deg. Fahr.	H Computed at Actual Heat Head	H Corrected to 70 Deg. Fahr. Heat Head	Relative Humidity	Condensation
No. 1 Single-glazed solid steel	73.9	34.2	33.6	28	All
No. 2 Double-glazed solid steel — $\frac{3}{4}$-in. air space......	76.8	21.8	21.6	29	15 outer panes out of 20 showed moisture on air-space side
No. 3 Double-glazed solid steel — $\frac{3}{16}$-in. air space.....	76.0	24.2	24.0	30	One outer pane showed moisture on air-space side; one inner pane showed moisture on box side. Three lights showed both the above
No. 4[1] Double-glazed wood — $\frac{3}{4}$-in. air space...........	79.8	15.5	15.1	31	None
No. 5 Single-glazed wood ..	67.6	28.9	29.3	29	12 lights
No. 6 Double-glazed hollow metal — $\frac{3}{4}$-in. air space.....	75.6	16.6	16.4	36	On air-space side of one outside pane
No. 7 Single-glazed hollow metal	77.7	30.5	29.7	25	All

[1] A test upon this same wood sash with the inside panes of single-thick plain glass instead of double-thick, showed an increase in H of 3 per cent at 70 deg. fahr. temperature difference.

reinforced-concrete buildings there arose the question whether to use single-glazed or double-glazed steel sash. The following data apply to the problem.

Wall openings to be glazed..........................	37,800 sq. ft.
Average inside temperature, heating season.............	70 deg. fahr.
Average outside temperature, heating season............	35 deg. fahr.
Average temperature difference (70 − 35 deg. fahr. =)....	35 deg. fahr.
Heat delivered from steam plant per pound of coal......	8500 B.t.u.

Length of heating season	4850 hr.
Cost of coal per 2000 lb............................	\$4.00
Assumed cost of hot-water-heating system per square foot of heating surface, including heater, based on — 10 deg. fahr. outside temperature..............................	\$0.60

Proposition A, single-glazed steel sash. From the test, H at 35 deg. fahr. = 27.7 B.t.u. per degree difference in temperature per square foot per 24 hours.

Comparative yearly coal bill[1]...............	\$1,750.00	
Comparative initial investment cost of sash erected......		15,600.00
Cost of heating system to supply window loss only, based on 18,400 sq. ft. at \$0.60..........................		11,040.00
Total investment..............................		\$26,640.00

Proposition B, double-glazed steel sash, $\frac{3}{16}$-in. air space. From the test, H at 35 deg. fahr. = 22.3 B.t.u. per degree difference in temperature per square foot per 24 hours.

Comparative yearly coal bill................	\$1,410.00	
Comparative initial investment cost of sash erected......		21,000.00
Cost of heating system to supply window loss only, based on 12,800 sq. ft. at \$0.60		7,680.00
Total investment..............................		\$28,680.00

Proposition C, double-glazed steel sash, $\frac{3}{4}$-in. air space. From the test, H at 35 deg. fahr. = 20 B.t.u. per degree difference in temperature per square foot per 24 hours.

Comparative yearly coal bill................	\$1,260.00	
Comparative initial investment cost of sash erected......		25,750.00
Cost of heating system to supply window loss only, based on 11,500 sq. ft. at \$0.60		6,900.00
Total investment..............................		\$32,650.00

TABLE 3 SUMMARY OF RESULTS

	Yearly Coal Bill	Initial Expenditures		
		Sash	Heating System	Total
Proposition A	\$1750.00	\$15,600.00	\$11,040.00	\$26,640.00
Proposition B	1410.00	21,000.00	7,680.00	28,680.00
Proposition C	1260.00	25,750.00	6,900.00	32,650.00

30 From Table 3 it is evident that an investment of \$2040.00 for B over that for A would result in an annual saving of \$340.00

[1] $\dfrac{\$4.00 \times 37{,}800 \text{ sq. ft.} \times 27.7 \text{ B.t.u.} \times 4850 \text{ hr.} \times 35° \text{ F.}}{2\,000 \text{ lb.} \times 24 \text{ hr.} \times 8500 \text{ B.t.u. (useful coal value)}} = \$1750.00$

worth of coal, or 17 per cent gross on the additional investment. Also, an additional investment of $6010.00 for *C*, over that for *A* would result in an annual saving of $490.00 worth of coal, or only 8 per cent gross on the additional investment, thus eliminating proposition *C*. The choice, therefore, in this particular case lies between *A* and *B*; and whether or not *B* should be preferred will depend largely upon what earnings the owners expect to make upon their other investments. It should be emphasized here that with a different unit cost of coal, a warmer or cooler climate, or a different type of heating system, the conclusion might be changed entirely. Hollow metal sash in this instance was also investigated, but was eliminated from consideration on account of cost.

THE CAUSES AND PREVENTION OF INTERNAL CONDENSATION IN DOUBLE-GLAZED SASH

31 The records kept during the tests on heat transmission showed that condensation appeared on the single-glazed sash when its temperature was low enough to cool the air confined within the test box below the dew point. This moisture was evident on the inside of the panes of glass.

32 Turning to the double-glazed sash, however, the reason for condensation, or its absence, was not so apparent. Where condensation occurred *in the air space*, it was always on the outside pane of glass, which, of course, was the cooler one. The wood sash showed no internal condensation, but in the two steel sash condensation was very marked, and was evident in the hollow metal sample. Moreover, on the metal sash, the deposition of moisture seemed to follow no obvious law, occurring on some lights and not on others. As this inconsistency was most pronounced on the solid steel sash, they were selected as the most suitable to use for investigating the cause and elimination of condensation. The steel sash having a $\frac{3}{4}$-in. air space was chosen for the preliminary test. Afterwards, experiments were also conducted upon the hollow metal sash and wood sash in order to verify or disprove the conclusions of the first test.

33 Holes were bored through several of the outer panes of wire glass, of which some had and some had not shown condensation. Smoke was then blown into the various air spaces through these holes, and leaks to the outside or to adjacent air spaces carefully noted. Each light tested was found to leak considerably, some to adjacent air spaces, others to the front or back, and still others showed combinations of these conditions. In many instances leaks occurred

through the joints in the steel sash bars, as well as through the putty.

34 It was also observed that in lights where condensation was evident, the greatest leakage was into the inside of the box, and that in those lights where no condensation occurred, the greatest leakage was to the air outside the box. Other lights showed serious leaks to both sides and to adjacent air spaces and also showed condensation.

35 Following this clue, the entire sash was reglazed and holes bored in seven of the outer panes and in five of the inner panes. Free communication was thus established either to the external cool air or to the warmer air inside the test box. Each pane of glass opposite those with holes was carefully bedded to confine and control the air leakage to the side having the hole. Smoke tests before and after each run showed that the desired object had been attained by this means. The remaining eight lights were made as airtight as possible on both sides by careful bedding and setting. The sash was then placed on the front of the box as in the previous tests, the temperature raised to about 70 deg. fahr. above that of the surrounding room, and records were kept as before of temperatures, relative humidity, and condensation, the latter being allowed to accumulate for a period of at least eighteen hours. The relative humidity in the test box was maintained as near 40 per cent as possible. That of the air outside varied from 60 to 90 per cent, apparently having little effect upon the results except to hasten or retard the drying-out process.

RESULTS AND OBSERVATIONS

36 A careful examination of the sash showed uniformly the following results:

a Lights in which the air space opened to the outside or *cooler air only*, showed no condensation.

b Lights in which the air space opened to the *warm air* inside the test box showed much condensation.

37 The sash was then reversed, with its outer side toward the interior of the test box, and after a test lasting three days a careful examination showed that the air spaces in which there had been condensation during the previous test were now uniformly dry, while abundant condensation appeared in those which showed none in the first test. It was also discovered that, by varying alternately the temperature in the test box, the processses of condensation or drying were accelerated. In each test, some of the lights which apparently

had been puttied tightly on *both sides* showed condensation, and some did not. (See explanation below.)

38 The experiment on the reversed sash, therefore, confirmed conclusions *a* and *b* above. A similar test was run on the hollow metal sash with the same results.

39 The explanation is simple. Changes in temperature on either side of the sash cause corresponding, though less marked, changes in the temperature of the air space. The pressure of the confined air therefore becomes greater or less than atmospheric pressure and air is correspondingly either forced out of or admitted through the drilled holes, and, if the temperature difference is alternately increased and decreased, a "breathing" action obtains in the air space. A similar action occurs through leaks in actual practice and is due to daily variations in temperature and the longer seasonal changes.

40 Air entering an opening of this kind *from the inside*, coming as it does from the warm interior of the test box or building, becomes chilled and its relative humidity correspondingly increased, condensation necessarily appearing if the cooling is carried on below the dew point. "Breathing," or a repetition of this process, necessarily results in a gradual accumulation of condensation.

41 Conversely, air entering an opening of this kind from the outside becomes heated, and its relative humidity correspondingly decreased, making impossible the precipitation of any moisture.

42 The double-glazed wood sash was tested in the same manner, eight lights being bored, but no trace of condensation in any of the air spaces could be found.

CONCLUSIONS

43 These experiments indicate that condensation in the air space of double-glazed sash can be eliminated almost entirely by connecting the air space directly to the outside air, and at the same time effectively sealing it from the entrance of the warm air within the building. It is not claimed, however, that this method will absolutely prevent condensation at all times, for extreme climatical conditions might arise, such as continued warm, humid weather out of doors followed by a sudden and extreme drop in temperature, which might cool the confined air below its dew point. Condensation caused in this way, however, would be slight and temporary, and instead of accumulating, would eventually dry up with a further change of atmospheric conditions.

44 Whether or not, as a practical problem, double-glazed steel

sash can be constructed with the inside panes sealed and an opening in the outside panes, is worthy of the careful consideration of steel sash manufacturers. In attempting such a design, it is suggested that the following points be considered:

a The opening should be *very small*, equivalent, say, to a $\frac{1}{8}$-in. hole, designed merely for a communication, during temperature changes, between the internal air space and the outside air. A large opening would result in direct heat loss by convection from the air space.

b From the standpoint of the elimination of condensation, the location of the "breathing hole" was found to be immaterial, but a consideration of heat economy would indicate that the bottom of the light is preferable.

c The "breathing hole" should be protected from the weather and dirt.

d A high-grade elastic putty should be used. On account of wind pressure, difference of expansion between steel and glass (about 65 per cent), careless setting, etc., absolute sealing of the inner panes probably will not be accomplished. However, these tests show that a sufficient degree of tightness can be obtained, and that the larger part of the "breathing" will occur through the opening made for this purpose.

e Leaks between air spaces should be eliminated.

45 In an attempt made to make several of the lights airtight on each side, it was found that, on the application of heat, leaks developed, due either to the different expansion coefficients of the materials or to the fact that the pressure in the air space, increasing with the temperature, was sufficient to force a channel of escape or both. Indeed, if such an opening had not been forced, the plain single-thick glass would have been broken by the internal pressure thus developed. An increase in temperature of only 32 deg. fahr. will create a pressure of 1 lb. per sq. in. on the glass, or a total pressure of 200 lb., which was found sufficient by actual experiment to break the single-thick pane. The reason why some of those lights which had been puttied tightly each side showed condensation and some did not is now apparent; that is, on heating up, leaks were developed either to one side or the other, condensation appearing or being absent according to the foregoing laws.

46 The results of these tests should not be construed as indicating that double-glazed wood sash never show interior condensation, but merely that they are superior to steel and hollow metal sash in this respect in the present stage of development of the latter types.

DISCUSSION

FREDERICK J. HOXIE inquired if any experiments had been made where two sashes were placed some distance apart, and the space between them heated. Theoretically, he thought it might be a good idea.

WILLIAM V. DEE[1] stated that he had found that one of the difficulties experienced by manufacturers was that the weep holes in the sash between the two lights of glass let in dirt, which accumulated on the inside in a manner which made cleaning, without taking out the glass, almost impossible. He believed that much of the dirt came from the inside. If a hole were put in the outside, with the colder air coming in from the outside, he thought the condensation on the inside would be materially lessened, if not completely prevented. In the Bates Manufacturing Company's mill a double-glazed steel saw-tooth sash was used, with ventilating sections, which could be opened in series by a mechanical hand operator, and he believed that this would more or less decrease condensation on the inside when the cold air was admitted, which would in a measure bear out the author's tests and the statement that an outside opening was preferable to an inside. The dirt, dust and water problem, however, is still presented, and is one for the sash and skylight manufacturers to work out.

CHARLES H. BIGELOW inquired to what extent double glazing was being used in factories, and also how much condensation there would be on the inside of a concrete roof, and what steps could be taken to prevent it at a reasonable cost.

WILLIAM W. CROSBY recalled a building of considerable size fitted with double-glazed steel sash with ¼-in. air spaces. Only about two-thirds of the usual amount of radiation was supplied, and seldom was the whole of this used, even on the coldest days. The building had a very cold exposure.

W. R. COBB[2] stated that his firm had probably done more double-glazing than any other in the country, and for twenty-three years had done more or less experimenting along that line. There was no doubt a great saving in coal by the use of double glazing. The principal objection to it was the accumulation of dirt between

[1] G. Drouvé Co., Bridgeport, Conn.
[2] With Lord & Burnham Co., 30 E. 42d St., New York.

the two layers of glass, and he would want the double sash so that one could get at all sides of the glass to clean it. His firm built hotbeds with double-glazed sash, and in the coldest weather they required no mats or shutters.

R. W. Weed [1] desired the author to state his experience in comparing a double-glazed steel sash with double sash. He had found that the great heat loss was due not to conduction through the glass, but to convection currents; that is, leakage through cracks along contact lines where the ventilated portion meets the fixed portion, and that these heat losses would run around 80 per cent, whereas that due to conduction would be only about 15 per cent. For that reason, he believed that the double sash was preferable to the double-glazed sash, because it provided insulation against the 80 per cent losses, whereas double-glazed sash provides additional insulation only against the 15 per cent losses.

Dwight Seabury stated that the double sash had an advantage where it was desired to put prism glass on the outside, as there was a chance to keep it clean.

The Author. Referring to Mr. Hoxie's query, I fail to see wherein any particular advantage would be gained by the use of two separate sash placed some distance apart and heating the air space between them. Of course, the practical objections would be serious. And, theoretically, from the standpoint of heat transmission, the value of double sash would be nullified by heating the air space, since the purpose of the latter is to keep the outside pane of glass as cold as possible. The heat loss through the sash is directly dependent, as stated in the paper, upon the difference in temperature between the air on the two sides of the outside pane of glass.

Also, from the standpoint of condensation within the air space, I can see no particular advantage. On the contrary, a considerable disadvantage would probably result from the increased *absolute* humidity of this warmed air within the air space. That is, if this air came from the inside of the building, it would have the same relative humidity as the room enclosed by the sash, the outer pane of the window then acting as a single-glazed sash. Or, if the warm air was in some means supplied by the outdoor air being heated, no better results would be obtained than by the method of openings to the outer air suggested in the paper. In either case the absolute humidity

[1] Detroit Steel Products Co., New York, N. Y.

would be the same. The only advantage apparent to me which would accrue from heating the air space would be that all possibility of condensation on the inside of the inside pane of glass would be prevented, at the expense, however, of the insulating properties of the sash. The inside panes of glass would then act merely as a transparent condensation shield. In fact, there is one manufacturer of skylights who uses an extra pane of glass for this purpose only, a free circulation of air being provided for by openings in the top and bottom of the under light of glass. This results in the temperature of the air in this air space being not greatly below the room temperature, all condensation then occurring on the underside of the upper light of glass only. The air space resulting from this lower pane of glass is in this case, of course, worth practically nothing as a heat insulator.

Whether or not the accumulation of dirt in an air space occurs more from the inside than the outside is a question which would seem to be dependent largely upon the relative amount of dirt and dust prevalent on each side.

With reference to Mr. Dee's experience with the Bates Manufacturing Company: The admission of cold outdoor air to a warm room with air at a fairly high relative humidity, of course, always results in a decrease of the average relative humidity of the room, since the process of heating cold air at a given relative humidity results in a considerable drop of that relative humidity with only a slight increase in temperature. This possibly might account for the decreased condensation for a short period after the cold air was admitted through the ventilators in the sawtooth.

In reply to Mr. Bigelow, I would say that double glazing is used to quite an extent in factories, especially in connection with wood sash. The ultimate economies, from the viewpoint of heat transmission, resulting from the use of double-glazed steel sash, as noted in the paper, do not show up so well on account of the additional initial outlay necessary and the decreased effectiveness of the double steel sash as an insulator. In fact, the installation of the double-glazed steel sash sometimes works out as a disadvantage. Usually there is a gross saving on double glazing of wood sash in this locality of about 32 per cent and, consequently, there is no question about this being a paying investment.

In answer to the question as to how much condensation there would be on the underside of a concrete roof, this, of course, depends upon the relative humidity in the room below and the thickness and insulation of the roof.

Regarding the protection of a concrete roof: Cork boards or quiltings made of seaweed or hair felt are often applied to reduce the heat loss and also the condensation on the underside of the roof.

We are at present investigating, by means of practical tests, the amount of humidity which may be carried under different types of roofs at various heat heads without condensation, and these data will be published when completed.

In reply to Mr. Weed, I would say that the leakage loss through a poorly glazed or set sash, whether it be of wood or steel, may be, of course, considerable, but it would seem to the author as if double sash was as open to this objection as double-glazed sash. Any slight improvement of heat-insulating properties of the former over the latter is probably due to the increased air space, which would be at least $1\frac{1}{2}$ in. as against $\frac{5}{8}$ in. or $\frac{3}{4}$ in. for a double-glazed sash. Also, the sash bars do not form a continuous path for the conduction of heat from the warm air at the back to the cold air at the front. This last point would apply particularly to steel sash. The use of double sash offers, of course, practical difficulties in construction, as compared with double-glazed sash, on account of transoms, stiffness, etc. Double-glazed sash are also less expensive than double sash. Triple glazing is sometimes resorted to in cold climates and in refrigerating plants.

Replying to the question by Charles T. Plunkett, I would state that the tests on heat transmission were made before the holes were put in the lights of glass. These holes were bored later and experiments on condensation conducted regardless of heat transmission. These holes were only $\frac{3}{16}$ in. in diameter and were placed near the bottom of each light, so that the heat loss through them must have been negligible.

No. 1558

VIBRATION IN TEXTILE-MILL BUILDINGS

By G. H. Perkins,[1] Lowell, Mass.
Non-Member

It is the purpose of this paper to outline the principal causes and effects of vibration occurring in buildings occupied by textile-manufacturing processes, with the view of stimulating investigation and promoting discussion on a subject which has received but scant attention from those most vitally concerned. The importance attached to the question of vibration in many other fields of work is strongly emphasized by the efforts made for its elimination from high-speed machinery, such as marine engines, steam turbines, electrical machinery and motor cars. Therefore, if vibration of a textile mill can be shown to produce any prejudicial effect on either buildings, machinery, product or employees, it would seem worthy of serious study. Information on this question must necessarily be of a fragmentary character, and nothing is here offered purporting to be a full analysis of the subject. The many and varied influences contributing to the resultant vibration of an entire mill building containing machinery in motion, make the problem a complex one, and it is often impossible to even ascribe specific results to specific causes.

2 Vibration is unquestionably present in every textile-mill building to some degree, although its amplitude may be so minute that it passes undetected either by the senses or by any material effect. While most textile manufacturers will concede that vibration exists in certain parts of their plants, they are rarely convinced that any real damage is done, except when exaggerated or serious conditions affecting production make remedial action imperative. In many of our older mills, extraordinary conditions of vibration have existed for years without apparent serious results. Like conditions of production, except for vibration effects, are so rarely found and so difficult to create, that it is practically impossible to separate the losses due

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Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

to vibration from those attributable to other causes, or to measure them in economic terms.

NATURE OF VIBRATION IN MILL FLOORS

3 The physical laws underlying the vibration of simple bodies, such as rods and plates, have been clearly defined and are well understood, but their application to the complex structure of a mill floor will not admit of any simple mathematical treatment. It is well known that all structures, simple or complex, have their inherent periodicity or rate at which "free" vibration will take place when they are set in motion, and that this period is dependent upon the mass, dimensions, and elasticity of the body.

4 In the textile mill, vibration most commonly results from the unbalanced resultant of forces set up by certain classes of machines, synchronizing to some extent with the natural period of the structure, or of one of its elements, usually the floor. From any evidence available it would appear that the actual movements do not agree, except for exceedingly brief periods, with the "free" natural vibration of the floor and are therefore of a "forced" character. This is doubtless a fortunate circumstance, so far as integrity of the structure is concerned, and cases are rare where vibration has been the direct cause of a building failure. The historic Pemberton Mill disaster occurred in a building noted for its freedom from vibration.

5 Two distinct classes of movements are commonly found in textile-mill floors:

a Horizontal movements of floor, more or less independent of walls, of comparatively low frequency and large amplitude. These may properly be classed as oscillations

b Movements of higher frequency and less amplitude, often in a vertical plane which may be considered as no more than tremors.

6 While vibrations may exist in all three planes simultaneously, they are under textile-mill conditions more apt to be strongly emphasized in only one direction. These motions may properly be considered as truly harmonic in character, and their period, frequency and amplitude defined as from a sine curve.

CAUSES OF VIBRATION IN TEXTILE MILLS

7 The principal factors contributing to vibration in textile-mill buildings are:

I Unbalanced machines, of which the following are the most important:

a Looms. The lay and pick motions of practically all looms are unbalanced reciprocating movements in a horizontal plane. Harness motions, including Jacquard heads, are reciprocating movements in a vertical plane. The frequency of these motions ranges from 90 to 180 picks per minute, depending upon type of loom and class of work

b Noble worsted combs. The dabbing brushes are actuated by vertical reciprocating motions at from 1000 to 1200 r.p.m., with two brushes per machine. Cotton combers, on which the nipping motion oscillates at from 100 to 125 nips per minute, giving slight vertical throw

c Mules. The carriage of a mule moves with a variable alternating motion in a horizontal plane at from 4 to 6 draws per minute

d Unbalanced drums, cylinders, rolls and pulleys are frequent causes of local vibration. Worn or defective gearing also often causes trouble on heavy roll drives, as on calenders

e Heavily loaded trucks moving over light floors cause deflection of beams and plank and some vibration.

II Inherent weaknesses in the structure, such as thin walls, light floors with long spans, and unsuitable connections between floors and walls.

III Poor soil conditions contributing to relative freedom of foundations and footings.

IV Sympathetic vibrations originating outside the building. These are frequently set up by water falling over dams, reciprocating engines in adjacent buildings, or by railroad trains. Their mode of transmission is often obscure, although it is sometimes direct, as through pipe lines or solid ledge.

VIBRATION RECORDS

8 In the analysis of specific cases of building vibration, graphical records of the motion produced have proved of much value. Such

records showing the period, amplitude and cyclic·variations of the vibrations serve as a guide in tracing their origin and assist materially in estimating the effects produced on building and machinery. Numerous instruments for this purpose, which need not be described here, have been constructed, involving the basic principle of the seismograph, originally developed for the study of earthquake phenomena.

9 The partial records presented are portions only of results obtained in the course of investigations carried on under various conditions in textile-mill buildings. As it is obviously impossible to present any extended records, the purpose of the paper will best be served by confining the results shown to those obtained from one typical weaving mill.

Description of building:

Length, 270 ft. 0 in.	Spans: 4 of 26 ft. each
Width, 116 ft. 0 in.	1 of 12 ft.
Basement, 10 ft. 6 in.	Bays, 7 ft. 6 in.
Two Stories, 16 ft. each	Roof of Saw-Tooth Type
Walls: 24 in. on first story, 20 in. on 2d story	
Pilasters, 28 in. wide	Windows, 5 ft. 0 in. wide
Floor Beams, 10 in. by 18 in.	Floor Plank, 4 in.

10 Floor plans and arrangement of machinery are shown in Figs. 1 and 2. Stations at which records were taken are numbered on floor diagrams. It will be noted that all looms are arranged across the mill and records show longitudinal vibration only. Records taken with instrument set transversely showed no appreciable motion on either floor.

11 Records taken in this mill after the installation of additional looms, indicated a decrease in the maximum amplitude, fewer cyclic variations and a somewhat higher average amplitude. (See Figs. 3 to 7.) The reduction in the maximum double amplitude was from 0.059 in. to 0.045 in. This is not an unusual experience when mills are equipped in installments. A mathematical analysis of the unbalanced horizontal force applied to the floor by a 72-in. Knowles worsted loom running at 120 picks per min. showed an average value of 154 lb., with a maximum of 262 lb. This gives some conception of the magnitude of the total force acting when a number of looms are in synchronism.

12 In all cases any movement estimated by the senses was far greater than actually recorded. This bears out the fact that even

small and harmless vibrations are often responsible for apprehension on the part of the operatives.

EFFECTS OF VIBRATION

13 In presenting information covering the effects of vibration, the writer wishes to acknowledge the assistance rendered him by the Aberthaw Construction Company, of Boston, Mass., who deserve much credit for undertaking an extensive investigation of this unpromising subject. They have placed at the writer's disposal all of the interesting material accumulated by them to date, covering a wide range of industries.

14 Objectionable effects of vibration in textile mills have been noted by various observers as follows:

a Settling of foundations and footings on poor soils, with resultant cracking of walls and unleveling of floors, due to a "shaking-down" process of buildings subject to excessive vibration

b Effect on operatives. This is usually manifested by apprehension of the failure of the building, loss of efficiency due to fatigue, and the serious effect of continued vibration on the nervous system

c Effect on production of textile machinery:

Carding Wider card settings are found necessary on vibrating floors, causing uneven and poor work.

Spinning Vibration of ring rails on frames spinning either fine numbers or coarse waste yarns, causing breaking down of ends.

Turning of roving bobbins on skewers of spinning frame creels, causing roving to unwind and kink. "Chatter" of rolls on long frames.

Weaving Vibration of tension weights on narrow fabric looms, causing dropping of weights when loom is at rest

Shaking of similar tension weights on elastic webbing looms, permitting rubber warp to slacken and causing defective work by allowing excess elastic to enter fabric

Increased sensitiveness of "feeler" mechanisms in filling changing looms.

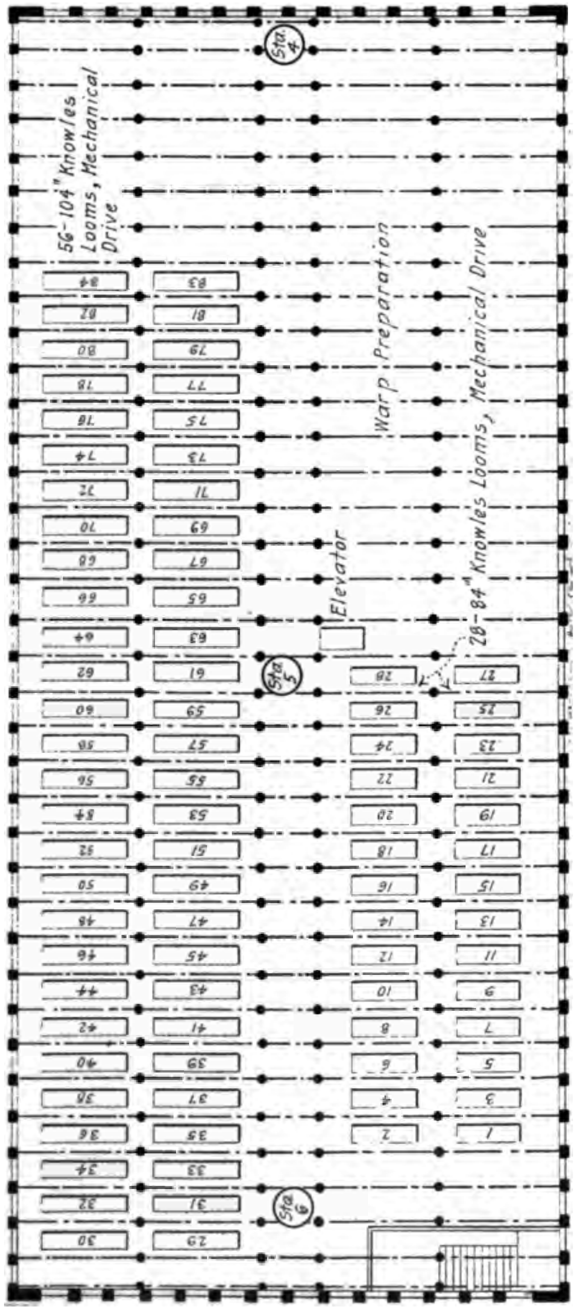

FIG. 1 FIRST-FLOOR PLAN

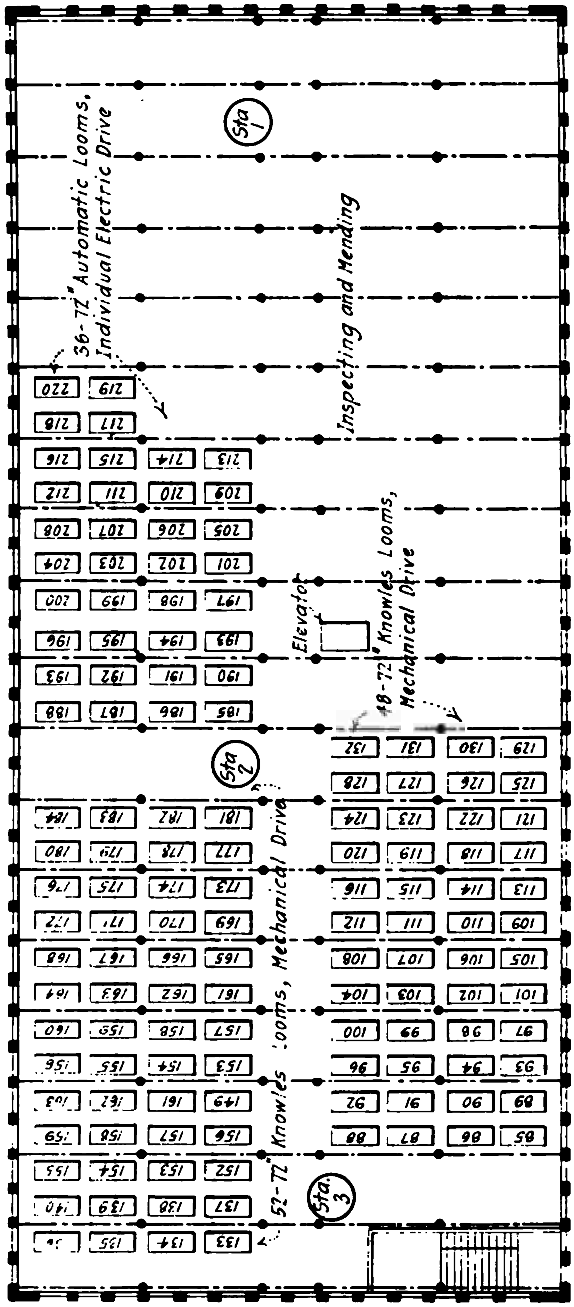

FIG. 2 SECOND-FLOOR PLAN

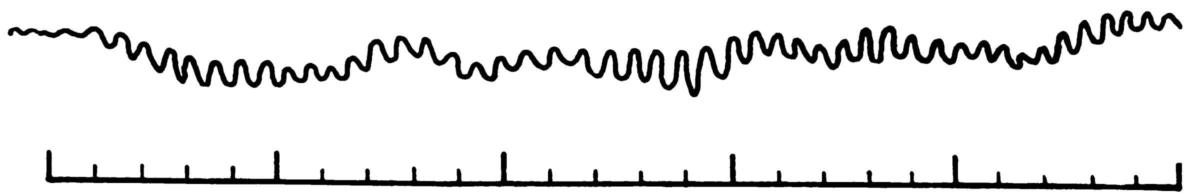

Fig. 3 Record No. 1, Taken at Station 3 on Second Floor. Shows High Average Amplitude. Period, 108 to 120 per Min. Average Loom Speed, from 110 to 120 Picks per Min. Maximum Double Amplitude, 0.03 In.

Fig. 4 Record No. 2, Taken at Station 3 on Second Floor. Shows Low Average Amplitude Under Same Conditions. Records Taken at Station 2 Showed Same Characteristics as Those at Station 3

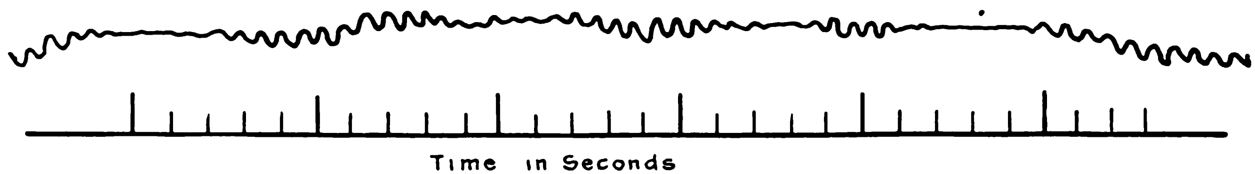

FIG. 5 RECORD NO. 3, TAKEN AT STATION 3 ON SECOND FLOOR. SHOWS EVIDENCE OF CYCLIC DISTURBANCES ABOUT EVERY 8 TO 10 SEC.

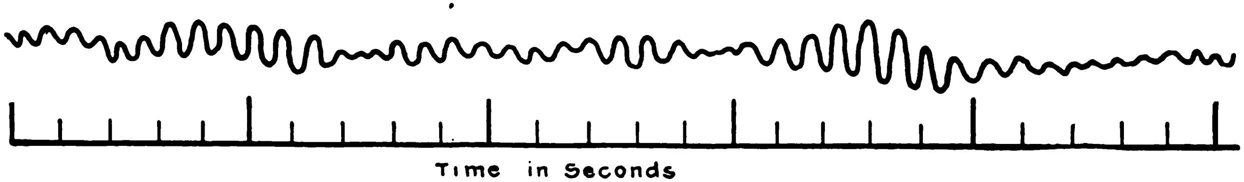

FIG. 6 RECORD NO. 4, TAKEN AT STATION 1 ON SECOND FLOOR, ABOUT 50 FT. AWAY FROM NEAREST MACHINE
Indicates How Readily Vibrations Are Transmitted Through Floor Undiminished in Amplitude. It May Be Noted That the Group of 36 Looms Nearest to Station 1 are Individually Electrically Driven. Maximum Double Amplitude = 0.045 In.

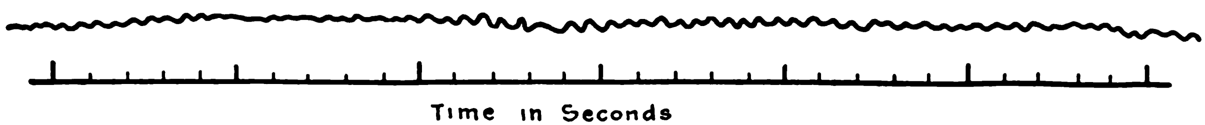

FIG. 7 RECORD NO. 5, TAKEN AT STATION 5 ON FIRST FLOOR
Shows Typical Record at This Point with Vibrations of Slightly Less Frequency and Small Amplitude. The Average Speed of the 104-in. Looms on This Floor was 90 Picks per Min., and the Speed of the 84-in. Looms was 98 Picks per Min.

Finishing Chattering of "doctor" blades on "back-filling" mangle, due to passing trucks, causing uneven distribution of starch on goods and imperfectly cleaned roll

Wave movement of water in pan of "damping" machine, causing uneven distribution of moisture on goods by spraying brush.

Miscellaneous Breaking of electric lamp filaments, making spring suspensions necessary. Difficulties in use of sensitive instruments, such as balances, scales, testing apparatus, etc.

15 Many consider that the matter of increased power consumption of shafting due to vibrating floors is important, but no authentic tests covering this point are available.

ELIMINATION OF VIBRATION

16 The elimination of vibration of an existing building of mill construction is usually attempted by:

a Stiffening and strengthening of floors by additional columns or trusses and making more secure connections to walls; also by stiffening outside walls with additional pilasters, or in the case of wooden-frame buildings even by braces or guys.

b Balancing of all machines possible and cushioning or absorption of the shocks of machines in which an unbalanced component seems unavoidable. More attention could advantageously be given to the matter of balance in certain classes of textile machines by their builders. Incidentally, the attendant reduction of noise would be beneficial.

17 The well-known advantages of reinforced-concrete buildings with particular reference to rigidity will not be discussed here, but such construction undoubtedly would obviate a large portion of the difficulties outlined above. There are certain conditions, however, where too great rigidity has been found to be most undesirable for certain classes of textile machinery as at present constructed. Such a large proportion of our textile plants occupy buildings of "slow-burning" mill construction, that it is the present purpose to show existing conditions rather than emphasize the advantages of any one type of construction.

EFFECT OF RIGID FLOORS ON TEXTILE MACHINERY

18 The data available covering the comparative operation of textile machinery under similar conditions on rigid and flexible floors are extremely limited, but experience seems to emphasize the following:

a In weaving heavy fabrics, the breakage of loom parts has been excessive when looms are mounted on concrete floors with no provision for absorbing the shocks produced. Cushioning the loom feet with absorbents such as wood, cork or rubber, has been found absolutely necessary under these conditions

b Transmission of the noise of machinery to offices, laboratories, etc., has been found exceedingly annoying in some concrete buildings.

NEED FOR FURTHER STUDY

19 The problem of mill vibration, while not new, has yet to receive either full recognition or sufficient serious attention. The resulting economic losses are unquestionably large, if analyzed from all standpoints, and the elimination of vibration is consequently an important factor in the efficiency of the plant.

20 All information at present available would seem to point to a general agreement as to the desirability of avoiding excessive vibration in manufacturing buildings. There is evidence of a growing interest in the subject, but the scarcity of authentic results from experimental work along this line, carried on under actual manufacturing conditions, makes exhaustive tests and observations necessary before any general conclusions can be reached.

DISCUSSION

Charles H. Bigelow told of one building, erected about nineteen years ago on piles, that swayed back and forth in time with two 1200-kw. engines 75 times a minute, so that the window weights could be heard to rattle back and forth in their casings. The actual movement of the corner of the building in relation to a pile of lumber 50 ft. away on a marsh was $\frac{3}{8}$ in. When the two engines were running, the motion would come to a maximum, and then would cease as the engines opposed each other. In another building he had found a 4200-h.p. engine under which the whole foundation moved around about ½ in. This was stopped by anchoring the foundation to other parts of the building. Vibration was certainly a serious matter; but how much power was lost in the machinery he could not say.

WM. W. CROSBY asked that further information be given by the author as to the curves shown in the paper, and that the plane of vibrations be defined. He also wished to know whether the looms were driven by motor or belt. He knew of a concern which had several foreign mills built of rigid construction with steel beams and solid brick arches. They had hesitated about building a new mill with heavy wooden beams and plank according to our slow-burning construction because they thought the machinery would not withstand the vibration. After several years' operation, the manager told him that as yet no undue wearing of parts had been detected which might be ascribed to vibration.

He further stated that he had heard of employees making serious objections to working on floors which were too rigid. Help that had been working on wooden floors had raised all sorts of objections because they had been put to work on concrete floors.

HENRY A. HALE, JR., stated that it had been found that the accidents to employees in spinning and weaving operations were largely due to carelessness. He thought that nothing engendered carelessness more than fatigue, and if by eliminating vibration this could be cut down, and thereby the number of accidents, the resulting economy would be something not to be overlooked.

IRA H. WOOLSON inquired of the author whether vibration was a matter of common complaint with the operators — whether it was physically objectionable to them.

MAURICE DEUTSCH[1] (written). To the writer the most striking facts revealed in the paper are the remarkably low maximum machine speeds and resulting low vibration frequencies and very large amplitudes compared with similar factors observed in other factories devoted to textile manufacture. The American looms referred to operate at speeds varying from 98 to 120 picks per min., producing approximately similar vibration frequencies in the floors and amplitudes as high as 0.045 in. and 0.059 in.

[1] Civil Engineer, 50 Church St., New York, N.Y.

It is often asked how the English make such good yarn out of poorer cotton than that used in America. Is this question not partly answered when it is found that many of these mills are of fireproof construction, some with concrete floor and granolithic portland-cement surface coat, some with heavy, smooth, flat stone slabs laid in brick arches, and that great attention is given to the cushioning and arrangement of machinery?

In such a sawtooth weave shed in England, known to the writer, looms standing on flagstones were running at 220 to 225 picks per min. It should be possible greatly to increase even this speed. If such high amplitudes as those given by the author existed with a frequency such as the above, or that of the ordinary reciprocating printing press, it would indeed be dangerous to the machinery as well as to the building containing it, and would make textile operations impossible. But if these high amplitudes of 0.045 in. were reduced, the frequencies could be very much increased without detriment. Whether or not vibrations are felt depends not only upon the amplitude but also upon the frequency. Lack of rigidity in a structure contributes to lowering the frequency and hence reducing the possibility of vibrations being felt, but this condition may also increase the amplitude — accordingly increasing the possibility of the vibrations being felt — whereas rigidity of structure would tend to increase the frequency and reduce the amplitude. From this point of view, there must, therefore, be some crucial point for a given class of manufactures where the factors of looseness and of rigidity of structure meet. Just how rigid a structure should be will depend upon the material fabricated and other conditions, but generally speaking, for textile mills at least, rigidity of structure should be sought for. Whatever may be done to vary the coincidence in periodicity and the rate of extinction of the mass being vibrated, the less will be the detrimental effect of vibrations upon machine, fabric or building. One of the most effective methods, the writer believes, for so minimizing vibrations in a rigid structure is by the proper application of composite absorption material between the machine and the structure.

From independent tests made by the writer, also by Prof. E. E. Hall, of Berkeley, Cal., and W. P. Digby, of London, England, it has been found that vibrations having a frequency of about 15 per sec. and a double amplitude (d. a.) of about 0.002 in. have generally been the cause for complaints made by the occupants of houses in the vicinity of railway trains or certain types of machinery; also that a frequency of about 7 per sec. and a double amplitude of about 0.0012 in. to 0.0016 in. can just be felt.

The important question of the human susceptibility to vibration is one to which little study has been given in this country. In England, however, some very interesting observations have been made by Mr. Digby and Capt. H. R. Sankey.

How far beyond the limit of perceptibility the vibrations recorded by the author are, may be somewhat appreciated when compared to Professor Hall's observations that when the frequency is 2 per sec. the double amplitude must be approximately 0.004 in. before it can be felt. He further states that for not too extreme limits the amplitude necessary in order that vibration be felt, roughly varies inversely as the vibration frequency. The writer, from many observations made in buildings, has found that amplitudes as low as 0.0002 in. when the frequency is over 10 per sec., are a genuine source of annoyance to occupants.

From observations made by the writer at Schenectady and concurred in by Professor Hall and also by seismologists, it has been found that vertical vibrations damp out very much more rapidly than do horizontal. It is the latter which are more apt to produce detrimental effects to the structure of a textile mill, while the former may produce the greatest effect on the machine and the fabric.

It has also been found that a vertical vibration of 0.15 in. with a frequency of 8 per sec. — of 0.012 in. with a frequency of 28 per sec., is just sufficient to counteract the force of gravity. Street-traffic vibrations have been found to vary in frequency from 6 to 20 per sec., with an order of magnitude of amplitude of 0.002 in., depending upon the cause of the vibrations, the character of the pavement, the nature of the soil, etc.

Morton C. Tuttle[1] wrote giving particulars of some experiences brought out in the course of an investigation of the effects of vibration conducted by the Aberthaw Construction Company.

In one flat-walled brick cotton mill, vibration and side-swaying had been eliminated by building firmly founded pilasters into the walls at every second bay, and by using heavier floors in the upper (3d) story. In another mill, satisfactory results were obtained by embedding the machines in concrete.

One mill had looms installed in both a weave shed and a four-story building adjoining. The production per loom in the weave

[1] Secretary, Aberthaw Construction Company, Boston, Mass.

shed over a period of several years averaged 15 per cent greater than that per loom in the building, and was attributed to the greater stability of the looms in the weave shed. Cases were also reported where removal to new mills made possible speed increases of from 10 to 25 per cent.

There was a certain diversity of opinion among the correspondents reporting, but this, it appeared, depended upon the type and character of building. It was made quite evident, however, that the most serious cases of vibration were found in old buildings or those unsuited for use as textile mills.

THE AUTHOR, in reply to Mr. Crosby's queries, stated that the records given in the paper showed longitudinal vibration only. He gave the particulars of the type of drive in the various banks of looms: All looms were mechanically driven, except the group of 36 individually motor-driven looms, shown near the center of the second-floor plan (Fig. 2). The record taken at Station 1, about fifty feet away from this group of looms, showed that the vibration was transmitted undiminished in amplitude through the vacant floor. The probability of synchronism in motor-driven looms is undoubtedly greater than in belt-driven machines.

In answer to Mr. Woolson's question, he said that in nearly all of the cases that had come to his attention, new help usually found vibration quite objectionable at first, but after a time they became accustomed to it. There were certain of the older mills with which he was familiar, where the vibration conditions were so extreme as to cause a decided feeling of apprehension in all new employees or strangers passing through the plant.

From Mr. Deutsch's remarks it might be inferred that if the amplitude of the vibrations recorded could be reduced by increased rigidity of the building, the frequencies could be largely increased without detriment to building or machines.

The frequencies in this case are low, but are fixed by the inherent speed limitations of broad looms, which cannot be successfully operated at much higher speeds than those given, even when much greater rigidity is obtained in the floor. The looms to which Mr. Deutsch refers as operating at 220 to 225 picks per minute must have been of narrow width; while those in the present case range as wide as 104 in.

No. 1559

STANDARDIZATION OF MACHINE TOOLS

SOME SUGGESTIONS REGARDING STANDARDS OF SPEED AND FEED SERIES AND STANDARDIZED POWER FOR MACHINE TOOLS, ETC.

By Carl G. Barth, Philadelphia, Pa.
Member of the Society

While engineers and manufacturers are realizing more and more the desirability — yes, necessity, for standardizing, not only their own individual methods and product, but also to some extent the methods and product of each group engaged in the same line of manufacture; and while encouraging progress is being made by associations of manufacturers of such modern products as automobiles, electric motors and lighting apparatus, and no doubt by other associations of which the writer has no personal knowledge, little, if anything, has been accomplished by the machine-tool builders of this country.

2 The object of this paper is to try to enlist the coöperation of this Society in encouraging this group of manufacturers to adopt certain standards for machine tools for even only the approximate attainments of which the writer has had to spend a large sum of money in each of a number of machine shops in which he has found such standardization highly desirable, if not to say absolutely necessary, as one of the steps in the installation of a more or less complete Taylor System of Scientific Management as practised by him. It is also hoped that machine-tool-using members of the Society may be encouraged to insist upon conformity to these standards when issuing specifications for the purchase of future machinery.

3 This is not the writer's first attempt, either to make his ideas generally known, or directly to enlist the machine-tool builders' interest in the matter, for he presented them quite fully in a discussion of the paper by L. P. Alford, Mem.Am.Soc.M.E., on Standardization of Machine Tools before the meeting of the National Machine

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Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

Tool Builders' Association, in New York, December, 1912; and he has both before and after that done a great deal, and with more or less success, to win over to his views a number of machine-tool builders with whom his activities have brought him into contact. However, the time seems to have come when the matter should be submitted to this Society for its consideration and discussion. It is hoped the result will be either an approval of the ideas submitted and a propaganda for their universal adoption, or the bringing forth of substitutes which the writer may either accept as unqualified improvements, or at least as equally satisfactory.

OBJECTION TO STANDARDS

4 Before proceeding to any definite recommendations, it will be well to consider the objection so often raised to the adoption of stand-

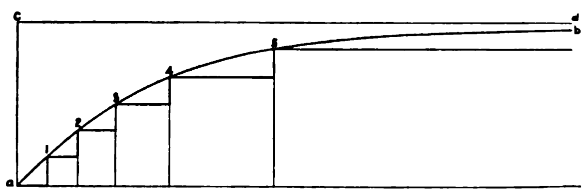


Fig. 1 Continuous vs. Step-by-Step Development

ards of any kind — that standardization blocks the way for further development and improvement.

5 If this were unqualifiedly true, standardization would almost be a crime; but if we adopt a standard merely as representative of the best a trade or profession knows of at any one time, with the understanding that as soon as a *decided* improvement is brought out, a new standard will be adopted to parallel temporarily and eventually replace the former standard, the danger of stagnation will be obviated.

6 Perhaps my idea will be understood more readily by reference to Fig. 1, in which the smooth curve *ab* tending towards the asymptote *cd* represents a gradual development with immediate adoption of improvements in any line towards an eventual highest possible attainment; and the broken line whose corners 1, 2, 3, 4 and 5 lie in this curve, the only occasional adoption of such improvements, when these have attained enough importance to make it worth while to change from one temporary standard to a new one, both standards then being in use during the period of change.

7 Such a period will, to be sure, seem exceedingly inconvenient after we have once tasted of the fruits that the use of standards bring forth, but it will be as nothing compared with the troubles and expense of dealing with a lot of unstandardized equipment all the time. "Where ignorance is bliss, 'tis folly to be wise," surely does not hold good in these matters.

STANDARDIZATION OF MACHINE EQUIPMENT OF SHOP AS A WHOLE

8 By this illustration I have particularly in mind the exceedingly one-sided manner in which most users of machine tools try to get the most up-to-date and best (if those users are not the kind that simply allow an unaided and ill-advised purchasing agent to buy on price alone) whenever they have to add to their equipment, regardless of whether or not the most up-to-date machine of a kind fits in with older equipment that will not be sufficiently behindhand to be discarded and replaced for years to come.

9 During his many years of experience the writer has gradually been forced to see an enormous advantage in also standardizing the machine equipment as a whole, aside from the standardization of the mathematical and certain constructive features of each machine. Wherever possible his effort is to have only the very same make of machines in a certain productive group, and in the expansion of such a group to add machines of exactly the same design, even to the extent of having a manufacturer furnish what is no longer his most recent product — only, however, if this or some other manufacturer's most recent product is not a sufficient improvement over the old to warrant the introduction of a new machine as a new standard for the work to be done, with the intention that this type will eventually crowd out and replace the older type.

10 As an encouraging example of this, the writer recently had the satisfaction of seeing a newly acquired client award the contract for a dozen large milling machines to one bidder, while the original intention had been to divide the order between two or more bidders, in order to obtain quicker deliveries as the paramount consideration.

STANDARD SPEED SERIES

11 As in the present state of the science and the art of cutting metals, it is, on the average, impossible to determine the most economical or suitable speed and feed at which to run a machine for any set

of conditions, closer than a certain percentage, it is by this time universally accepted by those who have a right to an opinion in the matter that the available speeds of a machine should be in a geometrical progression. A discussion of this will not be undertaken by the writer unless provoked by some one else, though even at a recent date the futile attempts of some designers to arrange a series of speeds in a geometrical progression are conspicuous. The reason for this is that a number of our machine-tool builders, while strong men of inventive genius or business ability who have worked their way to the front from the shop, have never acquired sufficient theoretical engineering training to appreciate the mathematical problems involved, fundamentally simple though they are, and usually employ draftsmen and designers equally deficient along their line, many of whom, on the other hand, display ingenuity in their work that is little short of marvelous.

12 It being first agreed that a geometrical series of speeds should be provided for any one machine, the writer's idea is that a universal speed series should be adopted by all machine-tool builders for all machines such as lathes, boring mills, milling machines, drill presses, etc. that are provided with a spindle for either the work or the cutting tool, and which, except in the case of certain single-purpose machines not included in this discussion, may be rotated at different speeds; for our present knowledge does not warrant us in asserting or assuming that a progression of speeds for one kind of these machines should have a lesser or a greater constant ratio than for any other kind; and, evidently, if there is no reason for making them different, there is every reason for making them just alike. The question then becomes, what should be the constant ratio of such a universal geometrical progression of speeds?

13 As far back as 1888 the writer designed for William Sellers & Company, Inc., of Philadelphia, a large lathe which had 30 speeds in a practically perfect geometrical progression, with a ratio of a little over 1.15, obtained by a correct relation between a 5-step cone, two back-gear reductions and two forward speeds of the countershaft. Again, in 1892, he designed another with 36 speeds which had a practically constant ratio of but little over 1.14, obtained by a 6-step cone, two back-gear reductions and two forward speeds of the countershaft.

14 These designs, only the latter of which was actually built, prove that this company was able long ago to appreciate the theoretical value of close speed regulation, though at that time the prevailing

style of shop management, or rather, lack of shop management, throughout the country was such that this knowledge was undoubtedly of no practical value to the operator, even in the shop of that company. In fact, when the writer finally got some real ideas about such matters during his first two years of association and co-operation with Mr. Taylor at the works of the Bethlehem Steel Company, immediately after Mr. Taylor and Mr. White had made the discovery of the high heat treatment of tungsten steels, the writer realized that he had overreached himself in those two lathes. Therefore, in designing some special lathes for the Bethlehem Steel Company for the better utilization of the new high-speed tools, he adopted 1.2 as a more rational ideal, and he has never since found any reason for deviating materially from this ratio; though it was several years later before he recognized the desirability of an absolutely constant ratio, not only for all lathes, but, as referred to above, for all machine tools with a revolving spindle.

15 However, if we construct a geometric progression with this simple ratio 1.2 and beginning with 1, all subsequent terms of this will naturally be found to be anything but simple numbers, thus: 1 1.2 1.44 1.728 2.074 2.488 2.986 3.583 4.300 5.160 6.192 7.430 8.916 10.70 12.84 15.41 18.49 22.19 26.62 31.95 38.34 46.00, etc.

16 In reality, therefore, we will have a simpler progression by slightly modifying this ratio 1.2 such that the fifth term becomes 2 instead of $1.2^4 = 2.074$, for then the whole progression becomes

$$1 \quad 2^{\frac{1}{4}} \quad 2^{\frac{1}{2}} \quad 2^{\frac{3}{4}} \quad 2 \quad 2\cdot 2^{\frac{1}{4}} \quad 2\cdot 2^{\frac{1}{2}} \quad 2\cdot 2^{\frac{3}{4}} \quad 4 \quad 4\cdot 2^{\frac{1}{4}} \quad 4\cdot 2^{\frac{1}{2}} \quad 4\cdot 2^{\frac{3}{4}} \quad 8, \text{etc., or} \quad \ldots \text{[I]}$$

1 1.1892 1.4142 1.6818 2 2.3784 2.8284 3.3636 4 4.7568 5.6569 6.7272 8, etc.

with every fourth term a power of 2, the simplest of all numbers except unity itself.

17 This is, to the writer, an exceedingly attractive progression, and represents his ideal for some six years past, particularly as he cannot help comparing the revolutions of a rotating spindle with the vibrations of a musical string; and in a piano, for instance, the number of vibrations of the strings also form a geometrical progression with 2 as a constantly recurring factor, namely, as the ratio between strings that sound notes an octave apart. Because of this similarity, the writer, for lack of a better term, has named a series of speeds conforming to the above geometrical progression, a *chromatic* speed series, though the term *chromatic* in connection with a musical scale has

reference to the geometrical progression involved, rather than to the ratio 2 of the number of vibrations of two notes in an octave.

18 It may be, however, that the majority of machine-tool builders will consider the ratio $\sqrt[4]{2} = 1.1892$ unnecessarily small, and would deem a somewhat larger ratio preferable in order to obtain a larger final ratio between the slowest and the fastest speeds of a machine which, for one consideration or other, may have to be arranged with a rather limited total number of speeds, as, for instance, a lathe with a single back-gear reduction, or a small drill press or milling machine with no back-gear reduction at all.

19 The writer's answer is that he has had in mind also, as a possibly more generally acceptable progression, one with the constant ratio $\sqrt[3]{2} = 1.2599$; that is,

$$1 \quad 2^{\frac{1}{3}} \quad 2^{\frac{2}{3}} \quad 2 \quad 2\cdot 2^{\frac{1}{3}} \quad 2\cdot 2^{\frac{2}{3}} \quad 4 \quad 4\cdot 2^{\frac{1}{3}} \quad 4\cdot 2^{\frac{2}{3}} \quad 8, \text{ etc., or} \qquad [II]$$
$$1 \quad 1.2599 \quad 1.5874 \quad 2 \quad 2.5198 \quad 3.1748 \quad 4 \quad 5.0397 \quad 6.3496 \quad 8, \text{ etc.}$$

which is as much entitled to the name chromatic as is progression [I].

20 In fact, in recently rebuilding eighteen vertical single-spindle drill presses for a client company, he made the speeds of these presses conform to this progression, in connection with a fixed gear reduction of 3 to 1, a 3-step cone, and a 2-speed countershaft, thus giving them only six speeds in all. The presses had originally a back-gear reduction in addition to the fixed-gear reduction, a 4-step cone, and a single-speed countershaft, or eight speeds in all.

21 Again, as a compromise between progressions [I] and [II] and to favor a possible preference for a progression with 10 rather than 2 as a periodically recurring ratio, the following progression may also be looked upon as a candidate:

$$1 \quad 10^{\frac{1}{12}} \quad 10^{\frac{2}{12}} \quad 10^{\frac{3}{12}} \quad 10^{\frac{4}{12}} \quad 10^{\frac{5}{12}} \quad 10^{\frac{6}{12}} \quad 10^{\frac{7}{12}} \quad 10^{\frac{8}{12}} \quad 10^{\frac{9}{12}} \quad 10^{\frac{10}{12}} \quad 10^{\frac{11}{12}} \quad 10 \quad 10\cdot 10^{\frac{1}{12}}, \text{ etc., or} \qquad [III]$$
$$1 \quad 1.2115 \quad 1.468 \quad 1.778 \quad 2.154 \quad 2.610 \quad 3.162 \quad 3.831 \quad 4.642 \quad 5.623 \quad 6.813 \quad 8.254 \quad 10 \quad 12.115, \text{ etc.}$$

The constant ratio of this progression $\sqrt[12]{10} = 1.2115$ is very close to the geometrical mean of the ratios of progressions [I] and [II], which is $\sqrt{2^{\frac{1}{4}} . 2^{\frac{1}{3}}} = 2^{\frac{7}{24}} = 1.224$.

22 And finally, a certain interesting simplicity would also be introduced by the adoption of the constant ratio $\sqrt[10]{10} = 1.2589$, thus:

$$1 \quad 10^{\frac{1}{10}} \quad 10^{\frac{2}{10}} \quad 10^{\frac{3}{10}} \quad 10^{\frac{4}{10}} \quad 10^{\frac{5}{10}} \quad 10^{\frac{6}{10}} \quad 10^{\frac{7}{10}} \quad 10^{\frac{8}{10}} \quad 10^{\frac{9}{10}} \quad 10 \quad 10\cdot 10^{\frac{1}{10}}, \text{ etc., or} \qquad [IV]$$
$$1 \quad 1.2589 \quad 1.5848 \quad 1.9953 \quad 2.5119 \quad 3.1623 \quad 3.981 \quad 5.012 \quad 6.310 \quad 7.943 \quad 10 \quad 12.59, \text{ etc.}$$

23 It will be noticed, however, that there is in reality a theoreti-

cal difference only between progression [II] and this progression, the tenth number (7.9431) of this being less than 0.8 of one per cent smaller than the tenth number (8) of the former.

24 But the best comparison of these four progressions is made by plotting them on a logarithmic scale, as in Fig. 2, on which is also plotted the progression that represents the relations between the number of vibrations of the strings of a piano, the constant ratio of which progression is $\sqrt[12]{2} = 1.05947$.

25 Having thus indicated that there may be a choice of standards, the writer will further on give some pretty strong reasons for favoring progression [I], admitting at the same time that under certain conditions every other term may be omitted, thus leaving the progression:

$$\begin{array}{lllllllll} 1 & \sqrt{2} & 2 & 2\sqrt{2} & 4 & 4\sqrt{2} & 8 & 8\sqrt{2} & 16, \text{ etc., or} \\ 1 & 1.4142 & 2 & 2.8284 & 4 & 5.6568 & 8 & 11.3137 & 16, \text{ etc.} \end{array} \quad \ldots\ldots \text{[V]}$$

STANDARD FEED SERIES

26 What has been said above in discussing the adoption of a standard speed series holds equally good for a standard feed series, except for the most up-to-date designs of lathes, in which the feeds for plain turning are obtained as a constant fraction of the screw-cutting feeds. For these latter it is suggested that both the cross feeds and the longitudinal feeds be made the same fraction of the screw-cutting feeds for all lathes regardless of size. Such an absurdity as a reputable manufacturing concern putting on the market at the same time a 20-in. lathe with this fraction about 0.25, and a 16-in. lathe with this fraction about 0.4, is indeed a severe indictment of the whole lack of system, both of designing and of buying machinery, that still prevails. It may be well to mention here that the writer considers it a mistake on the part of American machine-tool builders that they have practically abandoned the belt-cone feed in favor of all-gear feeds. While the modern gear feed box in its various forms, with its admirable means for effecting changes quickly, has become indispensable for certain classes of work, the simplicity and effectiveness of a sufficiently powerful and well-proportioned belt-cone feed is just the thing for certain other classes of work. Thus the 18 rebuilt drill presses referred to in Par. 20 were provided with a five-step feed cone giving feeds from a minimum of 0.011 in. to a maximum of 0.022 in., as for the class of work they were to do quickness of change of either feed or speed would have been of no value.

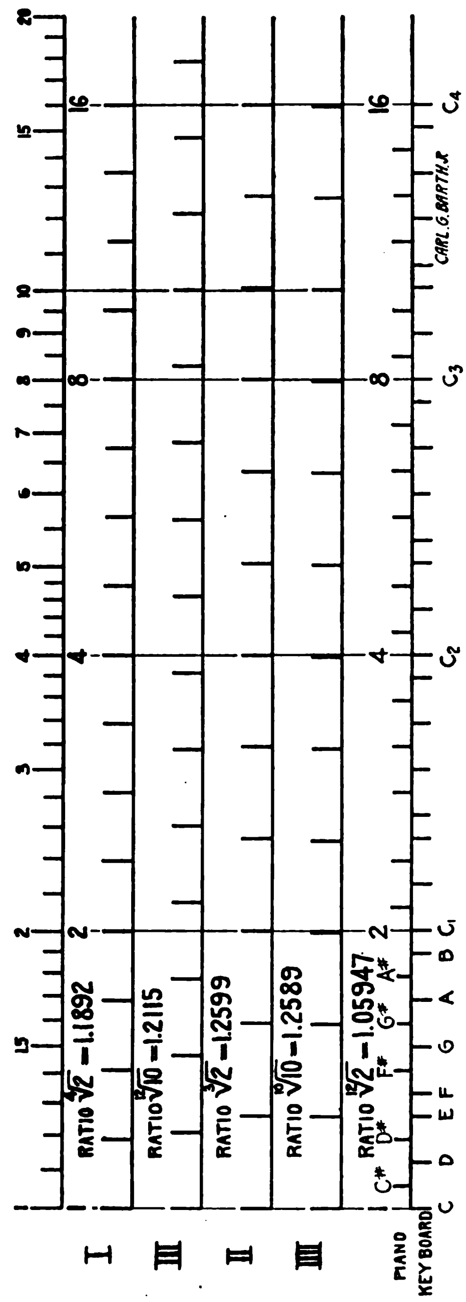


FIG. 2 LOGARITHMIC PLOT OF SUGGESTED ALTERNATIVE STANDARDS FOR SPEEDS AND FEEDS

FURTHER REASONS FOR ADVOCATING THE ADOPTION OF PROGRESSION [I]

27 In Par. 1134 of Mr. Taylor's On the Art of Cutting Metals,[1] the author gives his original classification of hardness of metals in terms of their relative cutting speeds, all other conditions being maintained uniform, thus: If 0 stands for the ideally softest grade of any kind of metal, he made class 1 represent a metal just enough harder to reduce the cutting speed by 1.1; class 2 by 1.1^2; class 3 by 1.1^3, etc., or by factors in the following geometrical progression:

Class	0	1	2	3	4	5	6	7	8	9	10, etc.
	1	1.1	1.21	1.331	1.464	1.6105	1.776	1.949	2.144	2.358	2.594

28 The writer has since modified this to

Class	0	1	2	3	4	5	6	7	8	9	10, etc.
	1	$2^{\frac{1}{8}}$	$2^{\frac{1}{4}}$	$2^{\frac{3}{8}}$	$2^{\frac{1}{2}}$	$2^{\frac{5}{8}}$	$2^{\frac{3}{4}}$	$2^{\frac{7}{8}}$	2	$2\cdot 2^{\frac{1}{8}}$	$2\cdot 2^{\frac{1}{4}}$, etc., or . . . [VI]
	1	1.0905	1.1892	1.2968	1.4142	1.5422	1.6818	1.8340	2	2.1810	2.3784, etc.

as being more rational, because then we have the great simplicity that a difference of eight hardness classes corresponds to a difference of exactly 2 to 1 in cutting speeds. It will be seen that this progression is just the same as progression [I] with intermediate terms interpolated, and hence the simplicity of this for a speed progression in conjunction with [VI] for a hardness scale.

29 It would mean that a tool would last just the same length of time on two materials two hardness classes apart, if run on the softer grade with a certain speed and upon the harder grade with the next slower, as perhaps best brought out by the logarithmic plot in Fig. 3.

30 Again, in Par. 732 of On The Art of Cutting Metals, it is stated that, *approximately*, the cutting speed varies inversely as the *square root* of the feed, which is a compromise between the special laws for steel and cast iron. Therefore, if in connection with a series of speeds conforming to progression [I] we also have a series of feeds in the same progression, the relation of these feeds and speeds to each other will, in order to maintain all other conditions uniform, be as represented by the logarithmic plot in Fig. 4.

31 Thus, for example, if 32 r.p.m. and 0.1 in. feed on a material of a certain hardness grade allow a tool to last a satisfactory length of time, this time would be practically maintained undisturbed if we increased the feed to 0.141 in. and dropped the speed simultaneously

[1] Trans.Am.Soc.M.E., vol. 28, p. 31.

to 26.9 r.p.m., the depth of cut remaining the same. Or, in general, in dropping or increasing the speed by a step at a time, we would increase or drop the feed by two steps at a time.

32 By combining the plots in Figs. 3 and 4, as done in the slide rule Fig. 5, we also most readily recognize that a change of one or more numbers in the hardness class of the metal cut will have no effect on the cutting speed if we meet this with a change of feed involving a corresponding number of steps, increasing the speed as the hardness goes down, decreasing it as the hardness goes up.

33 Finally, in Par. 700 of On the Art of Cutting Metals is given the relation between cutting speed and the life of a tool in cutting steel, which is, that the speed varies inversely as the eighth root of the time the tool will last, all other conditions remaining uniform.

34 Adding this relation to the slide rule in Fig. 5, we get the rule shown in Fig. 6.

35 Thus the further simplicity that for every change of one hardness class not accompanied by a change in feed or speed, or for a change of feed from one step to the next, the life of the tool will be affected 2 to 1, in the one direction or the other, as the case may be.

36 In other words, with speeds and feeds arranged according to progression [I], the application of the principal laws set forth in On The Art of Cutting Metals becomes so easy that, were it not for the question of power at times involved, the complete slide rules referred to in paragraphs 1188 to 1197, and first brought to the attention of this Society in the writer's own paper on Slide Rules in the Machine Shop as part of the Taylor System of Shop Management,[1] read in December, 1903, would lose a great deal of their importance. At any rate their construction would be greatly facilitated, and their applicability become more general.

37 The writer's preference for progression [I] and his hope for its eventual universal adoption will now be fully understood by those who have already given some attention to the scientific control of machine utilization, and also by others who with an unbiased mind have tried to follow the arguments here advanced.

38 A further argument is that this progression represents a fair average of the speed series provided by certain well-known machine-tool builders in some of the machines produced by them in recent years, after they had realized that the time had come for the provision of closer speed regulation, even if this meant a considerable reduction of the ratio of the fastest to the slowest speeds obtainable.

[1] Trans.Am.Soc.M.E., vol. 25, p. 49.

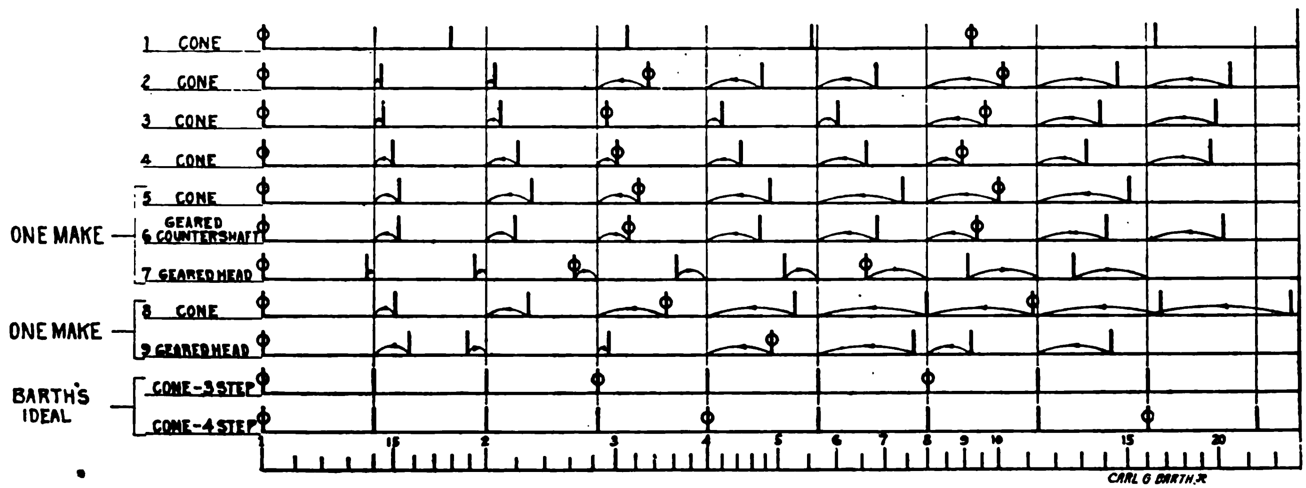


FIG. 7 LOGARITHMIC PLOT OF SINGLE COUNTERSHAFT SPEED SERIES OF NINE 16-IN. LATHES SUBMITTED BY VARIOUS MANUFACTURERS IN FEBRUARY, 1914

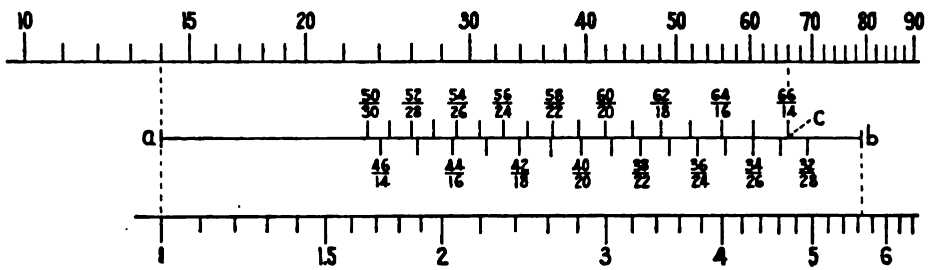


FIG. 8 SIMPLE LOGARITHMIC-SCALE METHOD FOR GETTING THE CLOSEST POSSIBLE APPROXIMATION TO A GIVEN RATIO WHEN TWO SETS OF GEARS ARE INVOLVED

39 An example of this fact is represented in Fig. 7, which shows the speed ratios of nine 16-in. lathes offered by six different manufacturers in answer to an inquiry in February, 1914. One manufacturer offered three different designs and a second two different designs. It will be noted that the difference between the speed series provided in the different designs of the same maker, differ as much among themselves as do the designs of different makers, except in the case of lathe No. 1, which represents the least up-to-date design.

40 To make the diagram clearer, the speed ratios are laid out for a single forward countershaft speed only, thus showing only every other ratio of the full progression intended.

41 The objection previously considered, that the adoption of standards may to some extent block the way for improvements, certainly does not hold good to the same extent, if at all, when it comes to such a purely mathematical matter as a speed series; but it will, on the other hand, relieve designers from wasting time in worrying over what sort of speed series to provide every time the question comes up, which has been the case in the past, and still is in a very large measure.

42 But, even so, numerous designers for the reasons given earlier in this discussion, will have difficulties enough in making their inventions conform to any standard speed series, and, in addition thereto, in selecting gears of such numbers of teeth that the closest possible approximation to the various theoretical transmission ratios may be obtained.

43 Most designers of the present day, whether mathematically inclined or not, are familiar with the use of the simple slide rule in selecting practical numbers of the teeth for a gear and pinion of a single set of gears to give a certain ratio as closely as possible; but a simple logarithmic-scale method for getting the closest possible approximation to a given ratio when two sets of gears are involved, which the writer has used for some years past, is not generally known, if indeed at all known to anybody who has not worked with the writer in connection with these matters.

44 This method will be fully disclosed by a single illustration embodied in the diagram Fig. 8, and we will take for our example the ratio $4\sqrt{2} = 5.6568$, with the restriction that neither pinion is to have less than 14 teeth, and that the sum of the numbers of the teeth in the one set of gears is to be 80, and in the other 60.

45 By means of a logarithmic scale, which is shown just below the significant part of the diagram, lay off the ratio 5.6568 from *a* to

b. Now place the graduation point for 14 on the logarithmic scale, representing a pinion of 14 teeth, at *a*, and make a mark at the graduation point representing the gear that would be a mate to this pinion, namely one having 80 − 14 = 66 teeth. This operation is indicated by the logarithmic scale above the significant part of the diagram. The distance *ac* then represents the ratio 66/14. Similarly, 15 is placed at *a* and a mark made at 65, 16 is placed at *a* and a mark made

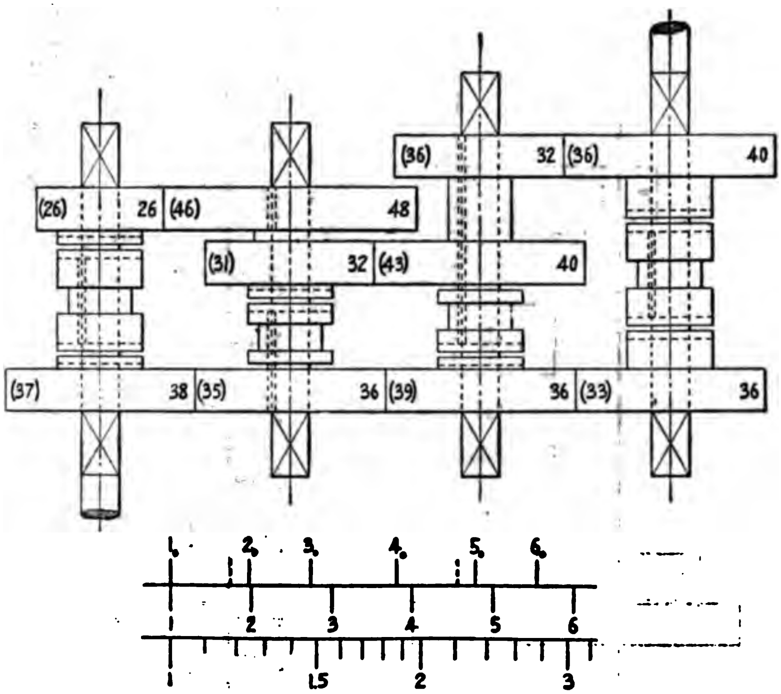


FIG. 9 DIAGRAM OF GEARING OF SPEED VARIATOR OF A WELL-KNOWN MAKE OF PLANER

Figures in Parentheses Show Number of Teeth in Gears as Made; Other Figures Show Number of Teeth in Gears to Improve the Speed Series; 1_0 to 6_0, Logarithmic Plot of Original Relative Speeds; 1 to 6, Logarithmic Plot of Improved Relative Speeds

at 64, and so on, as far as will eventually appear necessary. Next, below the line place the graduation point 14 at *b*, and mark off towards *a* the point for the mate for this pinion in the second set of gears, namely, 60 − 14 = 46. Repeat for 15 and 45, 16 and 44, etc.

46 On completion of this repeated operation we at once discover that the closest coincidence between the marks on both sides of the line *ab* is for the two sets of gears 65/15 and 34/26, whose com-

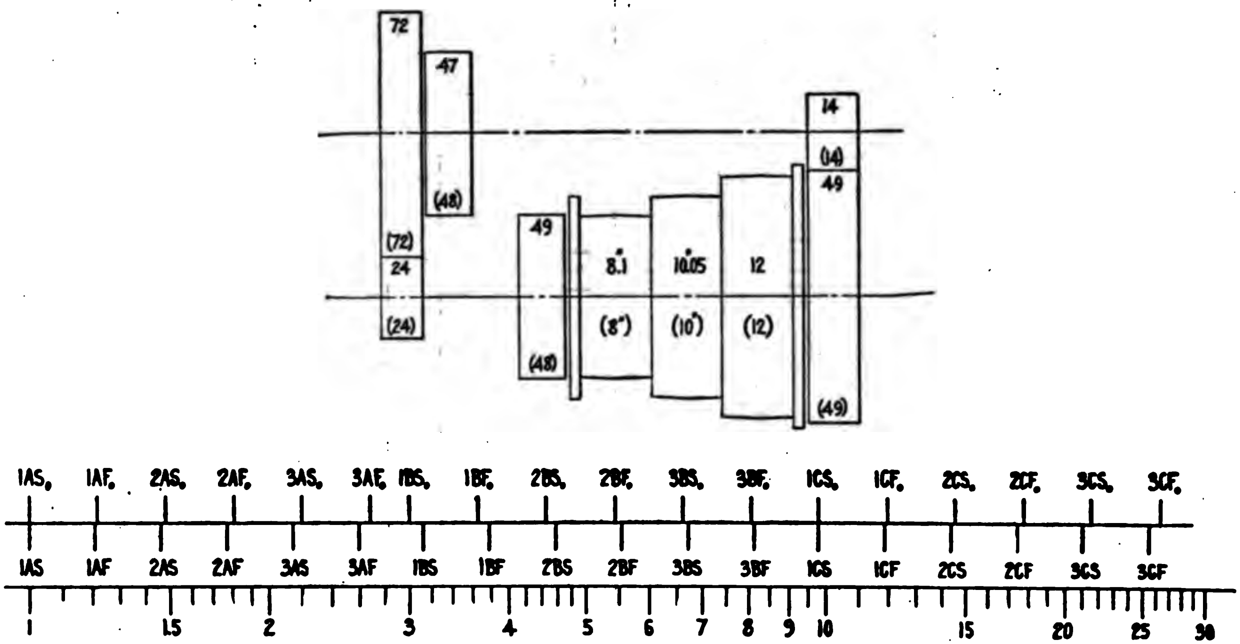


FIG. 10 DIAGRAM OF LIVE HEAD OF A WELL-KNOWN MAKE OF LATHE

Figures in Parentheses Show Respectively the Number of Teeth in the Gears and the Diameters of the Cone Pulleys as Made; the Other Figures Indicate a Simple Change that would Further Improve a Very Good Speed Series; $1AS_0$ to $3CF_0$, Logarithmic Plot of Original Speeds; 1AS to 3CF, Logarithmic Plot of Improved Series bined ratio is 5.6667, which is less than 0.2 of one per cent greater than 5.6568.

47 It will also be seen that the two combinations $\frac{68}{31} \times \frac{49}{19}$ and $\frac{68}{29} \times \frac{41}{17}$ come quite close to the required ratio.

48 Of course, special requirements may often complicate a problem of this kind in various ways, but the method described can always be used as a help in its solution. However no further consideration will be given the matter here.

49 As an answer to numerous designers who look upon the striving for such close approximations to theoretical ideals as pure non-

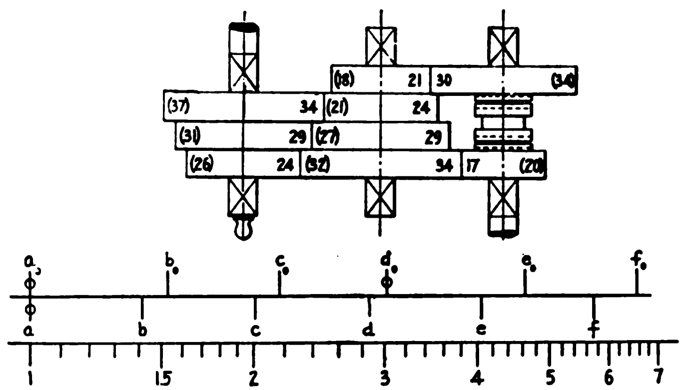


FIG. 11 DIAGRAM OF GEARS IN FEED BOX OF A WELL-KNOWN MAKE OF DRILL PRESS

Figures in Parentheses Show Number of Teeth as Made; Other Figures Show Improved Number of Teeth; a_0 to f_0, Logarithmic Plot of Original Relative Feeds; a to f, Logarithmic Plot of Improved Relative Feeds; These Conform Quite Closely to Progression [V], p. 901

sense, the writer will say that a little more intelligent persistence in these matters soon educates a man to a point where it takes him no longer to determine close approximations than rough ones, with a genuine additional pleasure added to his work.

50 Among additional helps in arranging speed and feed series as advocated, any otherwise desirable group of gears of the following numbers of teeth will be found to lend themselves admirably for use with a tumbler gear:

88	74	62	52	44	37	31	26	22
$\frac{88}{88}$	$\frac{88}{74}$	$\frac{88}{62}$	$\frac{88}{52}$	$\frac{88}{44}$	$\frac{88}{37}$	$\frac{88}{31}$	$\frac{88}{26}$	$\frac{88}{22}$
1.1	1.1892	1.4194	1.6923	2	2.3784	2.8387	3.3846	4

51 Comparing these ratios with the writers' ideal progression [I],

it will be seen that the deviation is nowhere greater than 0.7 of one per cent.

52 Cone gears made with teeth of any multiple of the following, also give ratios differing less than 1.02 per cent from conformity with the same progression:

$\frac{16}{8}$	$\frac{15}{9}$	$\frac{14}{10}$	$\frac{13}{11}$	$\frac{12}{12}$	$\frac{11}{13}$	$\frac{10}{14}$	$\frac{9}{15}$	$\frac{8}{16}$
2	1.6667	1.4000	1.1819	1	$\frac{1}{1.1819}$	$\frac{1}{1.4000}$	$\frac{1}{1.6667}$	$\frac{1}{2}$

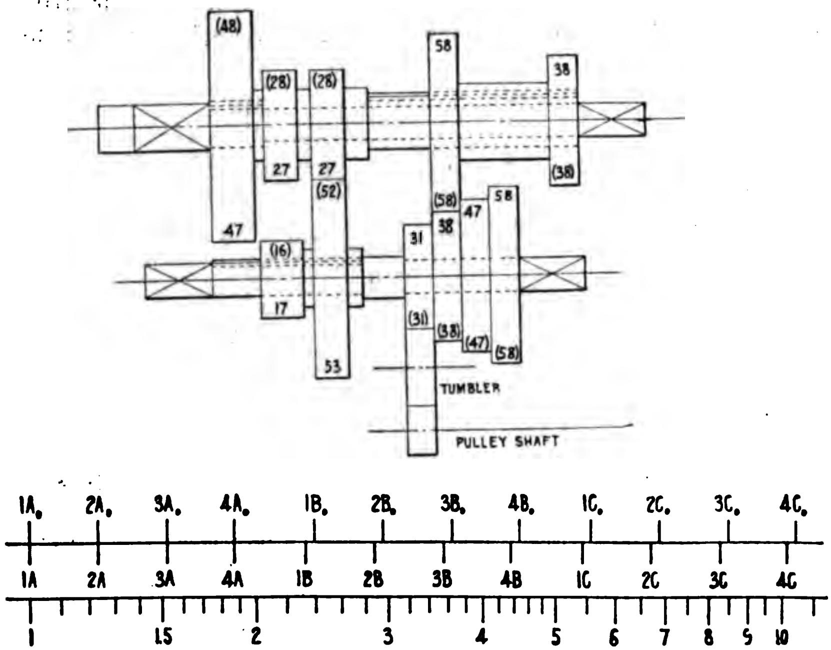


FIG. 12 DIAGRAM OF DRIVING GEAR OF A WELL-KNOWN MAKE OF MILLING MACHINE

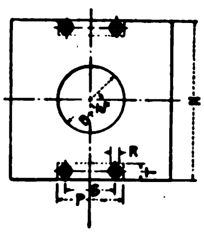
Figures in parentheses Show Number of Teeth in Gears as Made; Other Figures Indicate Number of Teeth in Gears to Further Improve an Already Excellent Speed Series; $1A_0$ to $4C_0$, Logarithmic Plot of Original Speed Series; 1A to 4C, Logarithmic Plot of Improved Speed Series

53 To further emphasize how even some fair attempts at regular progressions of speeds and feeds have fallen short of what might have been attained, Figs. 9, 10 and 11 are submitted as a few of numerous cases that have been investigated by the writer; but specific cases showing absolute ignorance on the part of designers of these matters will not be referred to at this time.

54 Fig. 12 also shows how even a quite excellent speed series may be further improved.

POWER OF MACHINE TOOLS

55 But while the adoption of a standard speed series for all machine tools would be a wonderful step in the right direction, to get the greatest advantage from this it would have to be accompanied by the adoption of a standardized amount of power for each size machine of a



WIDTH OF TOOL = W			
POST		PLATE	
A	W + 1/16	N	4 1/2 W + 1
B	2 1/8 W + 3/16	P	B
C	2 7/16 W + 5/16	R	5/16 W + 1/8
D	W	S	1 5/16 W − 1/16
E	5/16 W	T	5/8 W + 1/4
F	1 3/8 W + 1/8	SHIMS	
G	1/16 W + 1/16	W	W
H	1/2 W	V	5 1/2 W + 2
J	3 5/16 W + 1/8	T	[illegible]
X	3 5/16 W + 1/8	SET SCREW	
WASHER		Y	3 W + 5/16
K	3 1/4 W + 1/16	Z	3 1/16 W + 5/16
L	3 3/8 W + 1/16		
M	W		

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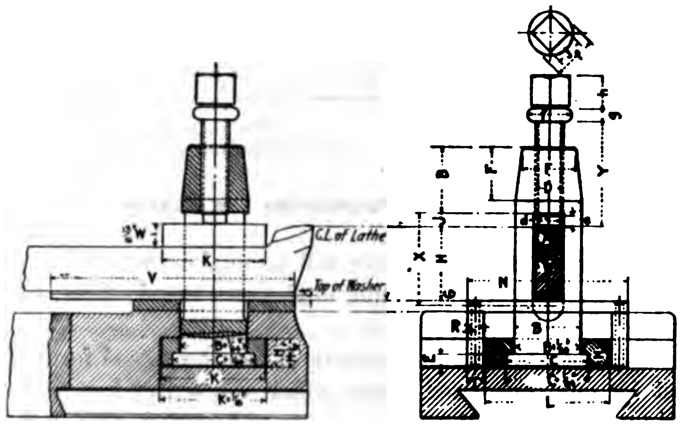


FIG. 13 PROPORTIONS USED BY AUTHOR FOR THE SWIVEL FORM OF TOOL POST FOR MODERATE-SIZED LATHES

certain type. However, the adoption of the latter without a standard speed series would be of but little, if any, advantage.

56 The author has a great deal more to say in advocacy of standards for speed and feed series and standardized power for machine tools, but believes that this will be brought out in a more profitable manner by the discussions that he hopes will be provoked by what has already been said.

TOOL POSTS FOR LATHES

57 In On The Art of Cutting Metals, which Mr. Taylor wrote under great pressure, and in which accordingly the proper consideration was not given to certain subjects, the mistake was made of advocating cutting tools with moderate clearance angles, without at the same time calling attention to the fact that these can be used only in connection with tool posts in which the body of the tool is raised parallel to itself as the tool is ground down to a smaller height. Several (so far as the writer knows, only unsuccessful) attempts have been made to construct tool posts for elevating a tool parallel to itself without the use of shims. At the present time it seems impossible, therefore, to recommend any but the time-honored forms of tool

D	A	B	C	E
BOLT DIAM.	$1\frac{1}{2}D$	$\frac{5}{8}D+\frac{7}{64}$	$1\frac{3}{4}D+\frac{5}{32}$	$\frac{1}{32}D+\frac{1}{64}$
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{11}{32}$	$\frac{13}{16}$	$\frac{1}{32}$
$\frac{7}{16}$	$\frac{21}{32}$	$\frac{25}{64}$	$\frac{59}{64}$	$\frac{1}{32}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{27}{64}$	$1\frac{1}{32}$	$\frac{1}{32}$
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{32}$
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{37}{64}$	$1\frac{15}{32}$	$\frac{1}{32}$
$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{21}{32}$	$1\frac{11}{16}$	$\frac{3}{64}$
1	$1\frac{1}{2}$	$\frac{47}{64}$	$1\frac{29}{32}$	$\frac{3}{64}$
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{13}{16}$	$2\frac{1}{8}$	$\frac{3}{64}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{57}{64}$	$2\frac{11}{32}$	$\frac{3}{64}$
$1\frac{3}{8}$	$2\frac{1}{16}$	$\frac{31}{32}$	$2\frac{9}{16}$	$\frac{3}{64}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{3}{64}$	$2\frac{25}{32}$	$\frac{1}{16}$

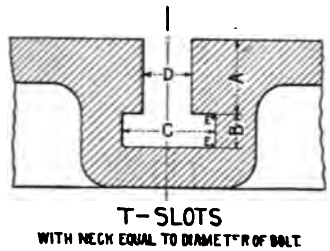


FIG. 14 AUTHOR'S PROPORTIONS FOR T-SLOTS

posts, so arranged that an operator will be unable to modify materially the effectiveness of the cutting angles of a standardized tool by careless insertion and adjustment.

58 In Fig. 13 are shown proportions for the swivel form of the tool post for moderate-sized lathes which the writer has used as a standard for several years past.[1] The most essential part of this, as regards size, is the width of the slot, which is for a tool body of approximately 1/30 of the swing of the lathe over the bed, and a distance from the center of the lathe to the tool-supporting plate that corresponds to the ground height of the roughing tools shown in

[1] It has only recently been learned that the form of the bottom swivel washer of this style of tool post was originally designed and patented by the late Charles A. Bauer. However, the writer believes that the application of the legs under the stationary washer on top of the slide is original with him. They prevent the bending of the washer, which often has to be made quite thin to provide height enough for a new tool when the tool is placed in a position parallel to the slot in the slide.

On The Art of Cutting Metals. The propriety of these propositions is decidedly open to discussion, however, particularly the latter.

LATHE CENTERS

59 The consideration of lathe centers will be taken up later on, in connection with drill-press and milling-machine spindles and sockets, etc.

T-SLOTS

60 An exceedingly important part of the machine equipment of a shop are the T-slots in various machines, and the writer has also spent a great deal of money in bringing these to a standard, a matter that has at times meant entirely new face plates or tables for some machines.

61 A great many years ago experiments were made by William Sellers & Company to ascertain the strength of T-slots relatively to a T-headed bolt, and on the strength of these, that company adopted a standardized set of slots, which, as later modified by the writer, have also been used by him for several years past. They are shown in Fig. 14.

LATHE CENTERS AND DRILL-PRESS AND MILLING-MACHINE SOCKETS

62 The writer has also had to spend a great deal of money in standardizing lathe centers and drill-press and milling-machine sockets, and ventures to suggest that the time has also come for the machine-tool builders to help this matter along.

63 It is well known that the Morse standard sockets are no standards at all, but a perpetuated, laudable, but unsuccessful attempt of years ago to establish standards. However, compelled for the time being to accept them as they are, everybody has now at least two standard tapers for sockets and shanks to contend with, namely the Morse, and the Brown and Sharpe. The best the writer has been able to do, therefore, has been to make all lathe centers conform to a Morse standard, so as to enable drills or drill sockets to be directly inserted in either spindle of certain or all lathes in a shop; and to make all milling-machine sockets conform to the Brown and Sharpe standard, with Morse drill sockets having Brown and Sharpe shanks for use with these whenever holes have to be drilled in a milling machine.

64 The writer unqualifiedly recommends the universal adoption of the Brown and Sharpe standards all around, and the use of Morse sockets with Brown and Sharpe shanks during the change. He also

recommends the universal abandonment of the tang as a means of driving. We have for years had the ridiculous inconsistency of drill makers, that they still furnish taper-shank drills with the old-style tang as a means of driving, and along with this extensively advertise and sell various forms of " use-them-up " sockets for drills with the original tang broken off. It puts me in mind of "Peer Gynt" in Ibsen's drama of the same name, when he manufactured and exported idols to China, and, to ease his conscience, also sent missionaries over there to convert the Chinese to Christianity.

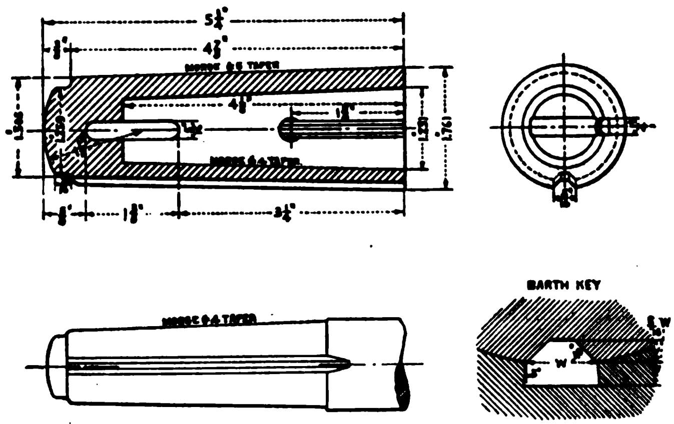


FIG. 15 AUTHOR'S KEY METHOD OF DRIVING DRILL, ARBOR AND BORING-BAR SHANKS

65 More than twelve years ago the writer adopted a modification of William Sellers & Co.'s key method of driving as a substitute for the tangs of drills, arbor and boring-bar shanks. This is illustrated in Fig. 15, and differs from the Sellers method in the use of a special form of key that has become somewhat known as the Barth key. The virtue of this as a means of driving consists in its being subjected to crushing pressures only, and in its having no tendency to work out of its seat; and for a taper drill socket it has the additional great advantage over the Sellers straight key that the drill shank can be inserted rapidly without any special care.

DISCUSSION

ADOLPH L. DE LEEUW said that the author had plainly shown how a good geometrical progression could be worked out, and had given the impression that it was an absolute necessity in all machine tools. Large boring mills, however, might be better without the geometrical progression, as, when a flywheel was turned, there was need for cutting at the rim and at the hub, but nowhere else, 90 per cent of the cutting being done at the rim and 10 per cent at the hub. Hence speeds should be bunched together to be correct for working at the rim and the hub. As a rule, that would lead to complications, and the maker would prefer to make the speeds in an even flow. He thought that the term geometrical progression as applied to speed changes in machine tools had been used by manufacturers for many years as a selling phrase without much knowledge of the correct meaning or full significance of it.

The time had come for a closer study of the proper geometric speed series. The attempt made by the author to point out a geometrical progression, which would be universally applicable deserved the highest praise. Personally, he believed the progression as laid out by the author was about as good as could be had under the conditions and with the knowledge we have at the present time.

He regretted that there had been no investigations made on the art of cutting metals except for lathe work and for roughing cuts. Also, whereas everything possible had been done to study the matter of changing speeds on lathes, practically nothing had been done to provide for the necessary speeds and feeds on planers, very little as to milling machines and drills, and almost nothing as to screw machines. Screw machines were at present a large proportion of the total shop equipment of the country. The conditions under which it paid or did not pay to have a tool wear out varied. Whereas it was probably very reasonable to change a certain tool every twenty minutes in some machines, it would be almost criminally foolish to change it so frequently in an automatic screw machine. The experiments on cutting angles had been carried on only through a small range. He said that he had personally made some little experiments with tools at an angle of not more than 25 deg. and found some very surprising results. He believed that if the Society as a whole were to stand behind such efforts as the author had spoken of, to standardize the range of speeds and feeds, much could be accomplished.

Harry V. Haight said that he agreed with the author on the subject of belt feed. It had been his experience in the building of some two hundred lathes for munitions work that a 3-in. belt was sufficient to supply any demand for power, and that any series of geometrical progression of feeds could be obtained by changing pulleys. In contrast to this experience was that with expensive drill presses which were idle so much of the time because of broken gears. He thought that if drill presses were built to work rather than to sell they could be made to stand up night and day.

H. M. Norris (written). That some machine-tool builders are learning to appreciate the value of a smaller speed variation is attested by the fact that fifteen years ago no radial drill was provided with more than eight changes, while now they may be obtained

TABLE 1 COMPARISON OF BARTH, NORRIS, AND USUAL METHODS OF CALCULATING DRILL RATIOS

Barth			Usual Method			Norris		
Ratio	R.P.M.	Diam., In.	Ratio	R.P.M.	Diam., In.	Diam., In.	R.P.M.	Ratio
1.000	43.3	7.09	1.000	67.9	4.50	4½	67.9	1.000
1.189	51.5	5.93	1.152	78.2	3.91	4	76.4	1.125
1.414	61.2	4.99	1.326	90.1	3.39	3½	87.4	1.285
1.682	72.8	4.20	1.527	103.7	2.94	3	101.9	1.500
2.000	86.6	3.53	1.759	119.4	2.56	2½	122.2	1.800
2.378	102.8	2.97	2.025	137.5	2.22	2¼	135.8	2.000
2.828	122.3	2.49	2.332	158.3	1.93	2	152.8	2.250
3,364	145.6	2.10	2.686	182.2	1.68	1¾	174.5	2.570
4.000	173,2	1.77	3.093	210.0	1.46	1½	203.8	3.000
4.757	205.8	1.48	3.552	241.0	1.27	1¼	244.4	3.600
5.657	244.8	1.25	4.103	278.6	1.098	1⅛	271.6	4.000
6.727	291.0	1.05	4.725	321.0	0.952	1	305.6	4.500
8.000	346.0	0.883	5.441	369.0	0.838	⅞	349.2	5.140
9.514	411.6	0.742	6.266	425.0	0.720	¾	407.5	6.000
11.314	488.9	0.625	7.215	488.9	0.625	⅝	488.9	7.200

with 20, 24, and even 30. The speeds of móst of these later tools are intended to be in geometrical progression, but I am of the opinion that a geometrical series is not the most utilitarian for a drilling machine.

The ratio of progression most favored by the author appears in the first column of Table 1. The second column gives the corresponding number of revolutions per minute, and the third the diameters of drills which this series would drive at a cutting speed of 80 feet per minute. To my mind this is not as efficacious a series as that obtained from a ratio giving both desired extremes, columns

4, 5 and 6. Here we have fifteen speeds for drills from $\frac{5}{8}$ to $4\frac{1}{2}$ inches in diameter, while under the former gradation there are but twelve.

But why use either? The operator is not interested in the size drill that should be used with *each speed.* What he wants to know is how to place his levers to obtain the correct speed for *each drill.* If diameters are fitted to speeds, they will have to appear on the speed plate as decimals or be designated by such fractions as $\frac{2}{3}\frac{3}{4}$, $\frac{4}{6}\frac{3}{4}$, $\frac{8}{8}\frac{1}{4}$, etc. Is it not better, therefore, to decide first upon the drill diameters and then try to obtain the exact speed for each, regardless of the ratio of advance? Suppose, for example, we set down, as in column 7, the diameters of drills we would like a machine to drive at a cutting speed of 80 feet per minute. It is an easy matter to ascertain at what number of revolutions per minute each should run, column 8. Here each *harmonic* group of speeds may be obtained from a five-change speed box or a 2-to-1 motor, while back gears made in the ratio of 1 to 1, 1 to 2, 1 to 4, 1 to 8, etc., will give as many successive *geometric* groups as desired.

In Par. 11 it is stated that "It is by this time universally accepted by those who have a right to an opinion in this matter that the available speeds of a machine should be in a geometrical progression," and that "a discussion of this will not be undertaken by the writer unless provoked by some one else." It is not my purpose to provoke an argument, but I would like to learn if I am in error in thinking that my series is the most practical.

FRED A. PARSONS (written). Regarding standardized progressions for speeds and feeds and standardized power, the paper presents what is without doubt the ideal condition, but in application the following points should be borne in mind:

1 At present a very small percentage of machine tools goes to factories where the management would appreciate the refinements in feed and speed ratios suggested. Indeed, the jobbing shop could never get together with the manufacturing plant on the question, since the former requires a wide range of speeds with large ratios and the latter a small range of speeds and feeds with small ratios.

2 Present machine-tool feeds and speeds are very largely the outgrowth of the above conditions, though there are no doubt many instances, as the author mentions, where no particular attention has been paid to anything except getting the high and low speeds required.

3 Present milling practice seems to require of general-purpose

machines, such as the plain knee-and-column type, about the same number of feeds as of speeds, but a total ratio of about 24 to 1 for the speeds as against about 48 to 1 for the feeds. Unless present practice is wrong, this would require two different progression ratios.

4 Some designs permit considerable economies to be effected in space and parts required by attempting only a fairly close approximation of the perfect geometrical progression. An instance may be considered of obtaining nine speed variations with nine gears on three shafts, the gears on first and last shafts being sliding gears and on the intermediate being laterally stationary. In this case it is not possible to reach any perfect geometrical progression exactly if the high and low speeds are even fairly far apart, but a close approximation can be obtained.

5 Considering that by practically any method of drive the variation from normal speed at no load and at full load will be up to 5 per cent, is it warrantable to add cost to a machine tool to come closer than a few per cent above or below a given geometrical progression?

6 Mr. Barth has stated in his paper that the standardization of speed ratios should be accompanied by a standardization of power for machines of a given type and size. Such a standardization must be considered from the standpoint of the user. At present the users seem to demand two power capacities of any given range, one light and one heavy, and it is hard to see how this could be avoided, as otherwise some will be purchasing power capacity they do not need, though they require the range, and in the other extreme the reverse will be true.

7 It seems unquestionable that a few years more will see the development and general application of a machine-tool drive for both feed and speed in which the speed variations can be represented by a smooth curve from minimum to maximum. Such a drive is typified at the present time by the variable-speed motor; the objections are that it is expensive to install and cumbersome. Such a drive will not of course apply to lathe feeds for screw cutting, or milling-machine feeds for spirals.

Regarding the standardization of details such as T-slots, taper shanks, etc., which the author mentions, there are not the difficulties which enter into standardization of speeds and power. The cases are quite different, inasmuch as if a T-bolt, for instance, $\frac{3}{4}$ in. in diameter, is being used in an appropriate place, there must certainly be only one correct depth of slot, considering strength for any given material, the same as there is one best size for the square and for the thickness of the bolt head.

The Author. While I am rather disappointed at the fewness of the discussions of my paper, the, on the whole, very favorable discussion by an engineer who has given so much independent attention to the subject as Mr. De Leeuw, is compensation enough for my having taken the trouble to prepare it. I feel that his discussion is a full endorsement of my main contention, namely; that this Society take some action in this matter. Regarding Mr. De Leeuw's comment that certain machines, as, for instance, large boring mills, would be better provided with two *bunches* of speeds, I agree with him fully; but will insist that each bunch should conform to the principles advocated by me. I also agree with him that more experiments with cutting tools should be made, and am pleased to inform him that more is known about the art of cutting metals with planer tools, milling cutters and drills than has as yet been published. For fully ten years I have thus had a set of slide rules covering certain milling cutters on the one hand, and gear cutters on the other hand, which have been of substantial assistance to me in my efforts to improve shop practices; and if I live long enough, I shall fully disclose their theory, construction, and application, together with much else relating to the art of cutting metals. Regarding Mr. Norris' discussion, it discloses such a fundamental lack of grasp of the whole matter considered, that I would be inclined not to waste any words upon it, beyond stating that it proves that he belongs to the other class I had in mind when I referred to "those who have a right to an opinion in this matter." However, being a great admirer of the many excellent features of the line of radial drill presses he has developed during a term of years, I am glad to point out that his mistake consists in neglecting to take account of the fact that the proper cutting speed of a drill — just as much as of a lathe tool and a milling cutter, etc. — varies with the diameter of the drill, the feed used, and the hardness of the material drilled; and that there is an exceedingly small increment between all the drills used in nearly every shop. Hence, theoretically, to meet every combination of these variables between a maximum and a minimum, infinitesimal increments would be required, in the absence of the possibility of obtaining which, *equal percentage* increments become our only rational practical approximation; and that is, in a nutshell, all the reason and argument there is behind a geometrical progression of speeds and feeds, and it is all that is needed. Regarding Mr. Parsons' discussion, I fully agree with him that at present but a small percentage of machine tools goes to the factories where the management would appreciate the refinements of the feed and speed ratios suggested by me. However, it has been my business for a number of years to help managements to such appreciation, and I think the machine-tool builders should again help *me* do it. The field I can personally cover is exceedingly limited, whereas theirs is unlimited. What he says about the relation of feeds and speeds of general-purpose milling machines is not clear to me, so I cannot comment upon it. I never expected all attempts at a geometrical progression of speeds to turn out even a practical perfection, but I have fully demonstrated in my paper that there is no inherent difficulty in doing better in numerous instances than what has been done. I have said nothing in my paper that excludes any number of sub-standards of machine tools, as regards power. Thus, I fully believe that there might to advantage be both a heavy and a light standard of any work capacity (as opposed to chip-producing capacity) of several of the more common types of machine tools. For reasons that I prefer not to reduce to writing at the present time, while much misunderstanding still exists regarding my work as a scientific-management expert, I do not favor any kind of continuous speed or feed variator for the mere sake of getting the closest possible speed adjustment (which nobody can hope to know closer than a rather uncertain percentage anyway, for any set of practical conditions), though under certain conditions I strongly advocate adjustable-speed motor drives, because of the quickness with which they enable speed changes to be effected.

In the presentation and discussion of this paper at the meeting, I brought out a number of additional points which I have not found it possible to prepare for incorporation in this publication.

No. 1560

A GAS PRODUCER FOR BITUMINOUS FUEL

By O. C. Berry,[1] Lafayette, Ind.
Non-Member

When bituminous coal is used in a standard up-draft producer of the type in which anthracite is gasified, the volatile hydrocarbons are immediately driven off by the heat of the fuel bed. These hydrocarbon vapors mix with the hot producer gas leaving the fuel column, and, upon cooling, condense and form a fog made up of finely divided particles of tar. This tar will deposit on anything with which the gas comes in contact, and will clog up the pipe lines, stick the valves, and prove a general nuisance. Therefore, the special problem that the designer of a bituminous gas producer must solve is to eliminate this tar from the gas, while still meeting all of the requirements that the standard anthracite plant must meet.

2 A solution is possible along two general lines. The first and most obvious is to develop a mechanical tar extractor that will eliminate the last trace of tar from the gas without affecting its quality in any other way. The plant manufactured by R. D. Wood & Co., and used by the United States Geological Survey in its experimental plant at St. Louis, is probably one of the best examples of this type of a bituminous gas producer. The second type of producer eliminates the tar vapors in the fuel column of the producer itself. This may be accomplished by applying one or both of the following well-known principles:

a If tar vapors are caused to pass through a bed of incandescent fuel, they will be "cracked," or split up into permanent gases, such as CH_4, and free H_2, and lampblack, or finely divided particles of pure carbon. In this way the tars may be destroyed completely, and a considerable amount of valuable fuel gas will result.

[1] Purdue University.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

b Tar is a mixture of a large number of complex hydrocarbons, and will therefore burn to CO_2, and H_2O. These gases may be passed through the bed of incandescent fuel and split up into CO and free H_2, exactly the same gases that are obtained in the gasification of anthracite coal.

3 The multiple-combustion-chamber producer, the down-draft producer with a combustion zone at the top and one at the bottom, and the recirculating producer are all examples of this second class. It is the recirculating producer that is to be discussed in this paper.

4 The volatile matter may all be driven off from a bituminous coal at a temperature considerably below the highest attained in a gas producer. In the recirculating producer advantage is taken of this fact, and instead of drawing the finished gas from the top of the fuel column, it is taken out at a point well below the top, where the coal is sufficiently hot so that one may be certain that the last trace of tar has been removed. The tar-laden gases from the top of the producer are drawn off and re-entered into the fuel column near the bottom. In the Whitfield producer they are introduced directly into the "incandescent" zone, while in the Daniels producer and several others they are passed into the combustion zone of the fuel column.

5 Fig. 1 is taken from the patent drawings of Charles Whitfield, and illustrates the general idea. The coal is trapped into the producer at the top, and the only air inlet is below the grates at the bottom. The producer is kept filled almost to the top with fuel, so that the outlet for the finished gas is well down toward the center of the fuel column. All of the combustion in the producer takes place just above the grates, forming a layer of ash immediately above the grates, a combustion zone just above the ash, an incandescent zone just above the combustion zone, and distillation zones of decreasing temperature continuing toward the top of the fuel column. The recirculating pipe *A*, in which flow is induced by the steam nozzle *B*, takes the products of distillation from the top of the producer down to and into the incandescent zone. The annular chamber *C* extends entirely around the producer, and makes it possible to distribute the tarry vapors to all sides of the incandescent zone.

6 In developing a producer of this type information was needed that could not be found in print and that therefore required special investigation, such as:

a The most efficient and best type of blower to use.

b The temperature at which the pipe, the blower and the gases themselves must be kept to recirculate the gases with the least deposit of their tar content and the least trouble from other sources.

c The temperature at which the top of the fuel column must be held to prevent troublesome tar deposit there.

d The lowest temperature at which one may be certain that the last trace of tar has been removed from the coal used.

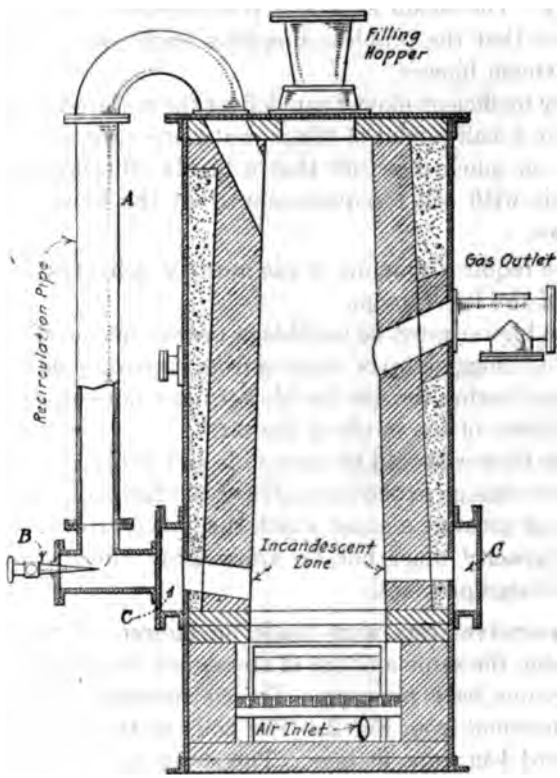


FIG. 1 WHITFIELD RECIRCULATING PRODUCER

The author has done investigative work along each of these lines. This work was started at the University of Wisconsin, but most of it has been done in the laboratories of Purdue University. The results obtained will be presented here, together with an enumeration of several requirements that a successful gas producer for bituminous fuel must meet, and a discussion of some of the factors that will influence its performance.

THE RECIRCULATING BLOWER

7 In good anthracite-producer practice about one-half pound of steam should be decomposed per pound of combustible burned. This steam is mixed with the air for combustion as it enters the fuel column, and is obtained either from a vaporizer on the producer itself, or from some outside source. The possibility of using this steam to circulate the gases was taken advantage of by the earliest investigators. The steam exhauster is so obviously the best type of blower to use that the problem simplifies itself into a choice of the type of the steam blower.

8 A very inefficient blower can deliver the required amount of gas with less than a half pound of steam, but some classes of bituminous fuel contain so much moisture that a highly efficient blower is desirable. This with other requirements that the blower must meet are as follows:

a The required amount of gas must be delivered with the use of the least steam.

b The blower must be capable of operating continuously without clogging up or varying from its normal delivery of gas.

c In delivering the gas the blower must not cause the precipitation of too much of the tar.

d The blower should be accessible and easily cleaned.

e Since change in the character of the fuel column will change the pressure against which the blower works, the blower's capacity must not be appreciably affected by the discharge pressure.

9 Comparative tests were made on blowers of three different types, all using the same amount of steam per hour, and all working against the same back pressure. The construction of the first type may be understood from Fig. 2. The body of the blower was made up of standard 4-in. pipe fittings. The steam inlet *A* is a 1-in. pipe 15 in. long, threaded its entire length in a lathe. The pipe *B* was also threaded in a lathe, to make it fit straight into the tee *C*. The plug *D* was screwed securely into *C*, and then put in a lathe, centered, drilled and threaded to fit the pipe *A*. The pipe cap *E* was put on the pipe *A* and then drilled in a lathe along the center line of the pipe with a No. 49 drill. Within the pipe *B* is a wooden cylinder *F*, 12 in. long and 4 in. diameter, bored out along its center line to a taper 1½ in. diameter at the upper end and 4 in. at the lower, and above this is a throat piece *G*, bored to 1½ in. diameter at its center, and rounded at the top

as indicated. Taken together, the wooden blocks *F* and *G* form the expanding pipe of the blower. When set up as indicated the stream of steam from the No. 49 hole in *E* will pass along the center line of the pipe *B*, and the whole apparatus will serve as a steam exhauster, the air being drawn in at the side opening of the tee *C*, and forced out through *B*. By turning the pipe *A* in the plug *D*, the distance from the lower end of *E* to the upper end of *F* could be varied from ½ to 7 in.

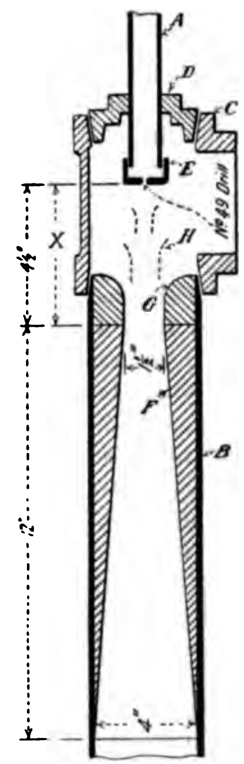


FIG. 2 RECIRCULATING BLOWER

The throat pieces *G* were made of different shapes and lengths, and the cylinders *F* were made in lengths varying from 4 to 12 in. Likewise the pipe caps *E* were supplied with holes of different sizes. The large number of duplicate parts was provided so as to arrive experimentally at the combination giving the highest efficiency and best performance.

10 The second type of blower was the same as the first, except that the steam orifice *E* had a number of holes placed so as to dis-

tribute the steam evenly over the entire area of the top of *F*. The size of the holes was chosen so as to use the same amount of steam as the nozzle in the first type with which it was compared.

11 The third type was the same as the first except that a varying number of air-expanding pipes was used between the steam orifice and the final air-expanding pipe, as indicated by the dotted lines at *H* in Fig. 2. These pipes were used in a variety of numbers, sizes, shapes and spacings, in an attempt to get at the best combination for the conditions under which they were to operate.

12 The assembled equipment for the tests is shown in Fig. 3 and consisted of a steam gage *A*, the steam line *B* from the boiler, a bleeder line *C* used to rid the pipe *B* of condensed steam, a sheet-steel cylinder *D*, 9 in. diameter, a Keuffel & Esser anemometer *E*, which was placed 10 in. back from the end of the cylinder *D* during the tests, a water manometer *F* to indicate the pressure against which the blower was working, and a valve *G* to regulate the pressure worked against. The anemometer indicated the number of feet of air that passed through the cylinder *D* in a minute, or in other words, the capacity of the blower under the conditions of the test.

13 In testing out the first type a chosen orifice cap was used with each combination of the different throat pieces and expanding pipes. With each combination of the three the distance from the orifice to the throat (*X*, Fig. 2) was varied from well below to well above the point of best performance, and with each setting runs were taken at different back pressures. These were usually the minimum that could be obtained, $\frac{1}{2}$, $\frac{3}{4}$, 1, 2, 3 and 4 in., and the maximum that the blower could maintain with no delivery. During each run a record was kept of the steam pressure, anemometer readings, time and manometer readings. From these records the best running conditions for the blower of the first type were determined. Next an orifice was chosen that could maintain a maximum pressure of about 12 in. of water with a zero delivery, and the best distance *X* was determined for this steam orifice. This was then used as the standard blower of the first type, and the one with which types II and III would be compared.

14 It was assumed that the combination of throat piece and expanding tube that was best for type I would also be best for type II, the tests on which were therefore confined to trying out different numbers of orifices in the cap *E*, Fig. 2, and different distances *X* with each of these new orifices. The tests run were similar to those on type I.

15 Similarly the tests run on type III consisted in trying out different numbers and different arrangements of the auxiliary expansion pipes *H*, Fig. 2, until the best arrangement was found, and then in testing out the possibilities of the blower under these arrangements. As a matter of fact, the fewer the number of the auxiliary pipes the better the performance, so that it finally developed that

FIG. 3 BLOWER UNDER TEST

the task was to find the best arrangement for these pipes when a given number was used, and then in comparing these results with those obtained with a different number.

16 The results showed that type I was more efficient than any arrangement of either of the other two. Type III was almost as efficient as type I, but when handling a tar-laden gas the smallest auxiliary throat was apt to become clogged with a mixture of tar and dust, or a piece of coal lodging in it. This would stop the action of

the blower entirely. Type I was therefore chosen as most efficient, most dependable for continuous performance with the least chance of trouble, and easiest to clean.

17 In interpreting the results it must be remembered that these tests were made to choose a blower to work against a back pressure of not more than about 2 in. of water, and that the results may or may not be similar to those that would be obtained if working against

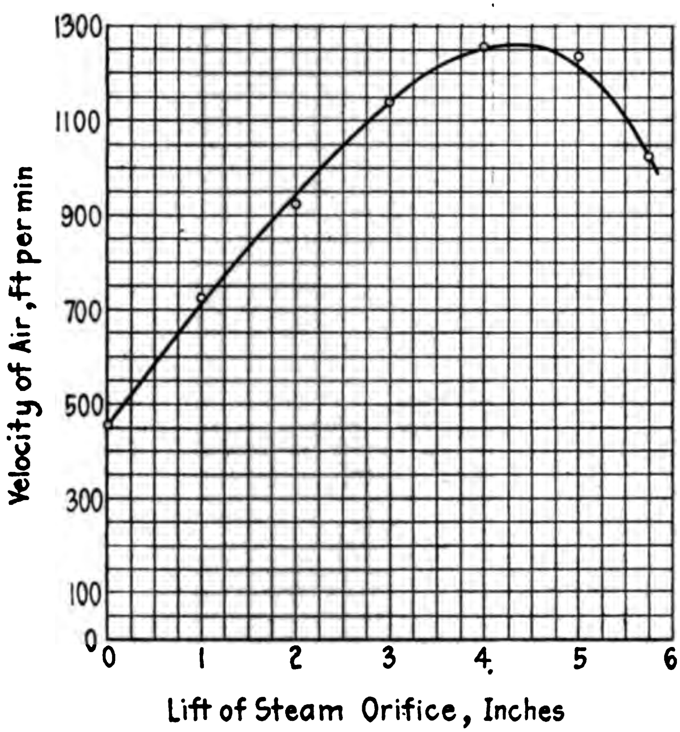


Fig. 4 Effect of Changing Distance of Steam Orifice from Top of Expanding Pipe

a higher pressure. The dimensions shown on Fig. 2 are those of the blower that was finally chosen as the best.

18 Fig. 4, a performance curve of this blower, shows the effect of changing the distance of the steam orifice above the top of the expanding pipe. Fig. 5 shows the effect on the capacity of the blower of changing the pressure against which the blower works, and Fig. 6 the effect of changing the steam pressure at the orifice. A blower of the dimensions here used can deliver between 12,000 and 15,000 cu. ft.

of air per hour against a back pressure of one inch of water, when supplied with steam at 135 lb. gage.

19 The following general conclusions may be drawn from the results of these tests.

a In working against small back pressures probably the most efficient and best type of steam exhauster is the one shown in Fig. 2.

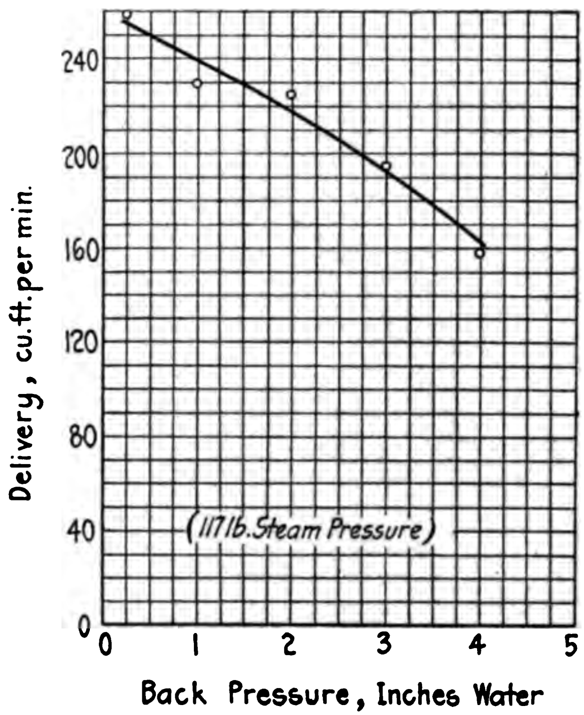


FIG. 5 EFFECT OF CHANGING PRESSURE WORKED AGAINST

b The height of the steam orifice above the expansion pipe is an important factor in this type of exhauster.

c The length and the angle of taper used in the expansion pipe are both important.

d The capacity of the blower is almost directly proportional to the steam pressure at the orifice.

e In a separate set of tests it was shown that superheated steam gave better efficiency and threw down less tar than saturated or wet steam.

PROPER TEMPERATURE FOR THE RECIRCULATED GASES

20 The second line of investigation was carried out in an attempt to determine the temperature at which the recirculated gases, the blower and the recirculating pipe should be held to throw down the smallest amount of tar, and give the least trouble. One of the most searching tests for tar is to cause the gas to impinge against a hard

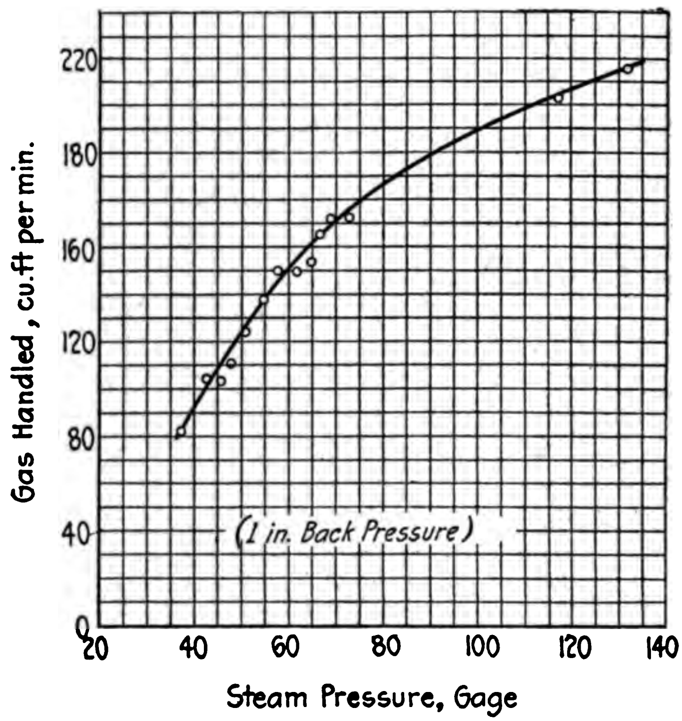


FIG. 6 EFFECT OF CHANGING STEAM PRESSURE AT ORIFICE

surface at a high velocity. This was done, the gas and the surface impinged against being held at different temperatures during the different trials. The apparatus used was therefore called upon to accomplish a number of different things:

a It must furnish a sufficient supply of the tar-laden gas.

b It must furnish an accurate means of measuring the gas temperature.

c The temperature of the gas must be under good control.

d A good means must be supplied to measure the temperature of the surface impinged against.

e The temperature of this surface must be under good control.

f The gas must be caused to impinge against this surface at a high velocity.

g An accurate means must be supplied for measuring the amount of deposit from a given amount of the gas.

21 The apparatus used is shown in Figs. 7, 8, and 9. The gas was generated in a furnace made of a 24-in. length of 6-in. pipe, fitted

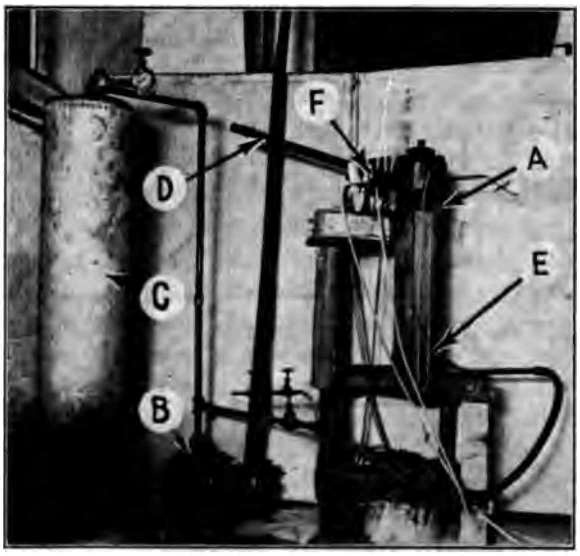


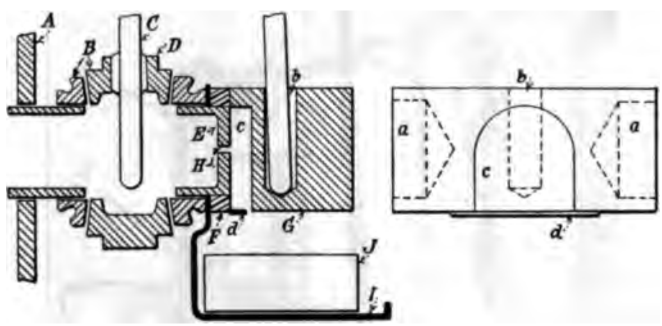
FIG. 7 APPARATUS FOR DETERMINING PROPER TEMPERATURE FOR RECIRCULATED GASES

with a coupling and plug at both top and bottom, and with a set of grates near the bottom. This is shown at *A*, Fig. 7. The air blast was furnished by the blower *B*, and the pressure of this blast was steadied through the tank *C*. The fumes from the furnace were carried away through the pipe *D*, the pressure inside of the furnace and the amount of blast passing through it being regulated by the valve at the top of tank *C*, and the valve on the pipe line *D*. The pressure in the furnace was indicated by the mercury manometer *E*.

22 The apparatus for testing for tar is shown at *F* and a cross-section is given in Fig. 8. Here *A* is the side of the furnace, which is

connected by a $\frac{3}{4}$-in. short nipple to the $\frac{3}{4}$-in. cross *B*. The thermometer *C* is securely packed in the plug *D*, and extends directly into the stream of gas, thus being able to indicate its temperature very accurately. The steel plug *E* is threaded to fit *B*, and is also threaded into a steel bar of $\frac{1}{2} \times 1\frac{1}{4} \times 2\frac{3}{4}$ in. dimensions, shown at *F*. The outer end of *E* and the corresponding face of *F* are machined to a single finished surface, and the copper block *G* is clamped against this surface. The copper block furnishes the surface against which the gas is to impinge. It is a very good conductor of heat, so that it is possible to keep all parts of the block at nearly the same temperature.

23 The shape of the block is indicated in Fig. 9. The ends are drilled as shown at *a* so that, to aid in heating the block, a blowpipe



FIGS. 8 AND 9 DETAILS OF TAR TESTER

flame may be directed into each hole. The face of *G* bearing against *F* is milled out $\frac{1}{4}$ in. deep as shown at *c* to furnish an eddy chamber for the gases, which would impinge against it at a point directly opposite the orifice *H* in the plug *E*. The temperature of the surface of the block at this point is indicated by a thermometer resting in a mercury bath in the hole *b*. To prevent eddy currents of cold air from passing up into the milled chamber *c*, it was found necessary to restrict the outlet by means of the strip of sheet steel *d*.

24 In Fig. 8, *I* is a sheet-steel platform designed to hold a paper tray *J*. This tray is held in such a position that any tar that is deposited in *c* and runs out at its lower edge will be caught in the tray. By keeping the pressure in the furnace constant at 3 in. of mercury, and keeping the hole *H* clean, a constant quantity of gas will pass through the device per minute. By holding the temperature of the gas and the block constant for 4 min., catching the deposit for that

length of time in the tray J and in the chamber c, this deposit could be accurately weighed and compared to the deposit from a similar unit quantity of gas at any other temperature.

25 The method of procedure in making a run is therefore as follows: The blower B, Fig. 7, is started and a fire is lighted in the furnace A. The furnace is then filled with coal from Carterville, Ill., the gas from this coal being heavily laden with tar. The top and bottom plugs are then put in the furnace and the fire is blown up vigorously, the temperature of the gas being thus caused to rise quite rapidly. The block G, Fig. 8, is then heated by the blow torches being applied to its ends. The operator brings the temperatures of the gas and the block, to the desired point and then regulates the blast and blow-torch flames so as to hold these temperatures constant, and also keeps the pressure in the furnace at 3 in. of mercury. Then the waste tray is removed, the test tray is put in place and the block is moved up and clamped in place. The temperatures of the block and gas are then held constant for 4 min., the pressure in the furnace being held at 3 in. on the mercury manometer E, Fig. 7. At the end of this time the tray and block are quickly removed, the block is cleaned of its tar, and the combined deposit found on the block and tray is carefully weighed. The fire is then cleaned, a new supply of coal added, the orifice H, Fig. 8, is cleaned, and a new run is made at a new set of temperatures.

26 The results show that at the lower temperature comparatively large amounts of tar are deposited, just as would be expected. This tar is soft and sticky at room temperature and contains considerable water. As the temperature of the deposit is raised, the amount of the deposit is continuously decreased, and the tar when cooled to room temperature becomes harder and more brittle. All of the tar is thin and liquid at the temperature at which it is deposited, so that so long as the deposit is a tar, the block "G" tends to have only a thin coat on it, the larger part of the deposit appearing in the paper tray. After a while a temperature is reached at which the deposit, though still a tar, is almost too small in amount to be weighed. This temperature of small deposit starts at about 280 or 300 deg. cent. and extends up to about 500 deg. cent. At no point is there a place where the deposit entirely disappears. At about 500 deg. the deposit begins to turn over from a tar to a sort of coke, and instead of being fluid, it all sticks to the block. Between 500 and 600 deg. this deposit increases slowly in amount, and then increases more rapidly as the temperature nears 700 deg.

27 The condition where the gas and block are at the same temperature is the only one which has been worked out carefully. The results for this condition are shown by the curve on Fig. 10. The best temperature at which to recirculate the tar-laden gas seems to be between about 300 deg. and 450 deg. cent. At any point within this range the deposit of tar is small, and such as there is tends to flow off from the surface of the blower. At temperatures higher than 450 deg. cent. there is a chance that the deposit will accumulate on the walls of the blower and stop it up.

28 Experience has shown that a heavily-tar-laden gas in passing through a cold bed of coal will deposit its tar in the coal until the whole layer of coal will become impervious to the gas. This state of

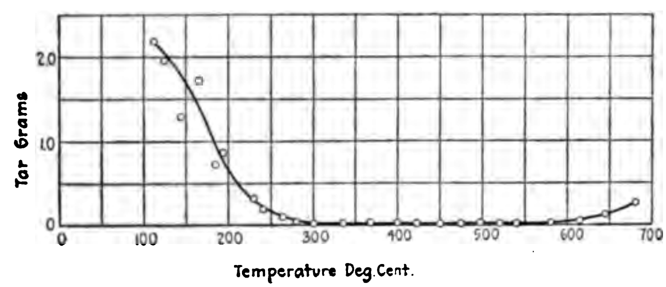


Fig. 10 Variation of Tar Deposit with Temperature

affairs would be disastrous to the operation of a gas producer. For this reason it becomes interesting to know the lowest temperature at which one may be assured that such a thing will not happen. The tests just reported answer this question as well. If the coal is above 280 deg. cent. this deposit cannot occur.

THE TAR-FORMING TEMPERATURES

29 The next line of investigation was carried out in an attempt to determine the tar-forming temperatures of different samples of American coals, or, more particularly, the lowest temperature at which one may be certain that the last trace of tar has been driven off from the coal. The temperature conditions met with in the fuel column of the gas producer were followed as closely as possible in these laboratory tests. This made it necessary to place the following requirements on the apparatus:

a The coal must be heated very slowly, and at a uniform rate.

b The heat must be conducted from the outside to the center of the body of coal by some good conductor, as the coal itself is a very poor conductor of heat and all particles in the body of coal must always be at a uniform temperature during the heating.

c The temperature of the coal must be accurately known at all times.

d The gases driven off from the coal must be driven out as soon as formed.

e The gases must be cooled down and continuously tested for tar.

f Any tar deposited in the pipes at a lower temperature must not be allowed to redistill at a higher temperature and then appear in the gas.

Apparatus was designed with the greatest care to meet all of these conditions.

30 After considering the various possible means of heating the coal, it was decided to use an electric resistance furnace. By this means the coal could be heated at any rate desired and the rate of heating could be controlled at all times, or the coal held at any desired temperature for long periods of time. The furnace used is shown in cross-section in Fig. 11. It is 20 in. long and 3½ in. in internal diameter. Nichrome wire was used for the resistance, and the lagging was made up of about two inches of asbestos pipe covering. A maximum of six or seven amperes of current was required, which was measured by an ammeter in the circuit, as shown in Fig. 12.

31 The cartridge in which the coal was placed to be heated was made of 2-in. pipe fittings, as shown in Fig. 11. The body of coal heated was therefore 2½ in. in diameter, and it was necessary to have all the particles of coal in this mass at the same temperature. As coal is a poor conductor of heat, it was decided to place iron disks ¼ in. apart throughout the entire length of the cartridge. These disks are shown at *A*, Fig. 11. They are large enough to touch the iron cartridge all around, thus taking on its temperature, and were drilled full of small holes to allow the gas to pass through them. Thus the heat had to be conducted through only ⅛ in. of coal. This, with the very slow rate of heating employed, probably caused the temperature throughout to be the same within very close limits. The temperature was read at the center of the coal body by a thermocouple the

end of which extended down to and touched the end of the thermometer well shown at *B*, Fig. 11.

32 The thermocouples used were iron and nichrome wires welded together in an electric arc. They were used with a Brown millivoltmeter with a resistance of 85 ohms. The pyrometer thus formed was carefully calibrated, and when rechecked after the experiments were completed was found correct to within 10 deg. cent. throughout the range of temperatures here reported. The couples were left in place throughout each test, and the temperature readings made whenever desired.

33 To sweep the gases out as they were formed, air was forced into the cartridge under pressure, through the ¼-in. pipe *C*, and allowed to pass out through *D* in a constant stream. The pressure of this gas inside the cartridge was measured by a mercury manometer, and was kept at about 2½ in. of mercury. This gas could not be allowed to contain any O_2, as it might then burn the coal or tar vapors at the higher temperatures, so air with the O_2 burned out was used. The arrangement of the apparatus used to accomplish this is shown in Fig. 12. The air was burned in a small furnace filled with an anthracite fire. The air pump pumped it through the furnace and cooling coil and compressed it into the large air tank, where it was stored for use. By this means the supply of gas for a complete test could be stored up before the test itself was started.

34 Another precaution had to be taken in using this gas, as it could not be allowed to affect the temperature of the coal as it passed through. To prevent this, a coil of pipe was placed over a gas flame and the gas passed through and heated up to the temperature of the coal, before it was allowed to enter. The temperature of the gas was measured by a thermocouple that extended into it through a tee in the pipe line. To make assurance doubly sure the end of the cartridge itself was filled with steel chips, in the space at *E*, Fig. 11. The entering gas was thus forced to pass through a considerable volume of these chips before coming in contact with the coal. It was found difficult to heat the gas up to the highest temperature of the producer. This might tend to affect the seeming temperature at which the last traces of tar appear. The tendency of the gas would always be to be lower than that of the coal. For this reason the end of the thermocouple was placed in the coal at the end where the gas enters it, and therefore at its cooler end, in case there should be any difference at all. Thus the temperature reported as the one at which the last trace of tar appears is as accurate as it is possible to make it.

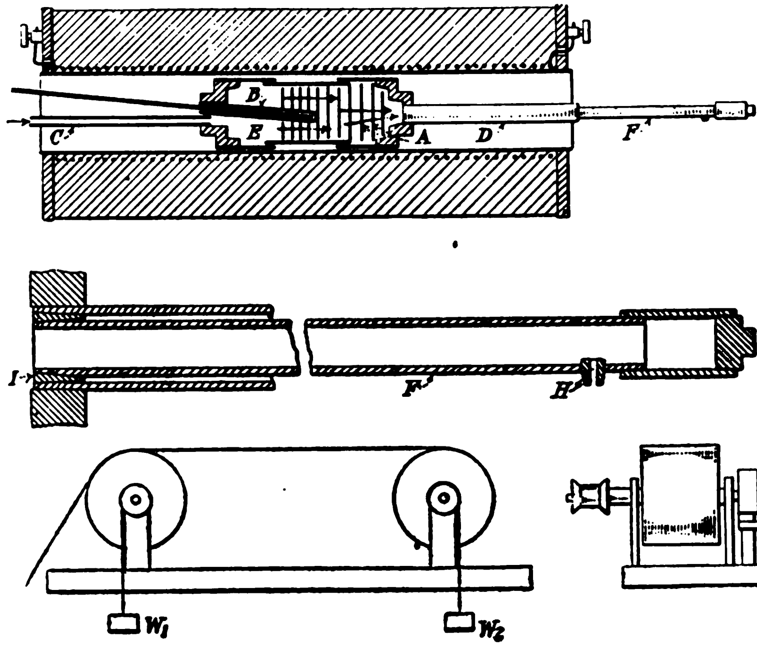


FIG. 11 CROSS-SECTION OF TAR-TEMPERATURE APPARATUS

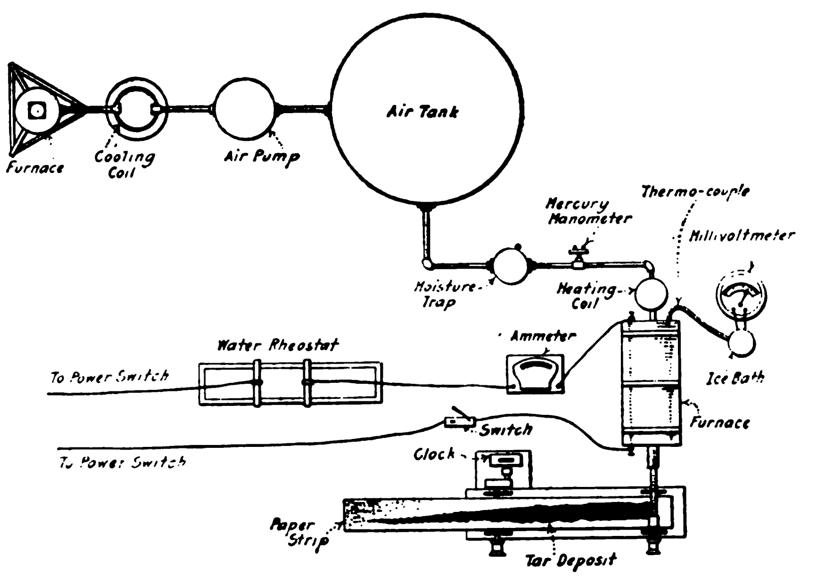


FIG. 12 GENERAL ARRANGEMENT OF APPARATUS

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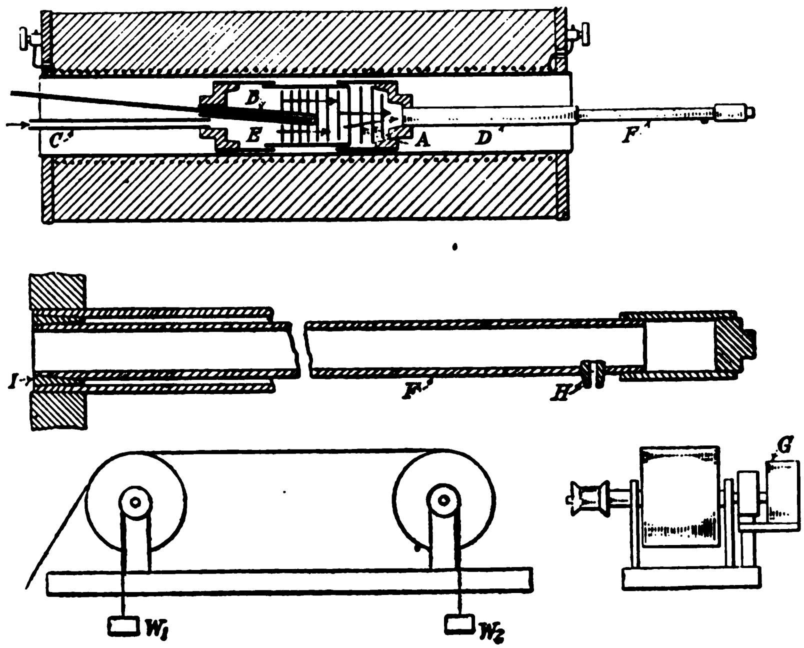

FIG. 11 CROSS-SECTION OF TAR-TEMPERATURE APPARATUS

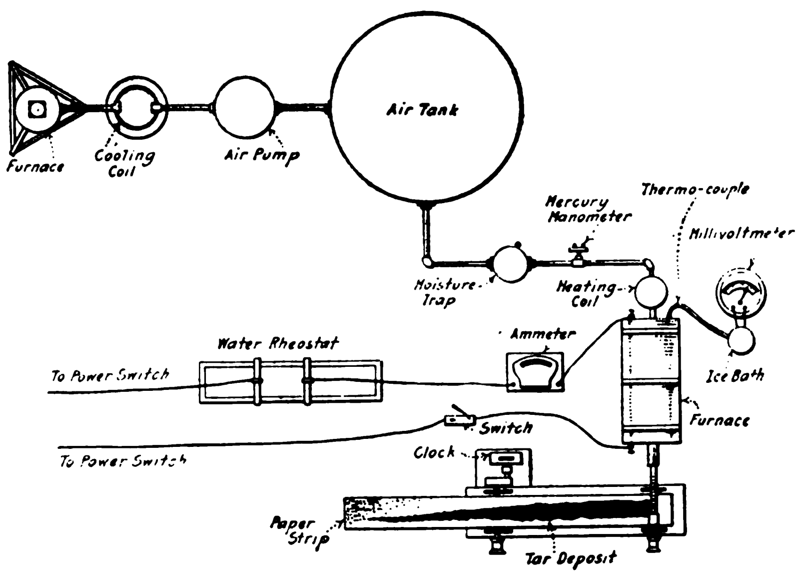

FIG. 12 GENERAL ARRANGEMENT OF APPARATUS

35 The next problem was to find a means of subjecting the gases from the coal to a continuous test for tar. The most satisfactory test known to the author, and the one used by gas companies, is to allow a small stream of the gas to strike a piece of white paper at a high velocity. If there is any trace of the tar at all in the gas, it soon leaves a spot on the paper. This test was adopted. To use it, the gas must be cooled down before it strikes the paper. This was

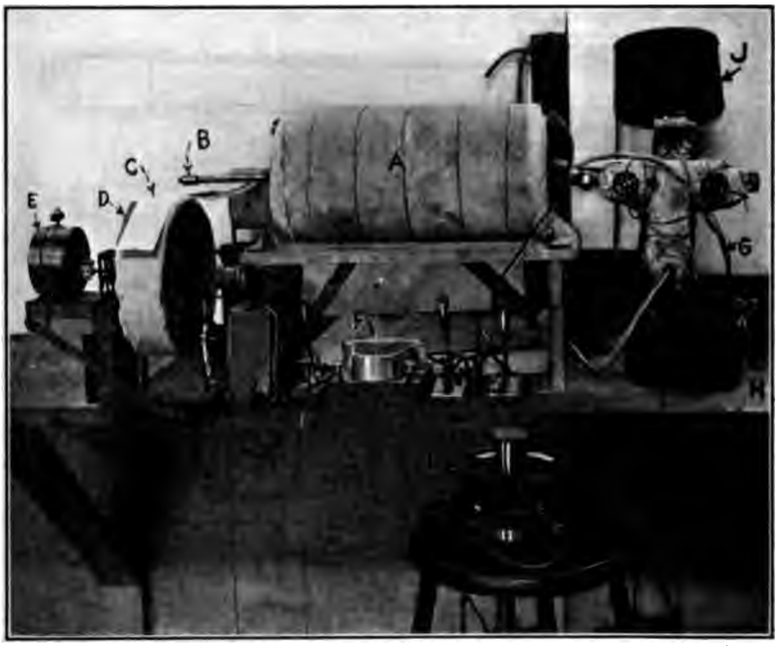

FIG. 13 PHOTOGRAPH OF TAR-TEMPERATURE APPARATUS

accomplished by keeping a cloth filled with cold water constantly lying on pipes *D* and *F*.

36 To make the test continuous, the device shown in Figs. 11 and 12 was used. Rollers about 10 in. in diameter are supported by steel rods through their centers, the rods turning freely in iron supports at either side. On the end of each of these steel rods was placed a small wooden spool, around which was wound a cord, supporting weights W_1 or W_2. A long strip of cloth was wound around one of these rollers and its end started around the other. A piece of paper ribbon was wound on with the cloth. W_1 and W_2 tend to turn the

rollers in opposite directions, thus keeping the cloth and paper strip tight. W_1 is enough heavier than W_2 to cause both to turn, unless held back in some way. The key of an alarm clock G was fastened to a train of gears and these in turn to the stem of the roller, so that the rate of motion was held back to a speed governed by the running of the clock. In this case the speed was 30 in. an hour. The rollers were set in such a way as to cause the paper to pass about $\frac{1}{4}$ in. under the end of the orifice H in pipe F. The strip of paper was marked at the beginning and end of the test, so that the exact time when any point on the paper was under the orifice could be determined. Thus a continuous record was kept of the amount of tar in the gas.

37 All of the tar is not taken out of the gas by this means. The tar particles in the cooled gas are very fine and light, and many of them are cushioned off from the paper, and never touch it to stick. Consequently the tar deposited on the paper does not represent all of the tar content of the coal. On the other hand, the slightest trace of tar in the gas will quickly blacken the paper, and the deposit is probably very nearly proportional to the entire tar content of the gas.

38 The remaining problem was that of making sure that no tar could be deposited in the pipe F at a low temperature and later be redistilled, to show up on the paper at too high a temperature. To accomplish this a large number of duplicate pipes were made and carefully fitted into the threaded bushing I. By changing the pipes F every five or ten minutes, the effect of such a tendency was quite completely eliminated. A photograph of this apparatus is shown in Fig. 13.

39 The coals tested were chosen to represent the different American grades. They were first ground and screened over a mesh of 20 wires per inch and through one of 10 wires per inch. About 200 grams of coal will fill the cartridge, and it was put in layers $\frac{1}{4}$ in. thick between the iron disks. The cartridge was then placed in the middle of the furnace, the thermocouple put in place, and the asbestos packed in against both ends, cutting off the radiation here, and causing the ends and middle all to keep at the same temperature. A current of 6 amp. was then passed through the resistance wire of the furnace. While the latter was coming up to the temperature where the light oils start to come off, the thermocouples were connected up, the flame placed under the gas preheating coil, the gas from the storage tank turned on, the paper rolls connected up to the clock, and the clock started. The pipes F were cleaned and prepared for immediate use, and the data sheet prepared. As soon as the thermo-

couple indicated a temperature close to that at which there was prospect of an oily deposit, the paper rolls were put in place, the paper marked and the test started. When the tar commenced to appear the temperature of the furnace was recorded and the pipe *F* changed every ten minutes. As the temperature of the furnace rose the current was increased to take care of the increased radiation and to keep the temperature rising at a constant rate. When the paper ceased to show any signs of tar deposit it was again marked and timed and the current shut off.

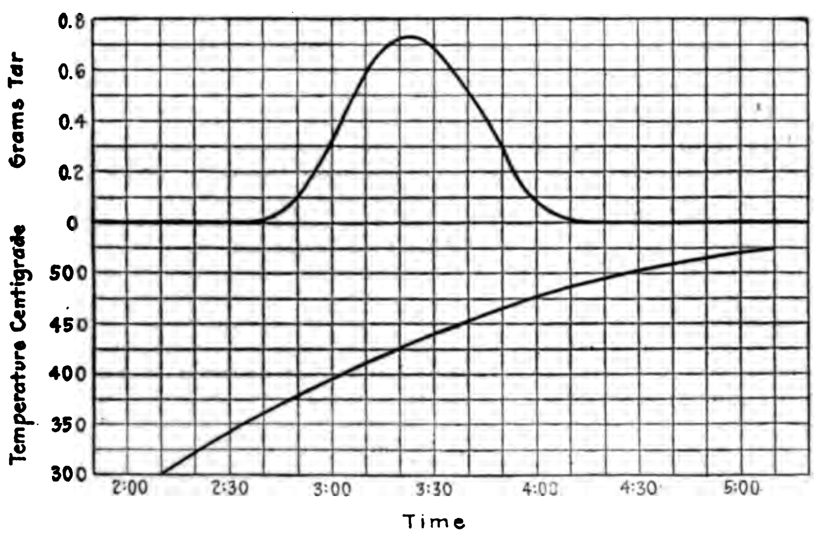

FIG. 14 TIME-TEMPERATURE TAR CURVE OF CENTERVILLE, ILL., COAL

40 The strip of paper was then cut up into lengths corresponding to 10-min. periods and carefully weighed. As the weight of the paper per inch was very constant, the excess in weight over that of the clean paper was in each case due to the tar. From this two curves could be drawn with time plotted horizontally, while one had temperature centigrade and the other grams of tar plotted vertically. These curves when placed one over the other, as in Fig. 14, indicate the amount of tar coming off at each temperature. The points where the tar starts and stops cannot be indicated by this curve, as the ends of the deposit are too thin to have appreciable weight. They are consequently separately noted elsewhere. A better understanding of what Fig. 14 represents may be obtained by referring to

Fig. 15. Here a strip of paper with the tar deposited on it is shown pasted to a time-temperature chart representing the time and the temperature at which each part of the tar was deposited. In order to make this strip short enough to photograph, the speed of the rollers was greatly reduced and the rate of heating the coal was increased.

41 The first condensable gas to be driven off from the coal and to appear on the paper record is water vapor. After the last of the water has disappeared there is quite a temperature range through which there is no deposit at all. Then the paper will begin to show a slight trace of oil. This will gradually increase in amount and give the paper the appearance of having been paraffined. The deposit

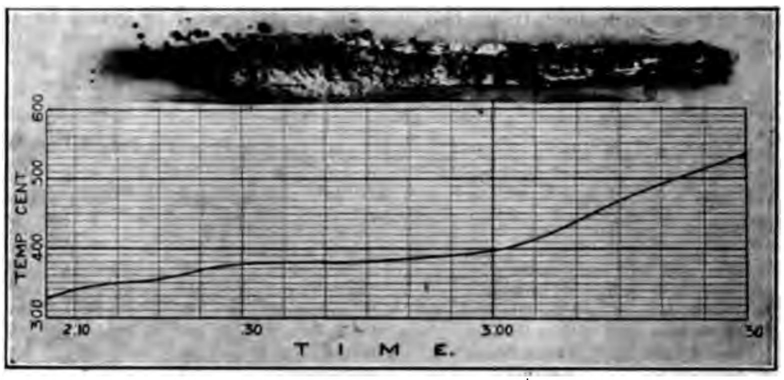

Fig. 15 Photograph of Strip of Tar Deposit on Time-Temperature Chart

will then gradually assume a brownish color, as though engine oil were appearing. Later a temperature will be reached at which the deposit will increase very rapidly in amount, and will assume a distinctly tarlike appearance. The first tar to be deposited is usually very soft and sticky at room temperature. As the temperature rises the tar becomes steadily stiffer, until it is finally hard and brittle when cooled. The temperature range through which the maximum deposit occurs will vary from about 100 to 175 deg. cent. At the higher limit the deposit becomes rapidly smaller in amount until it is too small to weigh, but the paper is still distinctly browned. This discoloration becomes less and less plain, until it finally disappears entirely.

42 There is no definite temperature at which the first and the last trace of the tar appear, in the same sense that water has a boiling

temperature. The first deposit is so indistinct that it is almost impossible to tell whether there is a deposit or not. The increase is also so very gradual that it is difficult to decide at what temperature to report the first appearance of a deposit. This gradual increase will extend over a temperature range of from 50 to 150 deg. cent. before there will be a sufficient deposit of tar to feel sticky to the finger. The determination of the temperatures between which the maximum deposit occurs is likewise more or less arbitrary, as is also the temperature at which the last trace of tar appears. Therefore the results as here reported must not be too literally interpreted. However, they are a very careful estimate of the facts as they are, and the highest temperature reported is one at which one may feel assured that the very last trace of tar has disappeared from the coal. A summary of the results obtained from the various coals tested is given in the following table:

TABLE OF TAR-FORMING TEMPERATURES

Field	State Found	Character	Temp. Tar Deposit, Deg. Cent.		
			Initial	Maximum	Final
Eastern	Virginia	Bituminous	300	385–550	580
N. Appalachian	W. Virginia	Pocahontas	310	480–550	600
N. Appalachian	Ohio (Hocking Valley)	Bituminous	284	422–515	684
N. Appalachian	Ohio (Jackson)	Bituminous	300	340–518	615
N. Appalachian	Pennsylvania	Bituminous	385	452–530	602
N. Appalachian	Kentucky	Bituminous	270	354–534	742
S. Appalachian	Tennessee	Bituminous	306	400–500	600
E. Interior	Indiana	Bituminous	240	340–510	560
E. Interior	Illinois	Bituminous	282	420–450	585
E. Interior	Illinois	Bituminous	300	375–520	610
E. Interior	Kentucky	Bituminous	288	400–520	600
N. Interior	Michigan	Bituminous	275	340–550	600
W. Interior	Arkansas	Bituminous.	....	420–580	700
W. Interior	Iowa	Bituminous	200	320–520	560
W. Interior	Missouri	Bituminous	270	400–499	590
W. Interior	Missouri	Bituminous	262	400–450	600
Western	North Dakota	Lignite	240	300–380	550
Western	Wyoming	Bituminous	260	400–450	557
Western	Utah	Bituminous	275	397–527	610
Western	Utah	Bituminous	291	340–517	560
Western	Utah	Lignite	235	393–534	665
Western	Colorado	Lignite	270	350–405	515
Western	Oklahoma	Bituminous	310	430–490	547
Western	New Mexico	Lignite	312	380–460	525
Southern	Texas	Lignite	200	325–483	670

43 The results obtained in these special lines of investigation have a direct bearing on the problem of developing a gas producer of

the recirculating type for the use of bituminous fuel. The more important conclusions may be summarized as follows:

- *a* The type of steam exhauster shown in Fig. 2 is efficient and reliable, and is probably the best to use in connection with a gas producer.
- *b* The recirculated gas and the pipes through which it passes should be kept between 550 deg. and 900 deg. fahr. in temperature.
- *c* The coal at the top of the fuel column should be kept above 550 deg. fahr.
- *d* The lowest temperature at which one may be certain that the last trace of tar has been driven off from the coal will vary with the coal used, but will lie between 1000 deg. and 1250 deg. fahr.

A table of these temperatures in degrees centigrade is given on p. 944.

44 Any producer for the use of bituminous fuel should meet all of the requirements which an anthracite plant must meet and some others. It is difficult to list all, but the following are among the more important:

- *a* The plant must be simple in construction, and must wear well.
- *b* All parts of the fuel column must be readily reached by the poker.
- *c* A ready means must be provided for cleaning out the clinker and ash.
- *d* The air must enter evenly over the entire area of the bottom of the fuel column.
- *e* The steam must be evenly distributed through the air for combustion.
- *f* The air must be equally free to circulate in all parts of the fuel column.
- *g* The gas must be so taken from the producer as to cause an even circulation through the upper part of the fuel column.

PRODUCER DEVELOPED BY AUTHOR

45 Fig. 16 is from a photograph of a 50-h.p. installation of a producer made in the laboratories of Purdue University at Lafayette, Ind., for experimental purposes. Peep holes *A* are provided to enable

the operator to examine the temperature of the fuel column. In the hottest part, where the temperature changes are the most important, this is especially easy. A tube with a rod inside of it may be thrust

FIG. 16 EXPERIMENTAL PRODUCER IN PURDUE LABORATORY

into the center of the producer, and the rod removed. By looking through the tube the temperature of the fuel at the end of the tube may be estimated. By drawing the tube out slowly and watching the temperature at its end, comparatively slight differences in temperature between the outside and the center of the fuel column may be detected. This is often of great importance in interpreting producer performance. These peep holes are drilled through the shell and tapped ¾-in. pipe size. Nipples 6 in. long are screwed into these holes, and are capped on their outer ends by standard unions with glass packed in them. The small outlets shown at the sides of many of the ¾-in. pipes are used to draw off samples of the gas to be tested for tar.

46 The trap *B* is filled with water and catches any tar that may be thrown down in the steam blowers or the recirculating pipes. The header *C* communicates with the gas-distributing device under the grates. The steam blowers *D* and the recirculating pipes are in duplicate, one being used and one held in reserve. The gas from this plant is cooled in a coke-filled scrubber that is part of a standard 50-h.p. anthracite plant made by Fairbanks, Morse & Co. From this scrubber the gas goes directly to a 43-h.p. Fairbanks-Morse engine.

47 Fig. 17 is a diagrammatic sectional elevation of this recirculating gas producer.[1] The coal is trapped in at the top, the only air inlet being below the grates. The combustion all takes place right above the grates, forming a layer of ash immediately on top of the grates, a combustion zone above this ash, an incandescent zone above the combustion zone, and distillation zones of decreasing temperature as they near the top of the fuel column. The finished gas is drawn off from the fuel column into the annular chamber, *A*, which is formed in the firebrick lining of the producer. From here it passes out through the pipe *B*. The products of distillation and the other recirculated gases are drawn into the pipe *C* by the steam blower *D*. These gases are then delivered into the header *E*, and from there into the distributer *F*. This distributer is so constructed as to deliver an equal amount of the recirculated gases to every part of the grate area, and in such a way that the gas burns as it mixes with the air, largely in the lower part of the combustion zone of the fuel. The recirculated gases burn to CO_2 and H_2O, the same gases that are obtained from the combustion zone of an anthracite fire. In passing through the incandescent zone above, the CO_2 is split up, and taking up one more carbon atom is changed into CO according to the equation $CO_2 + C + \text{heat} = 2CO$. In the same way the H_2O is split up into free H_2 and CO, according to the equation $H_2O + C + \text{heat} = H_2 + CO$. Thus the finished

[1] Patent No. 1116216.

gas obtained from the tars and the other hydrocarbons is the same as that from anthracite coal, and instead of being wasted the tars are converted into useful gas. The gas from this producer should therefore be free from the fluctuating hydrocarbon content that has been the source of so much inconvenience in connection with the gas from most bituminous plants.

48 Returning again to a consideration of the factors influencing the proper performance of this type of producer, if three things are accomplished the gas will be free from tar:

a All of the tar must be driven out of the fuel while it is still above the openings which communicate with the annular chamber.

b All of the tarry vapors driven out of the fuel in the distillation zone must be drawn out and recirculated.

c The recirculated gas must not be allowed to form a cold path for itself between the grates and the outlet on the one hand, nor form an explosive mixture with the entering air on the other. The former would allow the tars to get into the finished gas, and the latter would cause explosions which might prove serious.

49 It is possible to accomplish all three of these things continuously and certainly, and to get a gas from a heavily-tar-laden fuel which is entirely free from tar.

50 Inability to meet the first requirement has been the cause of failure of several recirculating producers. It must be remembered that coal is a poor conductor of heat and that very little heat can pass by conduction from one lump of coal to another. The heat distribution in the fuel column of a standard up-draft plant is due to the hot gases from the incandescent zone that pass up through the entire fuel column on the way out. Thus each individual lump of coal is heated by the hot gases that surround it. In the recirculating producer the finished gas is drawn out of the fuel column at about mid-height. If the only gases that pass through the fuel above this point are the products of distillation of the fuel, the heat-carrying medium will be absent above the annular chamber, and the coal will remain cold until after it passes this point. This means that the first requirement has not been met and the producer will not be able to furnish a tar-free gas, and is a failure.

51 To get around this it is necessary to install a strong recirculating blower of sufficient capacity to recirculate a considerable part

of the hot gases from the incandescent zone along with the products of distillation. The quantity of gas recirculated will depend upon the quantity of heat which it is necessary to have above the middle, and there is no reason why it cannot be increased until it is equal to

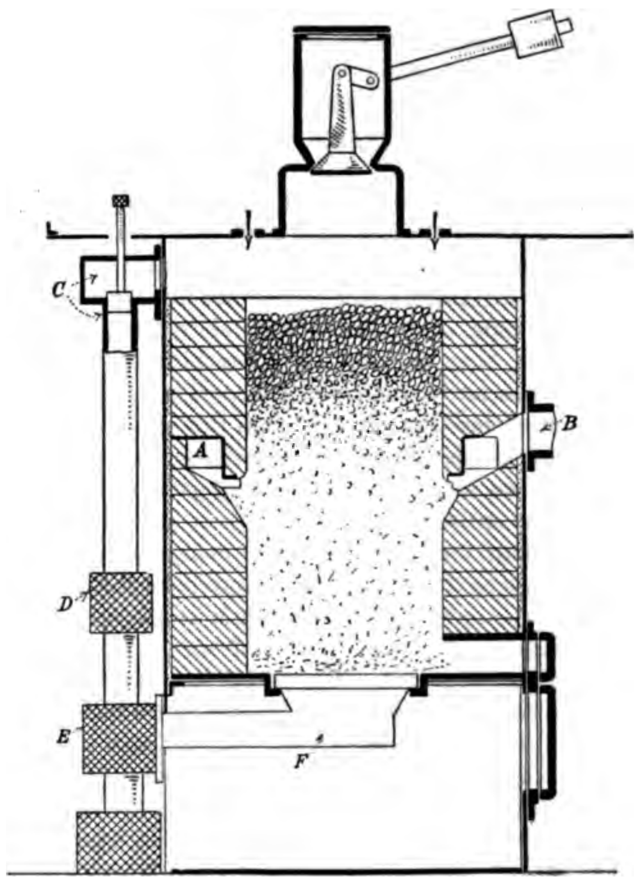

FIG. 17 AUTHOR'S PRODUCER IN SECTION

the quantity of gas drawn off by the engine, or, in other words, there is nothing to prevent having as great a quantity of hot gases passing through the upper part of a recirculating producer as pass through the same part of a standard up-draft plant of the same capacity. It is therefore possible to maintain the same high temperatures in the fuel column at the top of a recirculating producer that are met with in the up-draft producer, and the coal may be heated until the last trace of tar has been driven off from it before it reaches the gas discharge level, and the first requirement may therefore be met.

52 This positive stream of gas passing from the bottom of the producer to the top past the annular chamber also enables the producer to meet the second requirement, as it will prevent any of the products of distillation that are driven off from the coal above the discharge level from backing up and passing out at the discharge. As a gas cannot flow from a region of low pressure to one of a higher pressure, the tars will be forced to go through the recirculating pipe.

53 On the third requirement is where many recirculating producers have failed. The gases from any given point in the grate area will tend to establish their own individual path between the grates and the outlet. If they are inert, or do not mix with air and burn in the combustion zone, or, mixed with air, do not burn the coal in the combustion zone, they will not have any heat of combustion with which to establish an incandescent zone. They will therefore form a cold path for themselves right through the zone that should be incandescent, and escape through the discharge outlet unburned and uncracked. To avoid this the recirculated gases must be evenly distributed over the entire grate area.

54 To prevent the formation of an explosive mixture the gases and air must burn as they mix. In this producer this is accomplished by having the gas introduced into the air in the restricted area between the grates, thus preventing them from backing up and mixing with the air below the grates. The author has built a producer along the lines covered by the Whitfield patents, introducing the recirculated gases into the incandescent zone. A cold path was soon formed between the point where the gases were introduced and the outlet. This was learned by observing the temperatures of the gases in this path through the peep holes *A* of Fig. 16. In the producer as here proposed these cold paths are not formed, the temperature in all parts of the incandescent zone remaining practically uniform, and no explosions occur under the grates. The third requirement is therefore met satisfactorily.

55 The down-draft producer and the double-combustion-zone producer are both successful in delivering a tar-free gas. It is therefore entirely to the point to ask what is to be gained by developing another new type of bituminous plant, such as the one proposed. This can be best understood after reviewing the necessary short-

comings of these two types of plants. The down-draft producer is open to the following criticisms:

a The ash contains a high percentage of the carbon in the coal.
b A large part of the tars are cracked in a reducing atmosphere, and consequently produce large quantities of lampblack.
c The fuel column is subject to the formation of hard clinkers.
d A large proportion of the total heat value of the coal burned is lost in the sensible heat of the gases leaving the producer.
e Some types of the down-draft producer cannot be cleaned while running, but must be shut down every week or ten days to have the fire pulled. This is a disagreeable task, besides making it necessary to have two plants.

56 The double-combustion-zone producer gets around the first and fifth difficulties without adding any serious new ones, but it is an equally bad offender in the other three.

57 The high proportion of the total carbon in the coal that is lost in the ash is characteristic of all types of the down-draft producer. The older types that could not be cleaned while running obviously lose a considerable part of the carbon in the producer at the time the fire is pulled. In the newer types that are capable of continuous operation, the high carbon in the ash may be explained in that the combustion takes place at the top of the fuel column, but all of the carbon cannot be burned out of the fuel here, as about half of it must be left to reduce the CO_2 in the incandescent zone. This reduction in the incandescent zone is due not only to the high temperatures there, but also to the presence of a large excess of carbon. It is therefore utterly impossible to burn all of the carbon out of the fuel in the incandescent zone, and produce at the same time a gas that is reasonably low in CO_2.

58 When the tarry vapors are heated to a high temperature in an oxidizing atmosphere they burn and no lampblack is formed. If, however, the heating takes place in a reducing atmosphere combustion is impossible, and the tars will be cracked. This cracking process is always and necessarily accompanied by the formation of large quantities of lampblack. In both the down-draft producer and the double-combustion-zone producer the higher-temperature tars from the central part of each lump of coal are inevitably driven off in the incandescent zone. Hence the ever-present lampblack in the gases from these producers.

59 All of the factors that tend toward the formation of hard clinkers do not seem to be understood. Practice seems to establish the fact, however, that the trouble from clinkering will be augmented by concentrating the ash in the hottest part of the fuel bed, by stirring or poking this part of the fire, and by the existence of lampblack in the fuel. In both of these types of producers the coal is about half burned at the very top of the fire, and is as nearly completely burned as it ever gets to be while it still remains in the incandescent zone. In other words the coal at the bottom of the incandescent zone is nearly all ash. This incandescent ash is mixed with lampblack and is frequently and more or less vigorously poked. These conditions are inevitable in these types of producers, and help to explain the excessive clinkering uniformly met with.

60 The reports of the work done by the United States Geological Survey in its experiments with gas producers furnish much valuable information. In some cases they give the heat value of the coal, the number of cubic feet of gas obtained from one pound of the coal, and the chemical analysis of the gas. Knowing the chemical analysis of the gas a close estimate can be made of the quantity of undecomposed water that accompanied the gas as it left the producer. The specific heats of all of these gases are quite accurately known. They may be obtained from Volume I of Metallurgical Calculations, by Joseph W. Richards. From these figures the percentage of the total heat value of the coal that appears as sensible heat in the hot gases leaving the producer may be figured for different temperatures of the gas. It will be found that this percentage will be nearly equal to the number of hundreds of degrees fahrenheit of temperature of the gases. In other words if the gases are leaving the fuel column at 2200 deg. they are carrying away about 22 per cent of the total heat value of the coal.

61 The work of the Survey has also made it clear that to get a good reduction of the CO_2 it is necessary to have a temperature of at least 2200 deg. fahr. in the incandescent zone of the producer. In the down-draft producer, and in the double-combustion-zone producer as well, the only chance for the gas leaving the producer having a temperature less than that needed for the reduction of the CO_2 in the incandescent zone must be due to radiation from the shell of the producer. This radiation is not large, so the loss due to the sensible heat of the gases as they leave the fuel columns of these two types of producers will necessarily be above 20 per cent of the total heat value of the fuel used.

62 In the recirculating producer here proposed, all five of these difficulties are met without introducing any new ones of comparable gravity.

a The combustion takes place the same as in a standard updraft anthracite plant. The air all comes from underneath and passes through the ash before reaching the fuel. The last trace of carbon in the hot ash at the bottom of the combustion zone is exposed to the pure air which supports the combustion, and there is therefore no reason why all of the carbon cannot be burned up. In either the anthracite plant or the recirculating plant, the quantity of carbon in the ash is due largely to the care with which the fire has been tended, and should be and usually is small.

b All of the tarry vapors are burned in a highly oxidizing atmosphere, and therefore there is no lampblack formed.

c The fuel being burned after it has passed through the incandescent zone, just as it is in the standard anthracite plant, much of the tendency to clinker is avoided. As a matter of fact, no worse clinkers are produced in this plant than in the anthracite producer, and with proper care and a good coal they can be avoided entirely.

d The temperature necessary for the elimination of the tar in the coal is never more than 1250 deg. fahr. The gases will necessarily have to leave the producer at a temperature somewhat higher than this, say at 1500 to 1600 deg. This will effect a considerable saving in the loss of sensible heat in the gas.

e The ash can be removed from the producer while it is in operation, and the plant is capable of continuous operation, the length of the run being limited only by the life of the firebrick lining.

63 Thus far the author has not been able to carry out long and exhaustive tests on this plant, and regrets his inability to quote figures showing its performance. The best report that can be made is that the plant has run all day (10 hr.), each day for a week without making any tar. The heat value and chemical analysis of the gas were not obtained.

64 It seems logical to expect that some of the most successful

producers of the future will be built along these lines, and it is hoped that the lines of thought here presented will be of interest to power users.

DISCUSSION

GODFREY M. S. TAIT (written). The methods used by the author for the determination of the tar content are all highly practical as applied to the experimental apparatus used, but would, of course, have to be modified in connection with a test of a plant in regular service.

The author's theory as to the volume of recirculated gas is not original, having been tried with more or less success in Europe, the main difficulty experienced being the power consumed by the blower and the excessive quantity of steam thus introduced — unless some arangement were provided for connecting the governor on the gas engine to the by-pass blower so as to control same. With low load and the by-pass blower running full capacity, it was possible to lower the temperature of the fuel bed to such a point as to shut down the engine, hence the need of some form of automatic control.

Also the findings as to clinkers are apt to be misleading, due to the very small size of the producer in question.

The ideal conditions of draft current in a gas producer consist of a perfectly balanced draft at all loads. Any arrangement to get away from this balanced condition greatly increases the tendency to clinker in all kinds of fuels, for be it understood that the clinker is first formed by a fissure forming in the fuel bed, and by the concentration of draft through such a channel, with a blowpipe effect that fuses the ash and fuel, and that the only way to prevent clinkers is so to arrange the fuel bed that the tendency to form such fissures or "pipes" is reduced to a minimum. This is best accomplished by using fuel beds of generous dimensions and grates that insure equal distribution of the draft current to all parts of the fuel bed at all loads.

In practice it will be found that the resultant increase in efficiency due to the fixation of the tarry vapors of the gas, rather than washing them out and wasting them, is much less than would be expected. If such figures have been worked out by the author, the results would be of interest.

Personally I rather lean to the construction of bituminous producers along the line of simple single-zone up-draft, with attached tar washers, as being more adapted to the hard usage of practice, and have in mind a single 400-h.p. up-draft balanced-draft producer, operating on Illinois slack, costing 85 cents a ton delivered, said coal having 4 per cent sulphur content, 22 per cent ash, and 38 per cent volatile matter (10,300 B.t.u.). This producer was installed without any spare unit to help out, and has operated 24 hours per day since August, 1910, using 1½ lb. of coal per h.p-hour. I mention this plant as a case in point on the clinker question, as this fuel had a bad reputation in that respect (as well as others).

Also as to the reduction of CO_2 to 2 CO, my own experience indicates that perfect reduction occurs at 1800 deg. fahr., provided that the draft velocity is sufficiently low to allow the time necessary for the reaction. The faster the flow of draft through the fuel bed, the higher must the temperature be for this reason.

The author should continue his investigations under conditions entailing more commercial conditions, variable load, without special attention, and noting the possibility of keeping the gas tar-free during such variations without undue attention, and the effect of large grate areas; for example, in producers 10 ft. in inside diameter, for all producer builders have had the sad experience of discovering that the design that was most successful at 36 in. diameter was far from such on twice that diameter, etc.

EDWARD RATHBUN said that while the author's investigation appeared to be an excellent laboratory development, the actual gain in practical knowledge, as an aid to the present commercial operation of producers, was somewhat limited. He called attention to the author's statement that the gas was "almost as clean as anthracite gas," and said that in order to develop the producer-gas engine the gas must be clean, and that gas nearly as clean as anthracite was not clean enough.

He pointed out that in order to obtain a gas with a high heating value, the methanes must not be broken up in the producer. This was not possible in the author's producer. The result was that the capacity of an engine for a given bore and stroke was reduced, as well as the flexibility of operation, making a greater investment in the engine necessary where the heating value of the gas is low.

HARRY F. SMITH said that without question the complete oxidation of the hydrocarbon content of the coal to water vapor and carbon dioxide and the subsequent dissociation of these fixed gases and the carbon constituted an absolutely effective way of eliminating tar.

By means of an example he pointed out certain limitations in the process outlined in the author's paper. A West Virginia coal had the following percentage analysis:

Carbon	56.00
Volatile combustible	34.00
Water vapor	1.90
Ash	7.21

The complete analysis also gave the quantity of hydrocarbon, etc., and hence it was possible to determine that 0.22 lb. of carbon and 0.05 lb. of hydrogen were carried off in volatile from each pound of coal. It was calculated that 0.47 lb. of carbon was needed to reform carbon monoxide and hydrogen from the hydrogen and carbon originally burned out of the combustible in the gas. Therefore, with 0.56 lb. of carbon in the coal and with 0.47 lb. thus taken from it, 0.09 lb. was left, assuming that the distillation process was carried on with no gas drawn from the producer except the volatile combustible gases and 100 per cent efficiency in every step of the process. In this example there was no carbon left to decompose the water vapors, no additional gas for recirculation, and no contingency for the heat losses unavoidably associated with the process.

In other words, unless a coal contained a high percentage of fixed carbon, there was not enough fixed carbon to carry on the recirculating process continuously in a producer. If the producer were to run for several hours and then have a long enough stand-by period, sufficient coke could be formed during the latter period to cover the deficiency in carbon. One way to overcome the difficulty was to use coal with a high percentage of fixed carbon; another, to permit part of the products of combustion to pass out of the producer undecomposed, with a consequent loss in efficiency.

The objection to low heating value, mentioned by Mr. Rathbun, was of even more importance if the gas was used for industrial operations, such as brazing and forging. Since there were several methods available for cleaning producer gas, and the removal of the tar was so easy and so effective and the apparatus so simple, the justification of a more complex producer with the uncertainties of the process was questionable.

W. B. Chapman said that it was difficult to obtain a uniform gas from soft coal in the type of producer described by the author, or almost any type, unless it was almost entirely mechanically operated, and even then the gas could not be depended upon to be uniform to the same extent that steam could. In his experience, covering a period of twelve years, he had met with all the difficulties mentioned by the author and other more serious ones; he therefore doubted the possibility of solving the problem of obtaining uniform gas from soft coal, suitable for gas engines, along the lines outlined in the paper. The fuel bed of the author's producer was too deep for easy poking, a disadvantage in practical operation. Unless there was some means of controlling the gas when the engine was under a light load, the fire would grow cold because of the returning gases.

C. M. Garland (written). The studies recorded in the paper are very interesting and the methods of investigation, together with the design of the producer, are both novel and ingenious. There are several points brought up, however, which the writer feels have not been sufficiently described and on which further information would be very desirable.

Regarding the removal of ash and clinker from the producer: from the drawing it would seem that the burner for the recirculating gas takes up a large portion of the grate area. It would also seem that this burner would be in danger of clogging through an accumulation of ash. Regarding the formation of clinker: the combustion of the recirculating gases, while it would undoubtedly destroy the tar, would, however, produce a high temperature in the ash zone which would greatly facilitate the formation of clinker, even with fuels which would not clinker in the ordinary producer. Due to the recirculating of the gases and the high temperature, which are apparently maintained even in the coking chamber of the producer, the clinker formation might begin very high in the producer. In this connection temperatures through the fuel bed and of the gases leaving the producer would be very interesting. The removal of the gases from the side of the producer is another element that would tend to cause the formation of clinker, particularly around the side walls.

Data on the calorific value and the composition of the gas from the producer would also be very desirable. The design would unquestionably eliminate tar and produce a gas of low calorific value. While the gas of low calorific value, in so far as power is concerned, is not objectionable, the power end of gas-producer work today is a comparatively small end of the work and is more than likely to decrease rather than increase.

The ideal to be approached in the elimination of tar is a producer in which the tarry products are converted into fixed hydrocarbon gases which will raise the calorific value of the gas instead of lowering it. The demand today is for a producer gas to replace fuel oil. This gas must have a high calorific value in order that high furnace temperatures may be produced.

THE AUTHOR. Mr. Tait calls attention to the methods used in testing for tar. It is very important to know that the gas leaving this producer is entirely free from tar at all times. The method used in making this test is a very searching one and can be applied continuously and without difficulty in any power plant where either live steam or compressed air is available. It consists in maintaining a vacuum of about 20 in. of water inside of a large glass bottle by means of a steam jet or an aspirator. The bottle should have a wide mouth closed by a stopper. To this stopper is attached the support for a slip of white cardboard. The gas to be tested for tar is led into the bottle through a glass tube. This tube is drawn down to a small opening at its lower end and bent so as to cause the gas to impinge against the paper at right angles and at a high velocity. The presence of the slightest trace of tar in the gas will cause a brown spot to form on the cardboard immediately. This method is very easy to apply, and has the advantage of being very searching and at the same time very quick to react.

I feel that if Mr. Tait will make a closer study of the European producers he will find that they have not been worked out along the lines here presented, though some of them have been very similar. In some cases I have had a hard time to see for myself why the older producer should fail and my own be worthy of success. There has been a reason in each case, however, even though the search for it has caused some anxiety on my part in the early stages.

The work of O. Boudouard, published in the Comptes Rendus de l'Académie des Sciences in 1899 and 1900, has been widely referred to. It would seem to indicate that a temperature of 1800 deg. fahr. is sufficient to decompose the CO_2. The element of time has been left out in this work, however, as is shown by the later work done by J. K. Clement, L. H. Adams, and C. N. Haskins for the Department of the Interior, U. S. Bureau of Mines, and published in their Bulletin 7. Here it is shown that at a temperature of 1832 deg. fahr. the gas must be in contact with coke for 123.2 sec. in order to decompose 78.4 per cent of the CO_2. This is entirely prohibitive, as Mr. Tait will see.

I am sorry to be unable to present a detailed sketch showing Mr. Garland just how the gas burner avoids clogging up with ash. The removal of the ash and clinker is as easy to accomplish in this plant as it is in the standard anthracite plant, and the burner is so designed as to avoid any possibility of clogging up, and never has to be cleaned.

The burning of the recirculated gas tends to lower rather than raise the temperatures met with in the combustion zone of the producer, and has never been the cause of the formation of clinkers. The formation of clinker in the coking zones of the producer is unthinkable, as the temperatures there are below 1200 deg. fahr., while it is a poor coal having an ash with a fusing temperature as low as 2350 deg. fahr.; good coals will not clinker below 2750 deg. fahr.

The heat value of the gas from this plant seems to vary between 125 and 150 B.t.u. per cu. ft., the same as anthracite gas. The high heat values reported in connection with producer gas from bituminous fuel are due quite largely to the presence of tar vapors in this gas, and would be impossible without these vapors. Such a gas is ideal for some types of furnace work, but cannot be used in an engine.

Mr. Rathbun takes exception to the cleanliness of this gas. He knows that the gas from an anthracite plant is the cleanest gas that can be gotten from any type of gas producer, and is also the easiest to separate from what little dirt it does contain. The gas as it leaves this producer is in a class with anthracite gas. It is not clean enough to use in an engine in the condition in which it leaves the producer, but it is the easiest gas to clean that there is.

Mr. Smith is correct in his statement that it will take considerable fixed carbon in the coal to reduce all of the CO_2 and H_2O formed by the combustion of a large amount of volatile matter. It is probable that not all of these gases are completely burned. The tars can be completely destroyed by passing them through an incandescent zone, and some of them are probably eliminated in this way.

As a matter of fact, nearly all of the tests on this producer have been made with Indiana and Illinois coals having about 8 per cent moisture, 35 per cent volatile matter, 49 per cent fixed carbon, and 8 per cent ash. No trouble was experienced in keeping a sufficient bed of coke with any of these fuels.

variably means gas of high heating value and good producer efficiency. The determination of the carbon dioxide can be quickly, easily, and accurately carried out with a simple and inexpensive portable apparatus. The ordinary Orsat apparatus, or some modification of it, may be used. It is possible to use some form of automatic recording carbon-dioxide machine for this work, if the gas is properly cleaned. Such machines, however, are at the present time expensive, and require expert attention if they are to be kept constantly in working order. The chief objection to the more general use of carbon dioxide as an indicator of gas quality is the difficulty of collecting long-period samples over water, without a considerable change in carbon-dioxide content of the sample, due to the absorption or evolution of that gas by the water. This will be considered more fully in the discussion of sampling.

4 In some producer plants the percentage of carbon monoxide in the gas is used as the basis of a bonus system for producer operators. The carbon monoxide is not appreciably affected by the water of the sampling bottle, so that long-period continuous samples can be taken. Although it is true that the carbon monoxide furnishes the greater part of the heat in the gas, high carbon monoxide might be produced at the expense of hydrogen by changing the ratio of steam to air in the blast. If this is taken into account, or controlled, the carbon monoxide should give a good indication of the quality of the gas. To make rapid and accurate determinations of carbon monoxide, the apparatus for analysis should be equipped with two bubbling pipettes, each containing the regular cuprous-chloride solution which is used for this purpose. One of these solutions should be kept quite fresh, and should be used to absorb the last trace of carbon monoxide after the other has absorbed the greater part of it.

5 In the near future the standard method of determining the gas quality probably will be by some form of recording calorimeter. Such instruments have been in use for a number of years, and in many cases have proved very useful. In general, however, they require frequent adjustment if absolute rather than merely relative values are desired. This difficulty will doubtless be overcome in the course of time.

6 A rather crude but ideally simple method of estimating the quality of the gas is to use a constantly burning test flame. It is almost universal practice to provide such a flame for the benefit of the producer operator. By its use he can detect wide variations in the quality of the gas.

7 The purpose of the following discussions is to deal with the sampling and analysis of producer gas mainly from the practical standpoint. The aim is to describe methods which will give fairly accurate results in a simple and satisfactory manner.

SAMPLING

8 The sample of producer gas may be drawn directly from the gas main, or from a pipe through which there is a continuous flow of fresh gas from the main. In general, the latter method is to be preferred. The pipe from which the sample is drawn may well be that which supplies the test flame. For a continuous sample the pressure in the pipe should be practically constant at the point from which the sample is taken. A correct continuous sample is impossible if there is a variation of pressure where the sample leaves the pipe. Such a variation increases or decreases the flow of gas into the sampling bottle, even when the flow of water from the bottle is properly regulated. The sampling connection will be at practically atmospheric pressure if it is located on the pipe supplying the test flame, at a point beyond the valve which regulates the flame.

9 If there is no gas main with a pressure above atmospheric, some form of steam aspirator can be used to produce a steady flow of gas from the main. The rate of flow should be such that the gas in the aspirator pipe will always be substantially the same as that in the main. For a description of such an aspirator see Bulletin No. 97 of the Bureau of Mines, on Sampling and Analyzing Flue Gases.

10 The question of the proper form and position of the sampling pipe in the gas main has already received a great deal of attention. It is the opinion of the writer that, if the gas is sampled after it has passed through the scrubber, it is so thoroughly mixed that an open-end sampling tube at the center of the main will draw a representative sample. In the case of fuel-gas installations, where the gas is used without being washed, the mixture of the gas in the main is, of course, much less complete. At the same time, special sampling tubes with small holes would be particularly liable to become clogged, if used with unwashed gases, so that it would seem best to use the open-end sampling tube in such cases, placing it as far as possible from the producer to insure the maximum amount of mixing.

11 There are a number of methods which may be used for drawing the gas sample from the continuous-flow pipe described above, or from the main itself. Probably the one in most common

use is the so-called "two-bottle method." Two large bottles are connected by a rubber tube, several feet in length, attached to openings near the bases of the bottles. The bottle which is to collect the gas is fitted with a rubber stopper, through which passes a glass tube. This glass tube is connected to an outlet in the continuous-flow pipe by a short length of rubber tube. The bottles are partly filled with water, which acts as a piston to move gas into or out of the sampling bottle when the second bottle is lowered or raised. Before taking a sample, it is customary to draw in a certain amount of gas and shake it up with the water to saturate the latter, and thereby reduce its effect on the sample. The water is then forced to the top of the sampling bottle and preferably up to the point of attachment to the continuous-flow pipe. Next, the rubber tube connecting the bottles is clamped with a screw pinch cock, and the second bottle is placed some distance below the first. The pinch cock is then opened enough to give the desired rate of flow of the water from the sampling bottle.

12 It is well known that water has a great absorbing power for carbon dioxide, so that the carbon-dioxide content of the gas will be greatly lowered if it is left for any great length of time in contact with fresh water. If the water is first saturated with gas of the same composition as that to be sampled, this absorption is prevented. If the sampling period is extended and the carbon-dioxide content of the gas becomes greater or less than that of the gas with which the water has been saturated, there is an absorption or evolution of the carbon dioxide by the water. This interchange of carbon dioxide between the gas and the water can be reduced, to a certain extent, by using brine or dilute acid in place of ordinary water. This whole matter is treated at length in Bulletin No. 97 of the Bureau of Mines, to which reference has already been made. If mercury is used as the displacing liquid the absorption of carbon dioxide is entirely prevented, but this practice is too expensive for ordinary commercial use.

13 When continuous samples extending over long periods are taken, using the ordinary two-bottle method, a number of errors occur aside from that due to the absorption and evolution of carbon dioxide by the water. It is evident that a correctly drawn sample should have the same composition as that which would be obtained by collecting in a gas holder and thoroughly mixing all of the gas which passes through the main in the given period. To get this result, when the flow of gas in the main is variable, the sample must

be drawn continuously at a rate proportional to the rate of flow of the gas in the main. If the rate of sampling is not proportional to the flow of gas in the main, and if, at the same time, there are large variations in the gas composition, a considerable error may result. In cases where the gas composition does not vary a great deal, there can be a slight amount of deviation from true proportional sampling without serious error. The following examples give a general idea of the magnitude of this error under various conditions.

14 In the first case, let us assume that there is a uniform flow of gas through the main, but that there is a large variation in the rate of sampling. Suppose that during the first half of the period the gas contains 20 per cent of carbon monoxide and the amount of sample drawn is 2000 cu. cm., while the corresponding amounts for the second half are 30 per cent and 1000 cu. cm. A correctly drawn sample would contain 25 per cent of carbon monoxide. The actual sample has a total volume of 3000 cu. cm. and contains 700 cu. cm. of carbon monoxide ($0.20 \times 2000 + 0.30 \times 1000$), or 23.3 per cent.

15 A similar error occurs if the gas is sampled at a uniform rate while there is a large variation in the composition of the gas and its rate of flow through the main. Suppose that during the first half of the sampling period there passes through the main 20,000 cu. ft. of gas, containing 20 per cent of carbon monoxide, and that the corresponding figures for the second half are 10,000 cu. ft. and 30 per cent. The actual sample, consisting of equal parts of the 20 per cent gas and the 30 per cent gas, contains 25 per cent of carbon monoxide. The total volume of producer gas made during the period is 30,000 cu. ft., and the total amount of carbon monoxide is 7000 cu. ft. ($0.20 \times 20{,}000 + 0.30 \times 10{,}000$), or 23.3 per cent.

16 If the changes in gas composition and the deviations from proportional sampling are not extreme, the error is not serious. Suppose that the gas flow through the main is constant and that the volumes of sample for the first and second halves of the sampling period are 1000 cu. cm. and 800 cu. cm., respectively, while the corresponding per cents of carbon monoxide are 22 and 26. In this case, the actual sample contains 23.8 per cent of carbon monoxide, while the correct amount is 24 per cent.

17 In cases where both the rate of gas flow and the rate of sampling vary at random, the errors may add up or partly neutralise each other.

18 When a continuous sample is drawn by the ordinary two-bottle method, the aim is to sample at a uniform rate. Where the demand on the producers is fairly constant, and the rate of sampling does not vary more than 5 or 10 per cent, the sample obtained is very nearly correct (except for the possible error in the carbon dioxide). There are two reasons for the varying flow of water, and hence the varying rate of sampling. In the first case, the opening in the compressed rubber tube between the bottles is such that it has a great tendency to clog. This is particularly true when the tube is throttled down for a long-period sample. In the second case, the net head producing the flow of water, which is the difference in level in the water surfaces of the two bottles, is constantly decreasing as the water flows from the upper to the lower bottle. The per cent change in net head can be kept small by having the sampling bottle placed several feet above the other bottle. It can be reduced one-half by disconnecting the tube from the bottom of the lower bottle and allowing the water to drip freely from the end of the tube.

19 It is evident that the ordinary two-bottle method never gives an exactly correct sample, and that clogging of the thin tapering opening in the rubber tube may produce a large error without being detected.

20 A device which the writer has designed for the easy measurement and control of the rate of sampling is shown in Fig. 1. The connections of the upper bottle are the same as in the ordinary two-bottle method, but the water passes into the top of the lower bottle through a simple form of orifice flow meter. This meter is constructed largely of $\frac{1}{8}$-in. brass pipe fittings. The general construction of this device is evident from the illustration. A thin copper disk, having a suitable small orifice, is clamped between the two halves of the union, so that the head of water which is producing the flow through this orifice can be measured on the manometer, shown at the left of the union. There is a small opening in the top of the elbow, shown on the right, to prevent the nipple below from acting as a tail pipe to increase the head on the orifice. The water in the manometer can easily be brought to any desired point by adjusting the pinch cock on the rubber tube. The orifice, being made in a very thin plate, gives little trouble from clogging if clean water is used. If desired, however, a small cloth filter can be inserted in the nipple at the left of the tee. The iron fittings, shown in the neck of the lower bottle, serve merely as a support, and may

be replaced by a stopper with two holes, one to admit the ⅛-in. nipple, and the other to act as a vent. A series of orifices for various rates of flow may be prepared by piercing disks of copper foil with needles and smoothing off the rough edges on an oil stone.

21 For sampling at a uniform rate, the water in the manometer tube can be brought to any mark and kept there by an occasional adjustment of the pinch cock. If it is desired to take a continuous

FIG. 1 APPARATUS FOR PROPORTIONAL SAMPLING

proportional sample, it is necessary to have some form of flow meter in the gas main and a scale of rate of flow on the manometer. Since it is only necessary that the rate of flow of the water be *proportional* to the rate of flow of the gas in the main, it is immaterial what units of flow are used. Four scales of flow are used by the writer. These are so chosen that it is usually possible to pick one whose graduations (without regard to the decimal point) more than cover the probable range of readings of the flow meter on the gas main. These scales were laid out according to the law that the flow is proportional to the square root of the head. The upper marks on the four scales are 25, 50, 100 and 200. Let N be the top mark on any scale, and H the corresponding head. Let n be any other number on the scale and h the corresponding head. Then the scale can be laid out from the formula $h = H\,(n/N)^2$. By making H 15 in. or more, satisfactory scales are obtained.

22 A test was made to determine the accuracy of this flow meter, the rate of flow being determined by catching the water in a graduated tube and noting the time required to accumulate a given amount of water. Readings were taken on the 100 scale at the points, 20, 30, 40, 50, 60, 70, 80, 90 and 100. It was found that the quotient of the rate of flow by the scale reading was practically constant, with a maximum variation of 2 per cent from the mean. Below 20, the divisions are so close to the zero position that accurate reading is difficult. It is always possible, however, unless there are very extreme variations in the flow of gas through the main, to find a scale such that the lowest setting required will come at a reading which is more than one-fifth of the highest reading on the scale.

GAS ANALYSIS

23 Highly accurate analyses are carried out in laboratories by chemists, and the writer will not go into a description of their methods. In general, the engineer who is testing or operating gas producers desires to use a portable machine which is rapid, reliable and easy to operate, and which will give each gas constituent correctly within 0.2 or 0.3 per cent of the total volume of gas.

24 In the following discussion, it will be assumed that the reader is familiar with the operation of the ordinary forms of Orsat apparatus such as are used in the analysis of furnace-flue gases for carbon dioxide, oxygen and carbon monoxide. The pipettes for absorbing these gases usually contain the following reagents: for carbon dioxide, either caustic potash or caustic soda; for oxygen, either an alkaline pyrogallol solution or sticks of phosphorus (in water); and for carbon monoxide, either an acid or an ammoniacal solution of cuprous chloride. The preparation of these reagents is described in books on power-plant testing, but, unless a chemical laboratory is available, many engineers will prefer to buy them put up in bottles ready for use.

25 The absorption of carbon dioxide by either the caustic-potash or caustic-soda solution is rapid, and the solutions have a large absorb-

ing power, so that frequent changes are not necessary. The alkaline pyrogallol solution for oxygen is slower in its action, and has a smaller absorbing power than the solutions for carbon dioxide. The solution must be carefully protected from air by a rubber bulb. Another disadvantage of this solution is that it tends to give off carbon monoxide if it is old or improperly prepared. Phosphorus sticks ordinarily make an ideal reagent for the absorption of oxygen. The presence of 0.1 cu. cm. of oxygen will produce fumes that are easily visible, and if the reaction is once started, it is always complete in a minute or two if there is a large exposed surface of yellow phosphorus. The presence of a small amount of ethylene completely prevents the reaction between phosphorus and oxygen. On this account phosphorus should not be used for producer gas that contains ethylene, unless the ethylene can be absorbed before the oxygen. If the slightest fumes are visible it is known that the phosphorus is acting properly, and almost any producer gas will contain enough oxygen to produce visible fumes. The sticks gradually turn dark and become inactive if exposed to the sunlight, so that the pipette should be protected by wrapping it with dark paper. With this precaution, one set of phosphorus sticks can be used for years without any attention. The carbon monoxide is the least satisfactory to absorb of the three gases. The cuprous-chloride solution is slow in its action and has a small absorbing power. The solution is unreliable unless it is quite fresh, for there is no way of telling when the absorption is complete. It is impossible to absorb all of the carbon monoxide with a solution which has already taken up a considerable amount of the gas. In the case of producer gas, a fairly complete absorption of the carbon monoxide can be made by using two pipettes. The second pipette is used only to take out the last part of the carbon monoxide after the first has absorbed most of it. In this way the second solution can be kept fresh for a number of analyses.

26 Some of the newer types of Orsat apparatus are arranged so that the gases can be bubbled through the reagents. This greatly reduces the time required for absorption of all the gases, but of the carbon monoxide in particular. In the case of phosphorus, nothing is gained by the use of a bubbling pipette, as the reagent is not a solution but a solid.

27 For the complete analysis of producer gas, the ordinary Orsat apparatus is not sufficient; provision must be made for the determination of hydrogen, methane and illuminants (chiefly ethylene). There are now on the market for this purpose a number of portable instru-

ments similar to the Orsat but with certain modifications and additions. The Williams apparatus, shown in Fig. 2, is a good example of such an instrument. Bubbling pipettes are used for all of the liquid reagents. Going from left to right, the pipettes shown in Fig. 2 contain the following reagents: *first*, caustic-soda solution for carbon dioxide; *second*, bromine water for illuminants; *third*, phosphorus sticks for oxygen, and *fourth*, an acid cuprous-chloride solution (containing copper strips) for carbon monoxide. Since there are only four pipettes, a second cuprous-chloride pipette could only be obtained by omitting the bromine water for illuminants. If this is

FIG. 2 WILLIAMS APPARATUS FOR PRODUCER-GAS ANALYSIS

done, a pyrogallol solution instead of phosphorus should be used for the oxygen, because the presence of a little ethylene in the gas might prevent the phosphorus from reacting with the oxygen.

28 In the ordinary method of operation, the carbon dioxide, illuminants, oxygen and carbon monoxide are absorbed in this order, and the hydrogen and methane are then determined by explosion with air in the explosion burette. It is necessary to use a slight excess of air and to measure the contraction on explosion and the carbon dioxide produced by the explosion. The latter is determined by absorption in the first pipette. If the full amount of gas left after

the absorption of carbon monoxide is used for the explosion, the following equations are employed.

Methane = Carbon dioxide produced
Hydrogen = $\frac{2}{3}$ contraction − $\frac{4}{3}$ carbon dioxide produced

29 The method just described is the usual one for the analysis of producer gas with a portable apparatus, but it has certain disadvantages as follows:

a On account of the large amount of carbon monoxide in producer gas, the cuprous-chloride solution must be changed very often, with consequent trouble and expense.

b As the amount of methane present is small, the hydrogen content of the gas may be so low that the gas will not explode with air. In this case it becomes impossible to determine the hydrogen and methane by the ordinary method.

30 It is possible to eliminate the absorption of the carbon monoxide and to determine the carbon monoxide, hydrogen and methane by one explosion. This method does away with the troublesome cuprous-chloride solution and insures an explosion if the gas is combustible. This, however, requires the determination of the oxygen consumed as well as the contraction and the carbon dioxide produced. The absorption of the oxygen left over from the explosion is quicker than the absorption of carbon monoxide. On the other hand, the computations required are somewhat more extensive, but this work can be greatly simplified, as will be shown later.

31 The burette supplied with the Williams machine is graduated, as is that of an ordinary Orsat, with the zero at the point where the volume is 100 cu. cm. and the 100 mark at the top. This is the best arrangement where only absorption determinations are made, but is somewhat confusing when making a complete analysis. The instrument shown in Fig. 2 has been provided with another scale on the left which gives the volume directly. Moreover, the graduations have been extended nearly to the bottom of the burette, so that volumes up to 115 cu. cm. may be measured. This provides for the increased amount of air which must be used when the carbon monoxide is left in the gas which is to be exploded.

32 In the determination of carbon monoxide, hydrogen and methane by one explosion, it is necessary to measure the following three quantities:

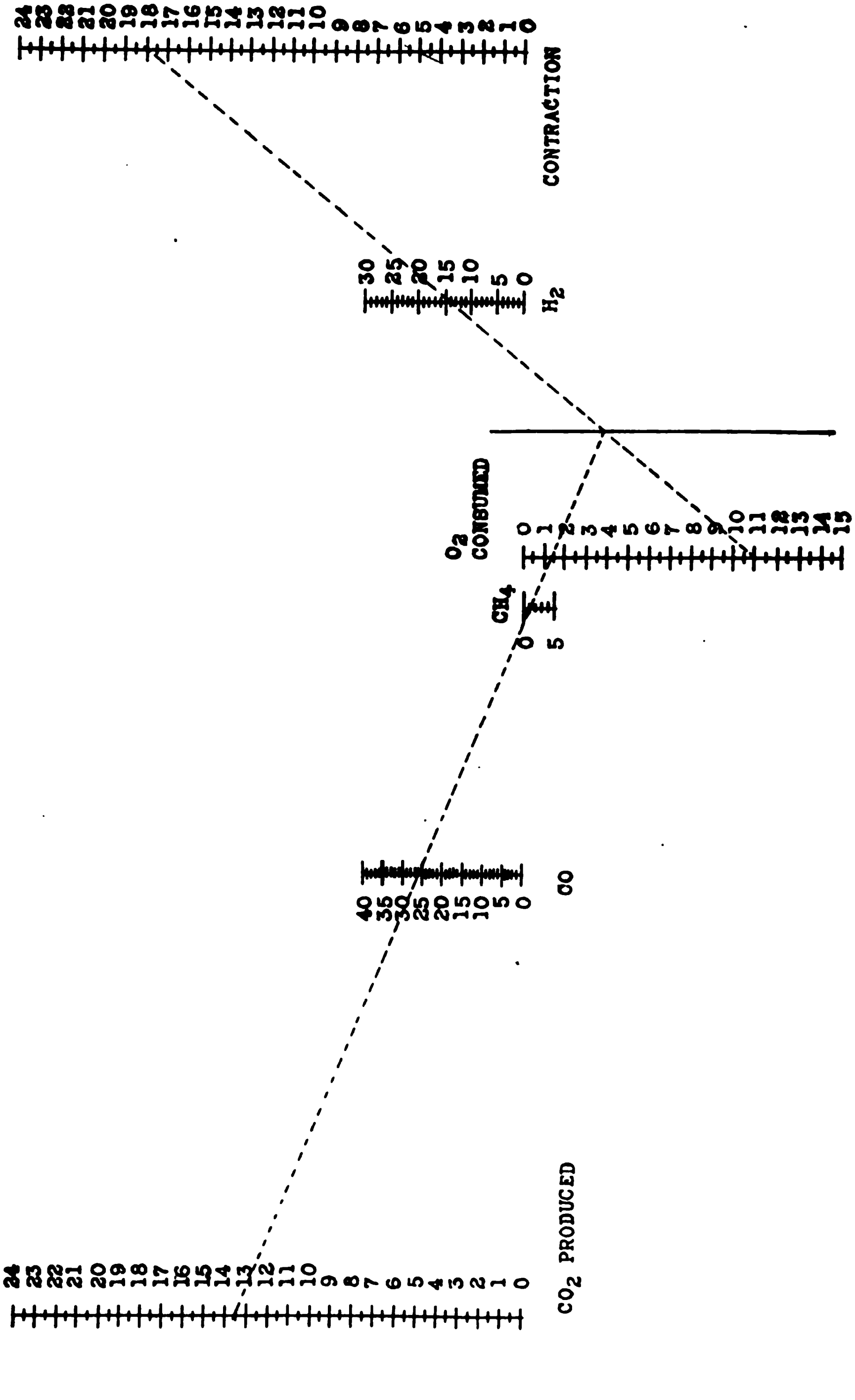

FIG. 3 DIAGRAM FOR DETERMINING HYDROGEN, CARBON MONOXIDE, AND METHANE

a The contraction due to the explosion
b The carbon dioxide produced by the explosion
c The oxygen consumed in the explosion.

The first two are determined just as in the ordinary method where only the hydrogen and the methane are exploded. To determine the quantity of oxygen consumed, it is necessary to measure the amount of air taken in and to measure the excess of oxygen by means of the phosphorus pipette. For 100 cu. cm. of ordinary producer gas it is sufficient to use 124 cu. cm. of air (26.0 cu. cm. oxygen). With this amount of air the explosive mixture would have a total volume of more than 200 cu. cm. and could not be contained in the burette, so that it is necessary to reduce the volume of gas before explosion. For simplicity of calculation it is well to throw away exactly one-half of the volume of gas remaining after the absorption of the carbon dioxide, illuminants and oxygen, and then to take in 62 cu. cm. of air containing 13.0 cu. cm. of oxygen. If one-half of the gas is used for explosion, the following equations give the per cent of hydrogen, carbon monoxide and methane in the original gas:

Hydrogen = 2 (contraction − oxygen consumed)
Carbon monoxide = 2 ($\frac{1}{3}$ contraction + $\frac{2}{3}$ carbon dioxide produced − oxygen consumed)
Methane = 2 (oxygen consumed − $\frac{1}{3}$ contraction − $\frac{1}{3}$ carbon dioxide produced)

33 Many will prefer to solve these simple equations by computation with or without the use of a slide rule, but for routine analyses time may be saved by the use of a diagram like that shown in Fig. 3. The dotted lines in Fig. 3 show the methods of using the diagram, but in ordinary solutions no lines need be drawn if use is made of a celluloid straight edge and a fine needle provided with a handle. The manipulation might be as follows:

a Place needle at contraction
b Swing straight edge around needle to the oxygen consumed
c Read hydrogen
d Place needle at intersection of straight edge and blank
e Swing straight edge around needle to carbon dioxide produced
f Read carbon monoxide
g Read methane.

This method of reading the diagram is rapid and accurate. More accurate readings can be made if a larger diagram is used.

34 The data and results of the analysis may be conveniently recorded on a form similar to the following:,

a	Take 100 cu. cm. of gas, absorb carbon dioxide......	reading[1]	....
b	.... absorb illuminants.........	reading	....
c	.... absorb oxygen..............	reading	....
d	Throw away one-half of remaining volume............	reading	....
e	Take in 62 cu. cm. of air (13 cu. cm. of oxygen).......	reading	....
f	Explode..	reading	....
g	Absorb carbon dioxide produced......................	reading	....
h	Absorb oxygen left over............................	reading	....
i	Contraction due to explosion........................	$(e-f)$	....
j	Carbon dioxide produced by explosion................	$(f-g)$	....
k	Oxygen consumed in explosion.....................	$[13.0-(g-h)]$	....
	Carbon dioxide............	$(100-a)$	....
	Illuminants...............	$(a-b)$	....
	Oxygen....................	$(b-c)$	....
	Hydrogen..................	(diagram)	....
	Carbon monoxide...........	(diagram)	....
	Methane...................	(diagram)	....

[1] Readings on true volume scale.

35 It has already been stated that phosphorus is inactive in the presence of illuminants. Producer gas is often free from illuminants so that time may be saved by passing directly to the phosphorus pipette after the carbon dioxide has been determined. If any fumes appear, the absorption of oxygen can be continued. In this case, the same reading is recorded at *b* as appears at *a*. If fumes do not appear, the gas is passed into the pipette containing bromine water, and then into the carbon-dioxide pipette to absorb the bromine fumes. The action is rapid, so that one pass through each pipette is sufficient.

36 In case the analysis is made with the explosion of hydrogen and carbon monoxide only, the following form may be used for the data and results:

a	Take 100 cu. cm. of gas, absorb carbon dioxide......	reading	....
b	.... absorb illuminants.........	reading	....
c	.... absorb oxygen..............	reading	....
d	.... absorb carbon monoxide....	reading	....
e	Throw away one-half of remaining volume...........	reading	....
f	Take in an excess of air............................	reading	....
g	Explode..	reading	....
h	Absorb carbon dioxide produced.....................	reading	....
i	Contraction due to explosion........................	$(f-g)$	....
j	Carbon dioxide produced by explosion................	$(g-h)$	....
	Carbon dioxide............	$(100-a)$	....
	Illuminants...............	$(a-b)$	....
	Oxygen....................	$(b-c)$	....
	Carbon monoxide...........	$(c-d)$	....
	Hydrogen..................	$(\frac{4}{3}i-\frac{8}{3}j)$	....
	Methane...................	$(2j)$	....

37 Fig. 4 shows an ordinary Orsat apparatus which has been modified for the analysis of producer gas. Going from right to left, the first three pipettes contain caustic-potash solution, phosphorus and bromine water. The fourth pipette is arranged for explosion by means of a heated platinum wire. The use of a heated wire instead of a spark gap eliminates the induction coil so that the apparatus is self-contained. The copper leads pass down through a glass tube.

FIG. 4 ORSAT APPARATUS WITH EXPLOSION PIPETTE

This tube is provided with an explosion-tight joint but can be easily removed or replaced if necessary. The results obtained are fairly accurate but, on account of the type of burette and pipettes employed, more time is required for an analysis than is needed with instruments specially designed for the analysis of producer gas.

DISCUSSION

W. E. REULING[1] (written). The subject of gas analysis as applied to producer-gas plants is a very interesting one and worthy

[1] Michigan Agricultural College, East Lansing, Mich.

of more consideration than many owners and operators give it. This lack of consideration or interest doubtless comes from two causes: first, failure to appreciate fully the fact that a gas analysis, when properly interpreted, furnishes a splendid check on the efficiency of the plant for the conditions under which it is being worked; and second, from a rather inherent feeling that the subject of gas analysis is a little too technical or savors too much of the chemical laboratory to be really a friend and working tool for the practical every-day engineer, owner, or operator.

It has not been many years since a similar aversion or lack of appreciation was very common in the matter of checking steam-plant performance by CO_2 determinations in the flue gas, and by indicator cards taken from the engine. However, through the educational advertising of firms manufacturing equipment for indicating steam-plant performance, and by the splendid work of the trade papers, this condition has been greatly changed. Owners and operators are appreciating more and more that the personal equation exerts a large influence as to what portion of the coal pile goes into useful work and what goes into thin air.

In gas-producer operation the personal equation can affect results possibly to even a greater extent than in a steam plant. I have known operators who maintained a thin fire, thinner than was intended by the manufacturer of the producer and far too thin for the completion of the proper reactions within the producer. They would also run with holes in the fire that allowed free air to pass, burning some of the gas on top of the fuel column. The result, of course, was that the gas went to the engine or furnace in an impoverished condition, carrying too high a percentage of CO_2. In one plant with which I have had experience, where engines totaling 3000 h.p. are operated on producer gas, there were times when quite an excessive amount of steam was used, with consequent detrimental effect on the quality of the gas. (In this plant, steam — taken from boilers used for pumps and heating — was at hand in unlimited amount so far as producer needs were concerned.) The fireman used it freely, his chief consideration being to keep the fire in such shape that the poker would drop through easily. In cases like these two I have mentioned, there is no doubt but that large savings could be effected by a better control of the gas quality — first determining by analysis the best steps to take for the conditions existing at the particular plant.

Harry F. Smith said that the author had made statements which meant something to the man who had had experience in the practical side of this work. However, he wanted to caution against the general statement that a good idea of the quality of the gas might usually be obtained from the carbon dioxide alone, to emphasize which he read from a number of analyses made by the same chemist, in the same plant and by the same method, on gas from the same coal. The extreme variation in the carbon dioxide content, percentage of combustible, and heating value gave evidence to show that it was not always possible to depend on the carbon dioxide indication alone.

Sanford A. Moss said he had been told that in sampling gas to determine the amount of dirt it carried, it was found absolutely necessary to have the velocity in the sampling tube exactly the same as the velocity in the pipes, and asked if this was necessary in producer-gas sampling.

The Author, referring to a statement made by Mr. Smith, said that there was considerable difference of opinion among men with wide experience in regard to carbon dioxide as an index of gas quality, some being so certain that it was an indication of the quality that only carbon-dioxide determinations were made by them.

In answer to Dr. Moss's question, he stated that it was quite important to have the velocity of the gas in the sampling tube the same as that in the main pipe if the dust and suspended matter were to be determined; but if not, there seemed to be no necessity for sampling at the exact velocity of the gas in the main. The idea was to take a sample continuously, and to take it at all times in proportion to the rate of flow of gas through the main, so that the final sample would be the same as though all the gas had been delivered into a gas holder and thoroughly mixed before sampling.

No. 1562

THE RATIO OF THE SPECIFIC HEATS AND THE COEFFICIENT OF VISCOSITY OF NATURAL GAS FROM TYPICAL FIELDS

By Robert F. Earhart,[1] Columbus, Ohio
Non-Member

The present paper may be considered an extension of a previous paper on the physical properties of natural gas.[2] The object of the study with which it deals was to secure measurements on the ratio of the specific heats and the coefficient of viscosity of products similar to those previously studied in the Boyle's Law tests.

2 A knowledge of the ratio of the specific heats is necessary in computations involving adiabatic changes represented analytically by the well-known formula PV^n = constant, where n = ratio of the specific heats. Physical tables give values of this constant for all of the common gases. The following values, taken from the tables prepared by the French Academy of Sciences in 1913, are for gases frequently found in our natural-gas products. The results apply to pure gases in a dry condition.

Air..............	1.405	Methane.........	1.318
Nitrogen.........	1.410	Carbon Dioxide...	1.300
Carbon Monoxide..	1.400	Ethane..........	1.182
Oxygen..........	1.398		

3 The composition of natural gas, while fairly constant for any field, varies greatly for different regions. The value for n used by engineers is usually derived from the data supplied by indicator cards of gas compressors. Various assumptions in securing these values

[1] Professor of Physics, Ohio State University.

[2] Deviation of Natural Gas from Boyle's Law, by R. F. Earhart and S. S. Wyer, p. 285 *ante*.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

[illegible]

[illegible]

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- [illegible]
- [illegible]

[illegible]

[illegible]

[illegible]

[illegible]

7 In order to compare the value of n for a gas with the value of n for air, we have to compare the velocity of sound in the two media under the same pressure conditions and to know the relative density of the gas to air.

Letting N = ratio of the specific heats for air
n = ratio of the specific heats for gas
D = density of air
d = density of gas
P = pressure
V = velocity of sound in air
v = velocity of sound in gas

we have the following ratio:

$$\frac{V^2}{v^2} = \frac{N\frac{P}{D}}{n\frac{p}{d}}$$

If the velocity be determined in the two media at the same pressure, then $n = N\frac{v^2}{V^2}\frac{d}{D}$, or $n = N\frac{v^2}{V^2}d$ (where the density of the air equals unity).

8 The experimental work resolves itself into comparing the density of gas with air at the same pressure and temperature and in comparing the velocity of sound in gas with that of air for this pressure. The latter comparisons were made by the usual method employed in laboratory investigations — that of Kundt, in which air or gas is set into longitudinal vibration inside a closed glass tube, and the nodal points of the resulting stationary waves are indicated by the disposition of lycopodium powder previously introduced. The distance between nodes equals a half wave length. In a 5-ft. tube 10 or 12 nodes may be distinctly observed, and these may be measured to within an accuracy of 1 mm. The relation between velocity V, frequency m, and wave length l is $V = lm$. Also, provided m is not altered by temperature or by displacements of apparatus, $\frac{V}{l_{air}} = m = \frac{v}{l_{gas}}$, which, substituted in the final equation of Par. 7, gives

$$n = \frac{N(l_{gas})^2}{(l_{air})^2}d$$

9 The following data from the record illustrate the application and indicate the magnitude of the quantities involved. The value

of N for air is taken as 1.405. The results here computed appear again in Table 1.

May 4, 1911.
Gas from the Geo. Secest farm, Vinton County, Ohio.
Tank filled at 200 lb. gage pressure.
Temperature, 20 deg. cent. Barometric pressure, 29.25 in.
To determine density,

Weight of glass bulb exhausted = 146.370 grams
Weight of glass bulb filled with gas = 146.750 grams
Weight of glass bulb filled with air = 146.946 grams
Weight of gas = 146.750 − 146.370 = 0.380 gram
Weight of air = 146.946 − 146.370 = 0.576 gram
Relative density = 0.380/0.576 = 0.666

DETERMINATION OF WAVE LENGTHS

Tube Contains Gas		Tube Contains Air		Tube Contains Gas	
Nodal points on meter stick	Half wave length	Nodal points on meter stick	Half wave length	Nodal points on meter stick	Half wave length
77.4 cm.	8.5 cm.	85.6 cm.	7.3 cm.	84.6 cm.	8.6 cm.
86.9	8.6	77.7	7.3	76.0	8.5
60.3	8.6	70.4	7.4	67.5	8.5
51.7	8.2	63.0	7.4	59.0	8.5
42.5	8.5	55.6	7.4	50.5	8.6
34.0	8.6	48.2	7.3	41.9	8.5
25.4	8.5	40.9	7.2	33.4	8.6
16.9	8.5	35.5	7.3	24.8	8.5
8.4	8.4	26.2	7.5	16.3	8.5
0.0	Av. 8.49	18.7	7.3	7.8	Av. 8.53
		11.4	7.2		
		4.2	Av. 7.34		

$$\text{Half wave length in gas} = \frac{8.49 + 8.53}{2} = 8.51 \text{ cm.}$$

$$n = 1.405 \times \left(\frac{(17.02)^2}{(14.68)^2}\right) \times 0.666 = 1.259.$$

10 In obtaining the data just given, gas was taken directly from the sampling tank without drying or processing of any kind. An experiment was made upon one sample both in the raw state and after passing through a calcium-chloride tube to remove water vapor. The density and wave-length measurements were modified by this, but when combined in the formula for determining n the

result was changed by less than one-half of one per cent, which is about the limit of accuracy of the experiment. The value of n found for the raw gas was 1.259; for the dried gas it was 1.257.

DETERMINATION OF THE COEFFICIENT OF VISCOSITY

11 The coefficient of viscosity is used in computations involving the flow of gas through pipes of considerable length. Two methods are commonly employed for determining this constant. In one a spherical mass is suspended by a torsion wire in the gas, given an angular displacement, and allowed to vibrate. The decrease in amplitude of successive oscillations is a function of the gas viscosity. The coefficient of viscosity may be computed from the logarithmic decrement of the vibrations. In the other method gas is slowly forced through a long capillary tube. The rate of flow is a function of the coefficient of viscosity and is determined by means of a formula. The two methods show excellent agreement for gases. The method used in this investigation was the capillary-tube, or transpiration, method, and the apparatus is very similar to the effluxometer used in comparing the density of gases. Instead of having the gas issue from a small orifice in a thin plate, it passes through a long tube. The formula[1] is

$$V_a = \frac{(p_a^2 - p_e^2) R^4}{16 C L p_e}$$

where

V_a = volume of gas passing through the capillary per unit time

p_a and p_e = pressure of gas before and after escape

R = radius of capillary

C = coefficient of viscosity

L = length of tube.

12 A consideration of this formula shows that if equal volumes of two gases under similar pressure and temperature conditions are passed through a long capillary tube, the coefficients of viscosity are in the ratio of the times of efflux. The value of the coefficient of air has been carefully studied during the past ten years on account of its importance in certain electrical measurements. Calling C_g (C_a) the coefficient of viscosity of the gas (air) and t_g (t_a) the time of transportation for the gas (air),

$$C_g = C_a \frac{t_g}{t_a}.$$

[1] For derivation, see Wüllner, Lehrbuch der Physik, vol. 1, p. 629.

TABLE 1 DATA ON NATURAL GAS FROM TEN U. S. FIELDS

Sample	Chemical Analysis					B.T.U. per Cu. Ft.[1] (at 32 deg. fahr. and 76 cm.)	Density (Air=1.0)	Ratio of Specific Heats, s,	Relative Coefficient of Viscosity	Coefficient of Viscosity in C.G.S. Units
	CO_2	O_2	CH_4	C_2H_6	N_2					
Air	....	....	....	....	....	....	1.000	1.405	1.000	178×10^{-6}
West Virginia, Ravenswood	0.0	0.0	82.0	17.5	0.5	1300	0.682	1.238	0.770	137×10^{-6}
West Virginia Connecting Gas Co.	0.0	0.0	86.1	12.1	1.8	1143	0.660	1.214	0.730	130×10^{-6}
Ohio, Sugar Grove Field	0.0	0.0	82.4	6.2	11.4	992	0.660	1.209	0.750	133×10^{-6}
Ohio, Homer Field, Licking County	0.0	0.0	78.8	14.2	7.2	1099	0.660	1.210	0.735	131×10^{-6}
Ohio, Vinton County Field	0.1	0.8	73.0	5.8	20.3	885	0.666	1.259	0.750	133×10^{-6}
Penna., Trafford City	0.2	0.0	95.0	0.0	4.9	1012	0.585	1.285	0.710	126×10^{-6}
Penna. Wet Gas, Ludlow East Branch Intake	0.0	0.0	66.4	34.6	0.0	1351	0.755	1.293	0.760	$135\times10^{-}$
Penna. Dry Gas Royston, Penna.	0.1	0.0	74.4	21.8	3.7	1198	0.678	1.220	0.750	$133\times10^{-}$
Texas, Beatty County	0.2	0.0	55.4	8.0	36.4	739	0.755	1.290	0.820	146×10^{-6}
Texas, Lone Star	0.1	0.0	50.0	10.0	39.9	718	0.770	1.224	0.830	147×10^{-6}

[1] B.T.U. calculated on Bureau of Mines values for CH_4 (= 1065) and C_2H_6 (= 1861).

13 It was customary to make three measurements with dry air, then a series of three measurements on the gas, to be followed by one or two additional measurements on air. A typical series is here introduced.

Air	Gas	Air
2 min. 20.8 sec.	1 min. 47.5 sec.	2 min. 21.0 sec.
2 min. 21.4 sec.	1 min. 47.3 sec.	2 min. 21.0 sec.
2 min. 21.0 sec.	1 min. 47.0 sec.	

Average for gas = 107.3 sec.
Average for air = 141.0 sec.

$$C_g = C_a \frac{107.3}{141.0} = 0.75\, C_a.$$

The volume of gas whose time of efflux is noted was about 300 cc.

14 In Table 1 are collected the data secured throughout the tests. The gas in each case was taken directly from the tank without drying or processing of any kind.

15 The chemical analyses were made in the Mines Laboratory of the Ohio State University by Mr. E. C. Smith. The author is indebted to a number of members of the engineering fraternity through whose coöperation the samples from the different fields were secured. A small grant from the Experiment Station of the Ohio State University defrayed the expense incident to the experimental work.

SUMMARY

16 To summarize:

a Determinations of the exponent n in the expression for adiabatic change PV^n = constant have been made on ten samples of natural gas. The lowest value obtained was 1.209, the highest 1.293. The maximum variation from the mean value (1.243) is 4 per cent.

b Determinations of the coefficient of viscosity give values ranging from 126×10^{-6} to 147×10^{-6} C.G.S. units.

APPENDIX

RELATION BETWEEN THE VELOCITY V OF A COMPRESSIONAL WAVE AND THE ELASTICITY E AND DENSITY d OF THE MEDIUM THROUGH WHICH IT PASSES

I In order to simplify the mathematical analysis we will consider a tube open at each end and having a cross-section of 1 sq. cm. Such a tube we conceive provided with a movable piston P. Let the pressure exerted by the medium be p. Consider a force applied to the piston $= p + dp$. The piston will now move in compressing the gas in front of it until the pressure becomes $p + dp$. No further compression will result, but this pressure will be transmitted to the next layer, which will in turn be compressed. During the compression of any layer, all layers previously compressed will move forward along the tube through a distance by which the layer under consideration is compressed. Assume that when the pressure was increased from p to $p + dp$ the diminution in volume of 1 cc. of the medium was dv. Calling V the velocity of the compressional disturbance, V cc. will be compressed per second, since the area of the tube is 1 sq. cm. The piston will move in at the rate Vdv cm. per sec. Now, since a layer of the medium merely serves to transmit the pressure, V cc. of the medium will be set in motion, with a velocity Vdv cm. per sec. Calling the density of the medium d_1, the mass moved per second will be Vd_1 and the kinetic energy which the medium acquires per sec. $= \frac{1}{2} Vd_1(Vdv)^2$. The amount of work done in compressing the medium in one second is equal to the average pressure multiplied by the compression produced.

Initial pressure $= p$
Final pressure $= p + dp$
Average pressure $= p + \frac{1}{2} dp$.

$\therefore$ Work done each second in compression $= (p + \frac{1}{2} dp)Vdv$. Again, the work done by the agent in one second is the force times the distance moved $= (p + dp)Vdv$. We can now equate the work performed by the agent to the kinetic energy communicated to the medium plus the work done in compression, as follows:

$$(p + dp)Vdv = \frac{1}{2} Vd_1(Vdv)^2 + (p + \frac{1}{2} dp)Vdv$$

$$\frac{1}{2} dpVdv = \frac{1}{2} Vd_1(Vdv)^2$$

$$\therefore dp = V^2 d_1 dv$$

$$V^2 = \frac{dp}{dvd_1}$$

or, the velocity of transmission $V = \sqrt{\frac{dp}{dvd_1}}$. Since dp = increase in pressure = stress, and dv = diminution in volume of 1 cc. of the medium = strain,

$$\frac{dp}{\frac{dv}{1}} = \frac{\text{stress}}{\text{strain}} = \text{elasticity}$$

$$\therefore V^2 = \frac{E}{d}$$

COMPRESSIONAL WAVE IN A GAS WHICH OBEYS BOYLE'S LAW AND IS ISOTHERMAL IN CHARACTER

II Let the initial pressure $= p$ initial volume $= v$
final pressure $= p + dp$ final volume $= v - dv$

and $pv = \text{const.}$

Then

$$pv = (p + dp)(v - dv)$$
$$= pv + pdv + vdp - dpdv$$

Cancelling terms and neglecting the product $dpdv$,

$$\frac{dp}{\frac{dv}{v}} = p$$

Recalling that for simplicity in Par. I we took dv as the decrease in volume of 1 cc., we have the isothermal elasticity

$$\frac{dp}{\frac{dv}{v}} = E$$

COMPRESSIONAL WAVE ADIABATIC IN CHARACTER

III Here $pv^n = \text{const.}$, hence

$$pv^n = (p + dp)(v - dv)^n = (p + dp)v^n\left(1 + \frac{dv}{v}\right)^n$$
$$= (p + dp)v^n(1 - ndv + \text{terms of a higher order})$$
$$\therefore\ pv^n = pv^n - npv^{n-1}dv + dpv^n - nv^{n-1}dpdv$$

Cancelling and neglecting terms of the second order,

$$v^n dp = npv^{n-1}dv$$
$$vdp = npdv$$
$$\frac{dp}{\frac{dv}{v}} = np$$

Here $\frac{dp}{\frac{dv}{v}}$ represents the adiabatic elasticity $E = np$,

$$\therefore\ V = \sqrt{n\frac{p}{d}}$$

IV The compression in ordinary sound waves is so small, increasing from a pressure of 1 atmosphere to perhaps 1 and 1 millionth atmospheres, that the neglecting of second-order terms is justified. The assumption that Boyle's Law holds for so limited a range as the one indicated is also justified. Lord Rayleigh has estimated that the variation in pressure of a sound disturbance barely audible is $+ 6 \times 10^{-9}$ atmospheres, an amount so small as to be incapable of direct measurement.

DISCUSSION

H. B. Bernard (written). Due to the many features encountered in piping natural gas, such as condition of pipe, type of joints, bends, temperature changes, etc., it is doubtful whether the author's determinations are of any value outside of the laboratory. For practical conditions, the formulæ derived by T. R. Weymouth from numerous observations and published in Vol. 34 of the Transactions of the Society are unquestionably of sufficient accuracy in problems involving the flow of natural gas.

Referring to Table 1, it appears that the coefficients of viscosity are computed from the coefficient of viscosity for air and the relative coefficients of viscosity as determined on p. 985. It is unfortunate that the author has neglected to give sufficient data to permit the checking of these values by the formula on p. 983.

In determining the relative densities by an effluxometer, the densities are in the ratio of the squares of the times of efflux. In view of the fact that the coefficients of viscosity are in the ratio of the times of efflux, the densities should be as the squares of the relative coefficients of viscosity. Using the latter values as given in Table 1, the computed densities as compared to those in the tables are as follows:

Computed	1.000	0.593	0.533	0.563	0.540	0.563	0.504	0.578	0.563	0.672	0.689
From Table 1	1.000	0.682	0.660	0.660	0.690	0.666	0.585	0.755	0.678	0.755	0.770

P. F. Walker (written). The author uses in his title the words "ratio of specific heats," and throughout his discussion shows that he is assuming that this ratio and the value of the adiabatic exponent are identities.

The statement is made at the outset that this paper and the experimental work on which it is based constitute an extension of an earlier paper dealing with the question of the variation of natural gas from Boyle's Law. In that paper the joint authors indicate a very marked deviation of the gas from the laws of perfect gases. While the numerical values shown in that paper are excessive, being made to appear large because of some unexplained fluctuations at low pressures which other experiments do not corroborate, it is true that the substance differs materially from a perfect gas.

It seems to be an ingrained notion in the minds of technical and scientific men that the adiabatic exponent must be the ratio of specific heats. This is far from being the truth, however, when the gas is of such a character that it fails to follow the laws of perfect gases to such an extent that the variation is worthy of notice. The point brought out in the previous paper by the author, and in substance fully attested to by all who took part in the discussion, is that this gas in its behavior does vary materially from the laws of perfect gases, and hence it must follow that the value of the exponent found for an adiabatic cannot be taken as the ratio of specific heats.

This point is of significance, provided the purpose of the investigator is to discover facts with reference to specific heat. We need not enter at this time into a discussion as to which is the important thing to be determined. Every contribution to our knowledge of matters in this connection is valuable, and we should be grateful to the author for having brought this interesting and instructive investigation to our attention.

Wm. D. Ennis stated that while the value for n might be determined from compressor indicator diagrams, no engineer believed that this was the ratio of the specific heats. He also pointed out that there must have been some error in the author's determination of n. In Table 1 the author gave values of n ranging from 1.21 to 1.29. There was no constituent of natural gas for which the value of n was less than 1.3 except ethane, which constituent has the unique value of 1.182. It therefore seemed surprising to an engineer that the value for n for natural gas should be so low, and it was further surprising that the value was not related to the percentage of ethane in the sample. The analogy of air was illustrative, the value of n being a fair weighted mean between those values which are correct for oxygen and nitrogen.

As to the author's method of procedure, there seemed to be nothing wrong, except that the determination of n for the gas as a function of that for air by comparing the wave lengths would have seemed more conclusive if the apparatus had been tried out on some other gas, such as carbon dioxide, for which the value of n was known.

Sanford A. Moss (written). While natural gas may depart considerably from the perfect-gas laws at high pressures, the departure at pressures close to atmospheric cannot be very serious. Hence the computation of density and of specific-heat ratio from gas analysis will give reasonably accurate results.

I have made such computations for the various samples employed in Table 1, using the volumetric analysis there given, and have obtained values for the density and specific-heat ratio which differ greatly from experimental values tabulated by the author.

The method of computing density or specific gravity of a mixed gas from the volumetric analysis is well known. It consists in multiplying the percentage of each component by density of said component at standard conditions, and adding the results. A

TABLE 2 SAMPLE COMPUTATION OF DENSITY, WEIGHT ANALYSIS AND SPECIFIC HEATS OF A GASEOUS MIXTURE, FROM VOLUMETRIC ANALYSIS

NATURAL GAS FROM VINTON COUNTY, OHIO, LINE 6, TABLE 1

Component	Volumetric Analysis %	Density of Components, lb./cu. ft. at 14.7 lb. abs., 60° F.	Weight of Components, lb./cu. ft. $= B \times C$	Analysis by Weight $= D/0.05174$	c_p of Components	c_p of Mixture $= E \times F$	c_v of Components	c_v of Mixture $= E \times H$
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*
CO_2......	0.1	0.11683	0.00012	0.23	0.2020	0.0005	0.1554	0.0004
O_2........	0.8	0.08442	0.00068	1.31	0.2175	0.0028	0.1556	0.0020
CH_4......	73.0	0.04261	0.03111	60.13	0.5920	0.3560	0.4490	0.2700
C_2H_6.....	5.8	0.08212	0.00476	9.20	0.4260	0.0392	0.3600	0.0331
N_2.......	20.3	0.07424	0.01507	29.13	0.2419	0.0705	0.1835	0.0535
	100.0		0.05174	100.00		0.4690		0.3590

Density of air, 0.07638 lb. per cu. ft. at 14.7 lb. abs. and 60 deg. fahr.
Specific gravity of gas, 0.05174/0.07638 = 0.6775
n or c_p/c_v for mixture, 0.4690/0.3590 = 1.306

sample computation of density is given in columns C and D in Table 2. The values of specific heat at constant pressure and specific heat at constant volume can be found in a similar way, using, however, analysis by weight instead of analysis by volume. The ratio of the specific heats of the mixture as thus computed gives a value which can be used to compute velocity of sound, etc., and which should agree with the experimental values of the author's n.

In other words, if we have a gas mixture containing various components whose individual specific heats at constant pressure are c_p', c_p'', etc., and whose specific heats at constant volume are c_v', c_v'', etc., and if the percentages by weight of each of the components are x', x'', etc., then it can be shown that the specific-heat ratio for the mixture is

$$n = \frac{x'c_p' + x''c_p'' + \dots}{x'c_v' + x''c_v'' + \dots}$$

This formula is almost self-evident. However, I have given a deduction in *Sibley Journal*, May, 1905, page 311. A sample computation of specific-heat ratio by this method is given in Table 2, columns E, F, G, H and I.

In making computations I have used the values of c_p and c_v listed in columns F and H, Table 2. All of these values are fairly well agreed upon by various authorities except the value for ethane,

TABLE 3 COMPARISON OF VALUES OF SPECIFIC GRAVITY AND SPECIFIC-HEAT RATIO OF NATURAL GAS BY COMPUTATION AND EXPERIMENT

Line in Earhart's Table 1	Specific Gravity (Density, with Air = 1.00)		Specific-Heat Ratio, n or c_p/c_v	
	Earhart's Experimental Value	Computed Value	Earhart's Experimental Value	Computed Value
2	0.682	0.6503	1.238	1.285
3	0.660	0.6278	1.214	1.294
4	0.660	0.6370	1.209	1.305
5	0.690	0.6622	1.210	1.290
6	0.666	0.6775	1.259	1.306
7	0.585	0.5795	1.285	1.318
8	0.755	0.7320	1.293	1.259
9	0.678	0.6870	1.220	1.278
10	0.755	0.7521	1.290	1.300
11	0.770	0.7760	1.224	1.297

which I was not able to find. However, it is well known that for a perfect gas the molecular specific heat or product of specific heat and molecular weight m, is given by

$$c_p m = \frac{1.97}{1 - 1/n}$$

I have used the value of n for ethane given in Par. 2. of the paper, and obtained c_p by this formula. The value is of course in error due to the imperfection of ethane, but the discrepancy cannot be very large.

I have made computations similar to those in Table 2 for each of the gas samples given in Table 1. A comparison with the experimental results of the author is given in Table 3. As will be seen, there is an appreciable discrepancy. I do not believe this can be explained by the fact that the gases in question do not obey the perfect-gas laws.

EDGAR BUCKINGHAM, commenting on the written discussion offered by Mr. Bernard, said that the statement that the densities are proportional to the squares of the times of efflux, was at the bottom of Bunsen's method of determining the relative densities of gases by means of efflux through small orifices. This statement was not exact even for perfectly adiabatic efflux of ideal gases which have no viscosity, unless it happens that the gases have the same specific-heat ratios. In practice the method is subject to very large errors due to viscosity. In the flow of a gas towards the orifice the different parts of the gas are not moving with the same velocity, thereby causing a viscous resistance. The kinetic resistance is proportional to the square of the linear speed, while the viscous resistance is proportional to the first power. As the velocity grows lower and lower, the viscous resistance, insignificant compared with the kinetic at high velocities, becomes very important. The method ought therefore to be used with the greatest caution, as it is not otherwise reliable. Bunsen's method is an approximate, not an accurate one, and his own determinations are correct only within five or ten per cent.

THE AUTHOR. A confusion of terms seems to have arisen involving the flow of gases through a pipe. The resistance offered to the passage of a stream of gas through a pipe is a complex problem and, as Mr. Bernard has pointed out, is influenced by the physical condition of the pipes, the joints and several other factors. The coefficient of viscosity, strictly speaking, is a measure of the internal friction of the gas and is but a single factor in the problem. In many cases, no doubt, it is less important than others; it is, however, a factor peculiar to the gas itself.

It appears that in some cases the viscosity effect (or computations for "viscosity" by engineers) is taken to represent the summation of a considerable number of factors. Some of these have been taken into consideration by Mr. Weymouth in the article to which reference has been made.

The experiments on the coefficient of viscosity were made with the idea of comparing the single factor of internal friction or viscosity for gases obtained from widely different sources under similar conditions of pressure and temperature and in the same apparatus.

The values obtained cannot, of course, replace those found serviceable in practice and which include several other factors, even though in a very loose way they go by the same name; nor can they be expected to give satisfactory density values.

The rate of escape of a confined gas through an orifice is a function of the pressure, temperature, density and coefficient of viscosity. In the effluxometer we compare the densities of two gases under similar pressure and temperature conditions by the rate of escape through an orifice which must be an opening in a thin diaphragm. To obtain density comparisons, the diaphragm must be vanishingly thin in comparison with the size of the orifice. This is approximated by making a small hole in a piece of thin foil, usually of platinum. The necessity for making the diaphragm thin is to reduce the effect due to viscosity.

Thus, by suitably dimensioning the apparatus, the viscosity effect becomes small and is assumed to be negligible. When, however, we make the diaphragm very thick or for convenience cause the gas to escape through a tube whose length is great compared with the diameter of the opening, the rate of escape is determined largely by the viscosity factor. Practically, when the length of the tube is 200 times the diameter, the viscosity term becomes so large in comparison with the density factor that the density factor is neglected. The conditions which prevail in the two cases are so different that the data obtained under one set of conditions cannot be treated by the formula applying to the other.

A comparison of density determinations made by different effluxometers indicates that those made in the usual way are not accurate; the discrepancies are sometimes several per cent. The writer is therefore brought to the conclusion that the accuracy of this method is greatly overestimated.

The point raised by Professor Walker concerning the application of the law of perfect gases to the theory of velocity determinations is well taken. The degree of accuracy obtained in the experiment will determine whether the correction factor should be applied or omitted. The value of n is obtained by taking the product of three quantities, one of which is the density of the gas. The density of the gas was determined to one part in five hundred in the laboratory. This is a higher degree of accuracy than conditions really warrant, for gas from any field will show variations from day to day of perhaps one per cent. However, it was easy to determine. This, however, places a limit on the accuracy of the result, and it is not desirable to introduce a correction factor of less magnitude in the other quantities.

The experiment by Mr. Wyer and myself showed that the departures from Boyle's Law were appreciable for several atmospheres' change and were large when the pressure variations were thirty or forty atmospheres. However, when sound waves pass through a gas the pressure changes are from one atmosphere to possibly one and one-millionth atmospheres. Even for such a pressure change there should strictly be a correction for deviation from Boyle's Law, which is assumed to hold between the limits of pressure employed.

The application of this correction factor is not justified when compared with the density determinations. The correction applicable in such cases is treated in a very able way by Capstick in a paper presented to the Royal Society of London (Phil. Trans. Roy. Soc. of London, 1894, Part I, page 1.) Dr. Capstick was determining the ratio n for a series of vapors some of which were but slightly removed from saturation. Under such circumstances the deviations from Boyle's Law are considerable. He made a careful study of the case and found a small correction necessary. The natural-gas products, however, are so far removed from this condition that any modification of the value obtained from direct experiment is not justified.

No. 1563

AN INVESTIGATION OF THE INTERNAL-COMBUSTION ENGINE AS APPLIED TO TRACTION ENGINES

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Within recent years several of the largest and most prominent builders of machinery began to realize that with the development of farm machinery the traction engine had become of such importance as to offer an excellent field for engineering ability and market for machinery carefully designed and built.

2 At the present time, over one hundred manufacturing concerns in the United States are building traction engines driven by internal-combustion motors. The designs differ greatly: some have motors with horizontal cylinders, others with vertical cylinders. In some designs the power of the motor is delivered to one wheel, in others to two, and in still others to all four wheels; several designs are of the so-called "creeping grip" types.

3 The development of the traction engine for agricultural purposes has been along lines entirely different from those of the automobile. The early engines developed 60 to 80 h.p. on the brake and 30 to 40 h.p. on the drawbar; they were expensive, complicated, and unsuited for any but the largest farms of the country. The present tendencies of manufacturers are to build smaller engines and to standardize the product.

OBJECT OF INVESTIGATION

4 The purpose of this investigation was to determine the fuel economy and thermal efficiency of a great variety of traction-engine

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TABLE 1 PARTICULARS OF THE TRACTION-ENGINE MOTORS TESTED AND THE FUELS USED

Engine	Type (all 4-stroke cycle)	Ignition System	Lubrication System	Carburetor	Cooling System	Governor	Fuel: Gasoline or Kerosene	Fuel: Specific Gravity	Fuel: Heating Value, B. t. u. per lb.	Rating, b.h.p.	Bore, in.	Stroke, in.	Rev. per min.
A	hor., twin, 2-cyl.[1]	high-tension[7]	mech. op. sight-feed oilers	....	pump	throttling[2]	K	0.809	19,700	60	..	..	....
B	(same motor as A, but tested when new)		....	....	....	....	K	0.800	19,680	60	..	..	...
C	vert., L-head, 4-cyl.[1]	high-tension[3]	splash and mech. op. sight feed	Bennett	pump	throttling[2]	G / K	0.739 / 0.786	19,670 / 19,500	65	7¼	9	500
D	vert., T-head, 4-cyl.[1]	high-tension[3]	sight-feed pump	Bennett	pump	diaphragm[4]	G	0.741	19,000	50	6	7	650
E	hor., opposed, 4-cyl.[1]	high-tension[3]	st.-feed pump splash	Kingston	thermo-syphon	throttling[2]	G	0.735	17,880	50	6½	7	500
F	hor., opposed, 2-cyl.[1]	dual	mech. st.-feed oiler	Rayfield	thermo-syphon	throttling[2]	G	0.730	20,600	40	8	9	350
G	vert., L-head, 4-cyl.[1]	high-tension[3]	pump splash	Bennett	pump	throttling[2]	G	0.737	20,000	35	5	7	700
H	hor., L-head, 4-cyl.[1]	high-tension[3]	mech. st.-feed oiler	......	pump	throttling[2]	K	0.789	19,900	25	5¼	8	575
J	hor., opposed, 2-cyl.[1]	high-tension[3]	pump splash	Kingston	thermo-syphon	throttling[2]	G	0.747	19,550	25	6½	7	570
K	vert., L-head, 4-cyl.[1]	high-tension[3]	pump splash	Bennett	pump	throttling[2]	G	0.737	20,000	20	4¼	5¾	800
L	hor., single-cyl.[1,5]	low-tension[6]	sight-feed oiler	....	hopper	throttling[2]	K	0.805	19,280	16	8	12	400
M	hor., L-hd., 2-cyl., opp.[1]	Kingston dual high-tension	mech. st.-feed oiler	Kingston	pump	throttling[2]	G	0.738	20,290	20	5¼	7	650-720

[1] With mechanically operated inlet and exhaust valves.
[2] High-speed throttling type of governor that regulates the quantity of mixture.
[3] With K-W impulse starter magneto.
[4] Diaphragm type, using pressure of circulating water to control the speed.
[5] Valve in the head.
[6] Make-and-break system using an Accurate oscillating magneto.
[7] High frequency belt driven K-W magneto with trembler coil.

designs, to find out the practicability of the fuels heavier than gasoline for traction-engine use, and to compare the rating, valve setting, timing and other details of commercial traction engines. Particulars of the motors tested and of the fuels used are given in Table 1.

METHOD OF TESTING

5 The various engines were tested by a four-wrap rope brake (circumference, 10.528 ft.), the speed of the brake being determined by an automatic counter. Platform scales reading to ½ lb. were used for the brake, and scales reading to ½ oz. for weighing the fuel, water and lubricating oil. Fuel samples were taken during each test, the heat of combustion of each sample being determined by a Junkers calorimeter and the specific gravity by a hydrometer. Revolutions of engine and of brake were taken every six minutes, and the weights of fuel, of injection water, and of lubricating oil every 30 to 60 min. Four-fifths of the tests were of 60 min. duration; the others of 22 to 30 min.

GENERAL CONCLUSIONS

6 From the results of this investigation the authors have derived the following general conclusions:

a The four-cylinder motor is better adapted for belt work on account of the greater number of impulses per revolution.

b The single-cylinder motor and the two-cylinder motor operate better than the four-cylinder motor with fuels heavier than gasoline.

c Carburetors now used are satisfactory for gasoline, but a carburetor jacketed with heat from exhaust gases should be employed when operating with kerosene or with the heavier fuels.

d The ordinary automobile motor is too light for traction-engine work. The traction-engine motor should operate at lower piston speeds than the automobile motor. Motors operating at piston speeds of 700 to 900 ft. per min. are giving satisfaction.

e The vertical types of motors are preferable on account of longer life and greater accessibility.

f The valve-in-the-head type of motor has the more efficient combustion space and is to be preferred to the T-head or L-head types.

TABLE 2 FUEL CONSUMPTION AND THERMAL EFFICIENCY OF THE TRACTION-ENGINE MOTORS TESTED

Engine	Per cent of full load	Fuel per hour per brake h.p., lb.	Thermal efficiency, per cent	Engine	Per cent of full load	Fuel per hour per brake h.p., lb.	Thermal efficiency, per cent	Engine	Per cent of full load	Fuel per hour per brake h.p., lb.	Thermal efficiency, per cent
A (60 B.H.P.) (Kerosene)	57.8	0.963	13.43	F (40 B.H.P.) (Gasoline)	8.7	3.210	3.85	J	69.1	0.700	18.80
	99.8	0.808	15.88		32.7	1.140	10.34		49.4	0.860	15.10
	70.8	1.032	12.53		32.7	1.120	12.36		49.5	0.930	14.00
	38.2	1.683	7.67		43.6	1.130	11.22		37.2	1.540	8.40
	111.3	0.928	13.93		43.2	0.950	13.02				
	98.8	0.874	14.77		43.9	0.980	12.54	K (20 B.H.P.) (Gasoline)	127.7	1.003	12.69
	70.4	0.996	12.97		64.7	0.808	15.28		66.4	1.178	10.80
	41.5	1.188	10.88		66.2	0.750	16.47		92.9	0.976	13.03
	23.0	2.154	6.01		88.0	0.730	16.94		34.1	1.638	7.77
					86.0	0.890	13.90		51.9	1.380	9.22
B (60 B.H.P.) (Kerosene)	18.1	1.780	7.26		82.4	0.780	15.25		75.8	1.071	11.87
	48.9	1.480	8.74		83.7	0.770	16.06		98.5	0.905	14.06
	17.5	1.750	7.39		84.0	0.770	16.06		36.9	1.948	6.37
	33.5	1.460	8.86						54.6	1.385	9.18
	51.1	1.100	11.76	G (35 B.H.P.) (Gasoline)	105.3	0.643	19.80		75.7	1.043	12.20
	51.9	1.230	10.52		127.0	1.047	12.10		110.3	0.818	15.44
	84.8	0.780	16.58		107.0	0.736	17.30				
	66.4	0.990	13.07		98.5	0.706	18.00	L (16 B.H.P.) (Kerosene)	73.4	0.865	9.71
	65.9	0.760	17.00		81.6	0.779	16.30		45.8	1.447	9.12
	84.6	1.090	12.87		50.3	1.030	12.40		99.8	1.330	9.92
	97.8	0.810	15.97		24.0	1.690	7.50		23.3	1.778	7.42
	104.3	0.980	13.19		80.6	1.045	12.20		48.8	1.108	11.92
	127.8	0.940	13.76		98.8	0.685	18.60		72.1	1.020	12.93
					81.8	0.808	15.80		103.3	1.248	10.57
C (65 B.H.P.) (Gasoline)	28.1	1.227	10.53		52.0	1.025	12.40		23.2	1.865	7.07
	65.1	0.793	16.34		23.6	1.791	7.10		70.8	1.303	10.13
	66.4	0.874	14.82						98.2	1.003	13.17
	100.6	0.731	17.73	H (25 B.H.P.) (Kerosene)	113.0	0.800	15.97		104 4	0.936	14.10
	102.1	0.767	16.89		131.2	1.019	12.54		71.5	1.279	10.32
	112.7	1.010	12.83		92.1	1.042	12.27		70.4	1.185	11.28
	27.5	1.382	9.38		58.1	1.161	11.02		44.8	1.187	11.22
					28.7	2.097	6.10		22.7	1.929	6.84
C (65 B.H.P.) (Kerosene)	68.1	1.008	10.06		110.7	0.794	16.08		71.1	1.103	11.97
	67.8	1.010	12.95		121.7	0.794	16.08		101.3	1.007	13.11
	101.8	0.910	14.36		91.4	0.951	13.45		93.0	1.043	12.64
	36.0	1.582	8.26		57.9	1.230	10.40		45.3	1.103	11.97
	101.3	1.328	9.83		27.2	2.099	6.09		22.9	1.702	7.76
	102.9	1.292	11.63		132.8	0.989	12.93				
	102.9	1.274	10.24		91.9	0.930	13.73	M (20 B.H.P.) (Gasoline)	79.5	0.884	14.20
									30.5	1.508	8.30
D (50 B.H.P.) (Gaso.)	90.0	0.997	13.43	I (25 B.H.P.) (Gasoline)	78.8	0.950	13.70		84.2	0.913	13.20
	94.4	0.899	14.88		80.4	0.810	16.10		18.9	1.753	7.20
	74.8	0.820	16.33		86.8	0.930	13.90		56.8	1.153	10.90
	52.6	0.883	15.17		39.9	1.050	12.40		87.2	1.103	11.40
					24.4	1.640	7.90		60.1	1.020	12.30
E (50 B.H.P.) (Gasoline)	23.0	1.413	10.10		72.4	0.760	17.20		104.1	0.811	15.40
	23.6	1.204	11.87		73.2	0.680	19.20		104.3	0.803	15.60
	47.1	0.670	21.32		96.0	1.040	12.50		60.5	1.103	11.40
	48.8	0.720	19.80		93.5	1.060	12.20		79.2	0.908	13.80
	73.2	0.710	20.10		95.9	1.030	12.60		77.8	1.051	11.90
	74.0	0.710	20.10		98.9	0.730	17.80		41.2	1.428	8.80
	90.6	0.740	19.41		70.1	0.740	17.50				

g The combination of the forced feed and splash oiling system gives good results.

h The jump-spark system, on account of its mechanical simplicity, is the best system of ignition for traction engines of more than one cylinder.

i The fuel-economy range is from about 1.30 lb. per b.h.p. per hour at one-fourth load to about 0.7 lb. per hour at full load. The fuel consumption in lb. per b.h.p. per hour is very nearly the same for both gasoline and kerosene.

j The thermal efficiencies at full load vary from 14.88 to 19.8 per cent for gasoline fuel, and from 13.7 to 15.97 per cent for kerosene.

RESULTS OF TESTS

7 The results of the tests relating to fuel consumption and thermal efficiency are given in Table 2, and from the former the curves of Fig. 1 have been plotted.

CONCLUSIONS FROM TESTS

8 The lower fuel economy of the tests recorded for engine A as compared with tests on engine B (the same engine) was due to the difference in the spark advance. During the operation of engine A a greater spark advance was used, and more cooling water had to be injected into the cylinder with the fuel to prevent preignition. Water injection had to be used also during the tests of several engines with gasoline fuel (engines D, E and F).

9 In the case of several engines the valve setting had to be changed before satisfactory operating conditions could be secured. In one case the preliminary tests indicated that the carburetor was too small for the engine. Some companies in such cases remove several balls from the auxiliary air valve, a practice which is followed by poor fuel economy of the engine. These experiences indicate the poor inspection and testing facilities prevalent with some manufacturers of traction engines.

10 In order to facilitate comparison of various types, the motors of the tests were grouped in Fig. 1 as follows:

Group I Motors which develop at full load 15 to 26 h.p. on brake. The tests recorded for motors H, J, K, L and M are included in this group.

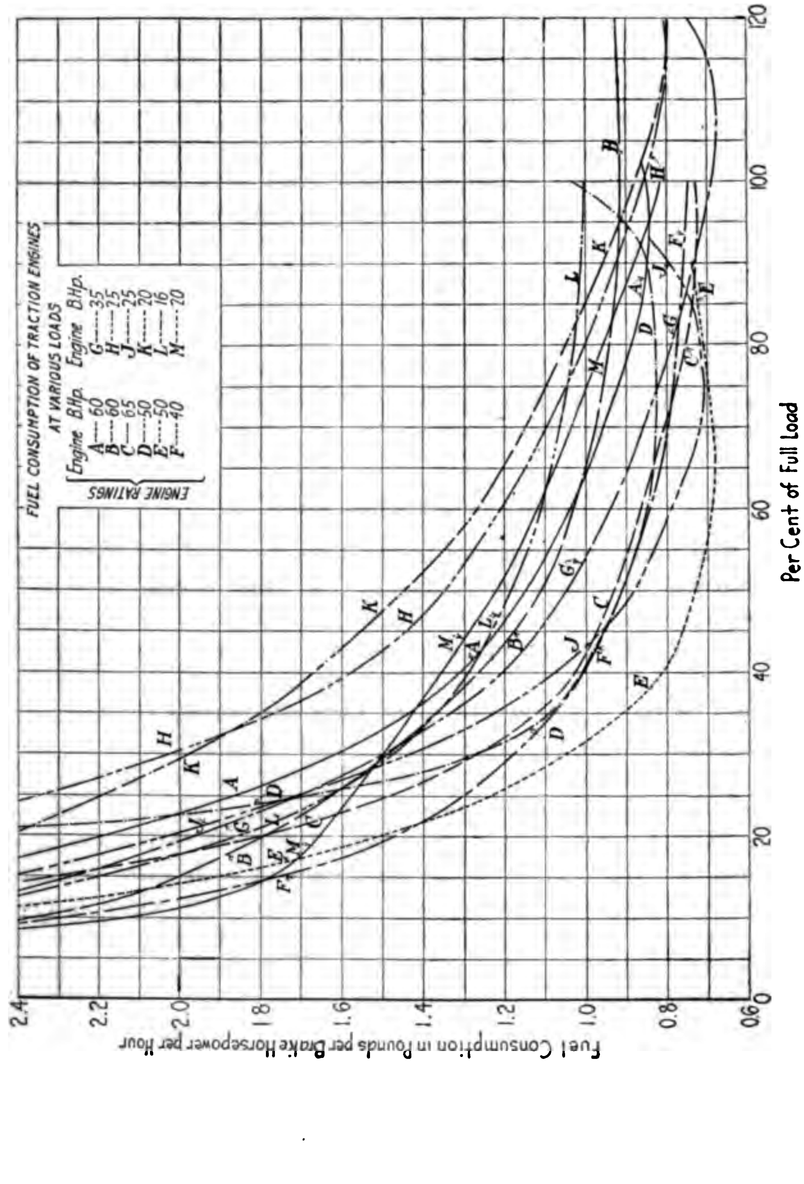

Fig. 1 Curves of Fuel Consumption of Traction Engines at Various Loads

GROUP II Motors which develop 26 to 51 b.h.p. In this group belong the motors recorded in the tables as D, E, F and G.

GROUP III Motors which develop more than 51 b.h.p. The tests recorded for motors A, B and C represent this group.

The average fuel economy for the various groups follows:

Per Cent of Full Load	Fuel Consumption, Lb. per B.H.P. per Hour				
	Group I		Group II	Group III	
	Gasoline	Kerosene	Gasoline	Gasoline	Kerosene
25	1.853	1.990	1.416	1.47	1.837
50	1.147	1.265	0.893	0.93	1.190
75	0.940	1.044	0.767	0.78	1.013
100	0.855	0.935	0.720	0.73	0.977

11 From the data on Group I and Group III Table 3 has been computed. Computations could not be made for Group II, as the engines in this class were not operated on kerosene.

TABLE 3 COST OF TRACTION-ENGINE POWER WITH GASOLINE AND KEROSENE AS MOTOR FUELS

Per Cent Load	Cost per Horsepower-Hour in Cents with								
	60° Baumé Gasoline at Prices per Gallon (in Cents) of					45° Baumé Kerosene at Prices per Gallon (in Cents) of			
	9	12	15	18	21	5	7	9	11
Group I[1]									
25	2.72	3.62	4.53	5.43	6.34	1.49	2.10	2.70	3.30
50	1.68	2.24	2.80	3.36	3.93	0.95	1.33	1.71	2.09
75	1.38	1.83	2.30	2.75	3.22	0.78	1.10	1.41	1.71
100	1.25	1.67	2.09	2.51	2.93	0.70	0.98	1.26	1.54
Group III[2]									
25	2.16	2.88	3.59	4.31	5.03	1.37	1.92	2.47	3.02
50	1.37	1.82	2.27	2.73	3.18	0.89	1.25	1.61	1.97
75	1.14	1.53	1.90	2.29	2.67	0.76	1.06	1.37	1.67
100	1.07	1.43	1.78	2.14	2.50	0.73	1.03	1.32	1.61

[1] Motors developing 15 to 26 b.h.p. on full load. [2] Motors developing over 51 b.h.p. on full load.

12 Table 3 shows the advantages of the kerosene-burning engine. Considering Group I, 10.07 gal. of kerosene will deliver as much

power as 10 gal. of gasoline. With kerosene at 10 cents per gal. and gasoline at 20 cents, the cost with gasoline fuel will be 1.99 times that with kerosene fuel for the same power developed. Considering Group III, 12.32 gal. of kerosene will deliver as much power as 10 gal. of gasoline. This gives a ratio of 1.62 to 1 with prices of fuel at 10 cents per gal. for kerosene and 20 cents for gasoline. The advantages of the kerosene engine are offset to a greater or less degree, depending upon the operator, by the added trouble in handling. The life of the motor will also be somewhat shortened when using kerosene fuel. To this should be added the lower reliability with the heavier fuel. In some work done by traction engines reliability is the main factor.

13 Due to the high price of gasoline new carburetors are being placed on the market which handle kerosene very satisfactorily, and eventually it will be used more as a fuel for traction engines than will gasoline.

14 A study of the valve timing of the different motors in these tests shows no uniformity except that the majority of the motors are so timed that the inlet valve does not open until after the exhaust valve is closed. The timing given in Table 4 is offered as a result of the authors' study and experience with traction engines.

TABLE 4 VALVE TIMING FOR MOTORS OF TRACTION ENGINES

Speed of Motor, r.p.m.	Exhaust Valve		Inlet Valve	
	Opens before outer center	Closes after inner center	Opens after inner center	Closes after outer center
200	20° to 25°	0° to 3°	0° to 3°	15° to 20°
300	22° to 27°	0° to 5°	2° to 5°	15° to 20°
400	27° to 32°	2° to 5°	2° to 7°	15° to 20°
500	30° to 35°	4° to 8°	5° to 10°	18° to 23°
600	35° to 40°	4° to 8°	8° to 12°	18° to 23°
700	40° to 45°	6° to 10°	10° to 12°	20° to 25°
800	45° to 50°	6° to 10°	10° to 12°	20° to 25°

15 By observing the slope of the curves at the full load in Fig. 1 it is evident that several of the motors are not rated at the capacity for best efficiency. The motors represented by tests G, H, K and M are underrated, E and J are overrated, while A, B, C and F are properly rated.

DISCUSSION

Wm. D. Ennis said that he was particularly interested in the reported economy of the engines tested, which, from the values given in Table 2, averaged not far from 0.8 lb. of fuel per h.p-hr. This was a strikingly favorable figure as compared with the hot-cap engines of the semi-Diesel type, in which liquid fuel was injected as a liquid into the cylinders. These hot-cap engines, at about 50 lb. compression, used about 0.9 lb. of kerosene per b.h.p-hr., and just about the same amount of fuel oil of the usual eastern grade. It was an interesting question whether the extraordinary economy of the carburetor type of engine, as presented in the paper, was due to a higher compression than 50 lb. or to better combustion. Some light would be thrown on the matter if the authors would tabulate the pressures realized, or at least the clearances of the engines. He said that he did not believe it would be generally admitted that with equal compression and the same fuels the hot-cap engine was inferior in economy to the engine of the carburetor type.

A. A. Potter. Motors for traction engines operate at compression pressures of 55 to 70 lb. per sq. in. Engine No. C operated at a compression pressure of 62.5 lb. per sq. in.

Tests by the author, of an 80-h.p. hot-cap low-compression engine, with 40-deg. Baumé oil, gave an economy of 0.8 lb. per b.h.p. per hour at full load and 1.15 lb. per b.h.p. per hour at half load. Another hot-cap low-compression engine of 125 h.p. capacity produced a brake horsepower per hour, at full load, for 0.79 lb. of fuel of 0.830 specific gravity.

The time of ignition can be more accurately controlled in the case of the carburetor-type engine, which gives this type of engine considerable advantage in the hands of an expert operator. The experience of the author with hot-cap and carburetor types of engines indicates that with light petroleum fuels the carburetor-type engine compares favorably with the hot-cap low-compression types. For fuels heavier than 40 deg. B. the hot-cap-type engine is superior.

No. 1564

CLASP BRAKES FOR HEAVY-PASSENGER EQUIPMENT CARS

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Non-Member

The essential requirements of a power brake are to stop the vehicle to which it is applied in the shortest possible distance consistent with maximum rail adhesion during emergency braking, and in the minimum distance consistent with accuracy and smoothness during service braking, all of which is largely dependent upon the type of equipment employed, the manner in which it may be operated, and the braking ratio (percentage of brake power) that can be successfully used.

2 The braking requirements for present-day heavy steel passenger-car equipment can best be appreciated by a careful analysis of the records of a number of passenger-train-brake tests with the earlier light wooden cars and the heavy steel equipment of to-day, and for those who care to make such an analysis the paper which was presented by S. W. Dudley at the February, 1914, meeting of the Society is unqualifiedly recommended. For ready reference, however, it might be interesting to state that in 1902 an exhaustive series of brake tests was made on the Pennsylvania Railroad, under the supervision of A. W. Gibbs, with trains consisting of one locomotive and comparatively light wooden cars, in which stops were made from a speed of 60 m.p.h. with emergency brake applications in approximately 1000 ft.

3 In 1903 similar tests were made on the Central Railroad of New Jersey, under the writer's supervision, in which passenger trains consisting of what was then considered modern equipment were stopped from a speed of 60 m.p.h. in an average distance of 970 ft.

4 Early in 1905 another series of tests was made on the Pennsylvania Railroad with equipment similar in weight and construction

[1] Consulting Air Brake Engineer, New York Central Railroad Co.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

to that used in the 1902 and 1903 tests, with substantially the same results.

5 The emergency braking ratio in the Pennsylvania Railroad and the Central Railroad of New Jersey tests did not exceed 125 per cent of the car weight, and a reducing mechanism was employed for automatically reducing the braking ratio during the stops, so that the mean effective ratio was approximately 100 per cent.

6 Based upon results obtained in the three brake tests just referred to, a distance of 1000 ft. was considered a desirable theoretical emergency stop from a speed of 60 m.p.h. for a passenger train having the ordinary "high-speed brake."

7 In the fall of 1905, closely following the second test of the Pennsylvania Railroad, similar tests were made on the New York Central Railroad, under the supervision of C. H. Quereau. The locomotive and cars used in this test weighed, however, considerably more than the ones used in previous tests, and the emergency stops from 60 m.p.h. were over 1200 ft. in cases where the air-brake equipment and braking ratio were substantially the same as had formerly produced approximately 1000-ft. stops with lighter equipment.

8 Results of the New York Central test immediately established the fact that as the weights of the individual vehicles of which the train was composed increased, the braking ratio would have to be increased if the length of the stop was to be no greater than was formerly made with lighter equipment, and to meet the requirements of the heavier locomotives and cars the air-brake manufacturers immediately developed an air-brake equipment with which could be had a higher braking ratio than was obtainable in previous tests with lighter locomotives and cars.

9 In 1908 another exhaustive series of tests was made on the Southern Pacific Railroad with still heavier locomotives and cars, in which it was found that a distance of over 1300 ft. was required for stopping the heavier trains from a speed of 60 m.p.h. with no greater emergency braking ratio than was formerly required for making a 1000-ft. stop with the lighter equipment.

10 In 1909, R. B. Kendig, Mem.Am.Soc.M.E., conducted still another brake test on the Lake Shore & Michigan Southern Railroad with trains consisting of locomotives and cars closely approximating present-day equipment in weight, for which was required an emergency braking ratio of 180 to 200 per cent of the car weight for producing approximately a 1200-ft. stop from a speed of 60 m.p.h. These tests demonstrated to the entire satisfaction of all who participated

in them that the emergency braking ratio for heavy steel cars would have to be not less than 180 per cent of the car weight if the emergency stops were to be made in no greater distance than formerly required for the lighter cars.[1]

11 Realizing that 180 to 200 per cent braking power applied to one side of a car wheel would probably produce ill effects on journals, brasses, trucks, etc., the writer had made a careful and thorough analysis of the force action on car journals as effected by high braking forces, and it is unfortunate that these analyses are of a character and magnitude which preclude the practicability of reproducing them in a paper of this kind, for they show conclusively the undesirability

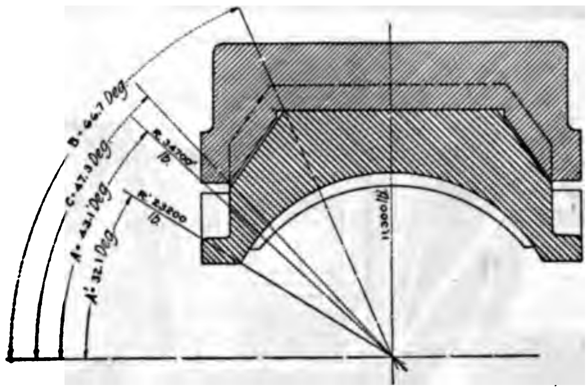

FIG. 1 ANALYSIS OF FORCE ACTIONS OF A SINGLE-SHOE BRAKE ON A 5 BY 9-IN. M.C.B. JOURNAL

of applying to one side of the wheel a braking ratio of sufficient magnitude for stopping the modern heavy steel equipment in no greater distance than formerly required for stopping the lighter wooden equipment. A summary of these analyses is, however, shown in Figs. 1 to 5.

12 Fig. 1 shows a section of an M.C.B. 5 by 9-in. journal brass and wedge under a 150,000-lb. car with an average nominal journal

[1] It is not the intention to show by the above references to brake tests the distance in which trains may be stopped in service. In conducting brake tests, variations in equipment by which stopping distances are affected are necessarily reduced to a minimum, otherwise the results would not be comparable. The stopping distances referred to should, therefore, be used only as a basis of comparison for different equipments, and it should not be assumed that such stops would be reproduced in actual train service. On the contrary, it may safely be assumed that the stops with service trains should be much longer than test records show.

load of 11,300 lb. Lines R and R' (Fig. 1) show the resultants of all loads acting on the journals with a single-shoe brake, arranged in accordance with the M.C.B. recommendations for such a brake and with an emergency braking ratio of 190 per cent. (The resultants R and R' are for different locations of wheels and directions of rotation.) It will be observed that the lines of action R and R' are at a considerable distance below the supporting point between brass and wedge; that is, angle A is less than angle B, and to push the journals out of the brasses during emergency braking is a natural thing to expect under the conditions stated.

Fig. 2 Actual Displacement of Journals and Brasses under Service Conditions Approximating those of Fig. 1

13 Fig. 2 is a photograph showing the actual displacement of journals and brasses under service conditions closely approximating those described in Fig. 1. While this photograph is made from a four-wheel truck, the brake arrangement, nominal journal load, braking ratio, etc., are, as previously stated, substantially as shown in Fig. 1.

14 As resultant R is affected in direction and magnitude by the distance from horizontal center line of wheels to center of brake shoes at face, Fig. 3 was made to show a summary of the analysis of the force action on journals with brake shoes suspended 10 in. from rail (8 in. below wheel centers), which is lower than the M.C.B. standard.

15 The braking ratio employed in this case is approximately 160 per cent of the car weight. Angle A is still less than angle B, and displacement of journals and brasses may be expected to result therefrom.

16 Fig. 4 is a photograph taken at the end of a stop with the car from which the summary analysis shown in Fig. 1 was made, and seems to confirm the analysis so far as concerns the effect of the braking load on journals.

17 There seems to have been an open question in the minds of some as to whether the displacement of journals and brasses is controlled by the difference in angles A and B or A and C, that is, the points between which the brass is supported by the wedge seem to

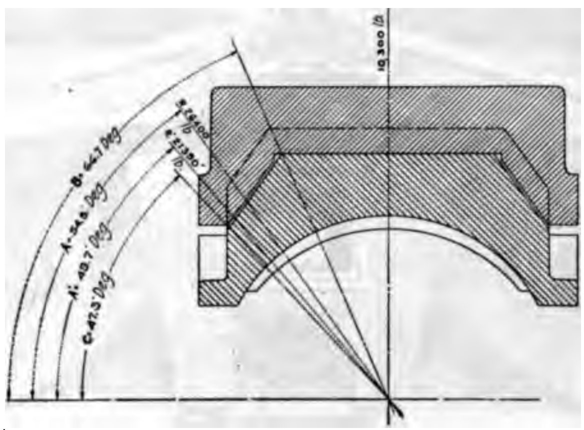

FIG. 3 ANALYSIS OF FORCE ACTIONS ON JOURNALS WITH BRAKE SHOES SUSPENDED 10 IN. FROM RAIL

have been debatable; but a comparison of Figs. 1 and 2, and 3 and 4 should justify the statement that they are supported in their normal position only by the horizontal surface contact with the wedge, and if angle A is less than angle B the journals will be displaced.

18 To check further the conclusions stated in the preceding paragraph, an analysis was made of the force actions on a 5 by 9-in. journal of a 142,000-lb. car having six-wheel trucks and a nominal journal load of 10,600 lb., a service braking ratio of 85 per cent of the car weight, and the arrangement of foundation brake gear the same as in Figs. 3 and 4. A summary of this analysis is shown in Fig. 5, from which it will be observed that angle A is practically 5 deg. less than angle B, and in testing the cars out in road service it was ob-

served that some journals were displaced during service braking while others were not. The analysis as summarized in Fig. 5 and the observations relating thereto strengthen the belief that if angle A is less than angle B the journals will be displaced. Also, that where angles A and B, as determined from drawings, practically coincide, there may be sufficient variations due to wear or construction of truck and brake details, or rocking of brasses and wedges, to change either of these angles sufficiently in service to cause the journals to be displaced or maintain a state of equilibrium.

19 It must be admitted that the high shoe loads applied to one side of the wheel only will produce undesirable results on journals

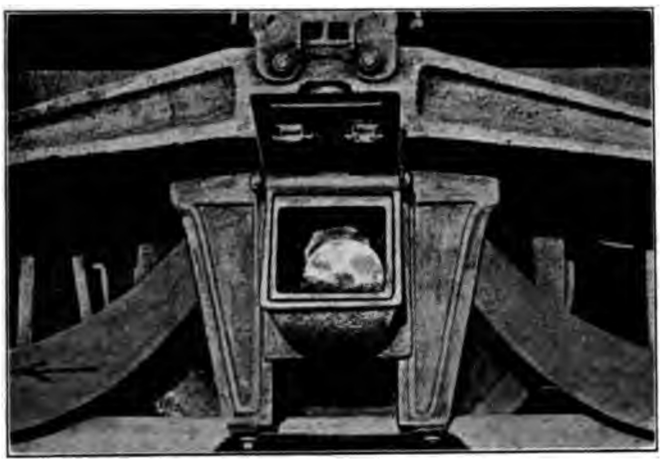

FIG. 4 PHOTOGRAPH TAKEN AT THE END OF A STOP WITH CAR ANALYZED IN FIG. 1

and brasses as shown in Figs. 1 to 5, inclusive, and in addition thereto it would seem from the discussion which is to follow that the conditions previously described are seriously objectionable from the viewpoint of train braking.

20 Consideration has been given to a change in brass and wedge design for the purpose of minimizing displacement of journals as referred to in the preceding discussion, but if this is done it will still be quite difficult to stop the heavy steel car in substantially the same distance formerly required for the lighter wooden car. While on the other hand it has been conclusively demonstrated that with a properly designed and constructed clasp brake the maximum avail-

able rail adhesion can be utilized in train braking, thereby reducing the emergency stops to a question of adhesion rather than permissible braking ratio.

21 It is therefore the writer's opinion that a suitable design and make of clasp brake should be used on modern steel passenger equipment, the advantages of which are briefly stated in the following paragraphs.

SAFETY

22 In case of danger, requiring an emergency brake application, a much shorter stop can be made with the clasp brake than with a single-shoe brake, other conditions except those affected by the brake gear being the same in both cases.

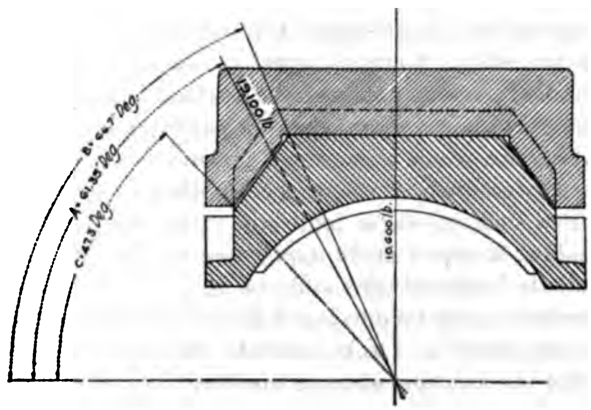

Fig. 5 Analysis of Force Actions of Brake Gear of Figs. 3 and 4 on a 5 by 9-in. Journal

23 If properly designed, manufactured and installed, there is no occasion to disconnect any part of the clasp-brake rigging between shopping of cars. The probability of the brake becoming inoperative through a failure to properly replace cotters when disconnecting the brake with the car in transit and the loss of brake pins resulting therefrom is reduced to a minimum.

24 A thin brake shoe, or the loss of a brake shoe, does not in all cases necessitate cutting out a brake to save the brake beam.

25 If the clasp brake is properly designed, manufactured and applied to the car it will be practically impossible to adjust the rigging so as to impair its efficiency or interfere in any way with its proper operation.

26 The axles and truck frames, in addition to performing their usual functions, become safety hangers for the major portion of the brake rigging, thus reducing to a minimum the possibility of derailment that might be caused by brake rigging dropping on the track in case of failure of the truck brake gear.

27 While the possibility of disconnected brake parts dropping on the track is greatly reduced in comparison with the single-shoe type of brake gear, the danger is further reduced on account of the clasp-brake parts being much lighter than those of the single-shoe type.

ROUGH VS. SMOOTH TRAIN HANDLING, ACCURACY IN MAKING STOPS, ETC.

28 Many modern passenger trains are, on account of the inherent shortcomings of the "single-shoe" type of brake, extremely difficult to handle smoothly. Careful investigation of the complaints of roughly handled passenger trains indicates that most of these troubles are due largely to non-uniform braking power and the *time in which it is developed*, as a result of improper piston travel.

29 In service braking at low speeds, whether for the purpose of stopping from such speeds or for completing stops from high speed, such as making a *second brake application* as the stopping point is approached, the brake power should be light and the retardation resulting therefrom must be developed slowly, or simultaneously on all cars, if smooth handling is to be insured. Smooth service stops from all speeds are also contingent upon the flexibility of the brakes.

30 The seriousness of slack-action shocks is greater than in former years on account of the greater average weight of cars and increased length of trains, and the chances for producing them are much greater with the single-shoe brake than was formerly the case with lighter cars and shorter trains.

31 Contrasting the desired rate at which the braking power should be developed at low speed, making service or emergency stops from high speed in a minimum distance necessitates developing a high nominal braking power, and in addition thereto it must be developed rapidly.

32 The rate at which both service and emergency braking power are developed is largely dependent upon piston travel, and with a view to producing the best results under all conditions the automatic brake is built on the principle of maintaining, as near as practicable, 8 in. of piston travel at all times and under all conditions. As an example,

if, during service braking at low train speeds the piston travel resulting from 10 lb. brake-pipe reduction is only 5 in. instead of 8 in. (with some brake riggings it is 5 in. or less), the braking power will be fully 100 per cent greater than with the predetermined standard piston travel of 8 in., and with the shorter travel a 10- or 15-lb. reduction will practically equalize the auxiliary reservoir and brake-cylinder pressure, thereby materially reducing the flexibility of the brake. While the vibration of the car may cause the 5-in. piston travel to increase to practically normal before the stop is completed, it will not do so except when stopping from high speed. Moreover, if the travel does increase before the stop is completed it will contribute nothing to smooth handling, as the shock will have occurred while the travel was short.

33 Other things being equal, the clasp brake will develop a higher percentage of braking power than the single-shoe brake during heavy service or emergency applications, but for light service braking at low speed the brake power developed from a given brake-pipe reduction is much less with the clasp brake than with the single-shoe brake, and it is developed at a much lower rate, thereby insuring smoother train handling than can be had with the single-shoe brake.

34 The results just cited are due to the fact that with the single-shoe brake the piston travel is practically proportional to the train speed and cylinder pressure, whereas with the clasp brake, with a shoe on each side of the wheel, the horizontal wheel or shoe movement relative to the brake cylinder is reduced to a minimum, and such movement if produced from any cause will have no effect on the piston travel. Moreover, with the clasp brake the shoes are located sufficiently close to the horizontal center line of wheel centers to obviate the *pulling down* of truck frames and variations in piston travel resulting therefrom.

35 The removal of worn shoes and their replacement by a given number of new shoes without readjustment of slack, as is frequently done on long runs, will not affect the piston travel with the clasp type of brake to the same extent as would occur with the single-shoe type of brake.

36 The only remedy that can be offered for the difficulties arising from improper piston travel, which so seriously affects the braking power resulting from a given brake-pipe reduction and the rate at which it is developed, is to apply a truck-and-body brake gear that will substantially insure uniform piston travel under all conditions of speed and cylinder pressure. The use of the clasp type of brake

rigging with body brake gear to suit will, to a large extent, accomplish these results and restore the flexibility of brake operation which existed prior to the adoption of extremely heavy cars and long trains of the present day equipped with single-shoe brakes.

IMPROVES RIDING QUALITIES OF EQUIPMENT

37 The high brake-shoe loads developed on one side of the wheels with a single-shoe brake produce a binding effect between pedestals and oil boxes, which interferes with the proper action of the truck springs during an application of the brakes, and when the shoes are hung low, as is necessary with the ordinary six-wheel truck and single-shoe brake, the pulling-down effect of the truck defeats in many cases the purpose of the truck equalizing springs. This binding between pedestals and oil boxes and the increased load on truck springs cause the car to ride hard when brakes are applied. These evils do not exist with the clasp brake.

ELIMINATION OF HOT BOXES

38 With the single-shoe type of brake rigging it will be observed that the high pressure exerted by the shoe on one side of the wheel causes the tilting of brasses sufficiently to lift one side of the brass a considerable distance away from the journal (see Figs. 3 and 4), so that a wide space is open for waste to be caught between the brass and the journal when the brake is released and the brasses and journals resume their normal position.

39 Investigation has shown that waste has been found wrapped around the journal, and that the collars on the axles are forced against the sides of the boxes. Further, these effects are not confined to emergency applications but will also be noted in service applications of the brake and are all in the direction of producing hot boxes, while the unequal distribution of braking power and binding between boxes and pedestals has a tendency to cause slid flat wheels.

DECREASE IN MAINTENANCE COST AND BRAKE-SHOE COST

40 While the principal advantages inherent in the clasp brake, of greater flexibility in service braking, etc., are outlined in the foregoing, and the primary consideration for its adoption must be the increased emergency efficiency over the single-shoe type of brake, providing as it does for the possibility of greatly shortened stops, with a lesser tendency to slide wheels and consequent increase in

safety, the clasp brake will also, due to the principles involved in its design and construction, show not only a decided decrease in cost of maintenance in the brake rigging itself, but a substantial decrease in the cost of brake-shoe material for equal amounts of energy dissipated.

COST OF TRAIN OPERATION

41 Investigation has developed the fact that in some cases with a single-shoe type of brake on modern passenger-equipment cars and the piston travel adjusted to proper limits, approximately 35 per cent of the available tractive effort of the locomotive was consumed in pulling the train against the effect of brake shoes dragging on the wheels with the brakes released.[1] With the clasp type of brake and the resulting increased shoe clearance, this loss is eliminated, leaving better maintenance of schedules and corresponding decreased cost of train operation.

CONCLUSION

42 In considering the application of clasp vs. single-shoe brakes to the modern heavy steel passenger car of to-day, the *advantages* of the former over the latter, as enumerated above, are but secondary to the primary question to be settled, namely: Are the present-day trains to be stopped from given speeds in no greater distance than was required ten to fifteen years ago for stopping the lighter wooden cars? If so, the question of whether or not an efficient clasp brake should be used on such trains is conclusively settled.

43 The collision energy of the heavy steel passenger train as compared to the lighter wooden train has increased directly in proportion to the increased weight, and in geometrical proportion to the increased speed in cases where speeds have been increased, to say nothing of the increased density of traffic. It would therefore seem that the use of a clasp brake is essential in successfully controlling the speed of present-day or future passenger trains, and without regard to nominal increase in first cost or multiplicity of parts of brake gear resulting therefrom.

44 The foregoing discussion on the relative performance of the clasp and single-shoe brake is with the distinct understanding that the former is designed upon a scientific engineering basis and is constructed and installed in accordance with the principles involved in the design; for while the claims made for the clasp type of brake have

[1] See M.C.B. Assn. Proceedings, 1910, p. 97, par. 3.

been conclusively demonstrated by exhaustive tests and road service, it has likewise been demonstrated that where the clasp brake is improperly designed or carelessly manufactured and installed, the results obtained in service are in many respects less desirable than with the single-shoe brake.

DISCUSSION

H. H. VAUGHAN commended the author for calling attention again to the desirability of the clasp brake, which had been so clearly established by experiment and experience. He said the only reason the brake was not adopted by the Canadian Pacific Railway was that the type of truck used made the application difficult, and that with the low speeds there was apparently no advantage in the clasp brake over the ordinary brake with a large area of brake shoe.

S. G. THOMSON agreed with the author as to the superiority of the clasp type of brake. He said it was perfectly reasonable to think that better results would come from applying the braking force on both sides of the wheel; the riding of the car was much better, there were fewer vibrations, and the decrease in the number of hot boxes was undoubtedly an advantage.

His road — the Philadelphia and Reading — had had the advantage of having Mr. Burton design some of their equipment in the early days of clasp brakes, and it was under his supervision that a great deal of the preliminary detail work was done.

C. D. YOUNG also heartily endorsed everything in the author's paper. He said the Pennsylvania Railroad had and still has quite a large number of steel cars with four-wheel trucks with single-shoe brakes, and they had come to the conclusion that the saving of brake-shoe material would justify their reconstruction; so that, as far as this road was concerned, there would be nothing but clasp brakes on their steel equipment in time.

He emphasized the author's contention that for successful and economical operation of the clasp brake the design and construction must be correct. Without proper design and good workmanship the results with clasp brakes would be disappointing, the decreased shoe pressure would not give the proper efficiency, and the distance in stopping would not be diminished as it should with the use of two brake shoes per wheel. If users of clasp brakes were not obtaining the advantages claimed for them by the author, they should consider carefully the design of their equipment. He knew of installations today which were incorrect and which should be modified to get the advantages expected from the use of two brake shoes per wheel.

Mr. Young thought that some day clasp brakes would be adapted generally to tenders and freight cars where the brake-shoe load for the maximum braking power would exceed 12,000 lb.

O. C. Cromwell (written). In about 1912 the Baltimore & Ohio Railroad, along with other railroads, began to replace wooden passenger-equipment cars with all-steel cars of greater length and heavier weight, and with these cars more or less difficulty was experienced in the proper controlling of the speed and braking of the trains, necessitating very close and frequent adjustment of the brake apparatus on the trucks.

In 1914 the same road again purchased a number of passenger-equipment cars, including 60-ft. steel smokers with 4-wheel cast-steel trucks, 70-ft. steel coaches, combination passenger and baggage, baggage and mail and baggage cars with 6-wheel cast-steel trucks and weighing about 135,000 lb., and 73-ft. dining cars weighing about 157,000 lb. Clasp brakes were applied, and the clasp brake is now considered the standard brake for this class of equipment.

As passenger-equipment cars have become larger, the question of increasing the size of cylinders and the brake apparatus has become quite a serious one. Relief has been found in the use of clasp brakes, in that while an 18-in.-diameter cylinder may be used on cars with a maximum weight of 143,000 lb. with a single-shoe brake-beam type of truck, however, with clasp brakes a maximum weight of 153,000 lb. is allowed.

The clasp brake prevents the rapid accumulation of piston travel. The old-style brake with a standing piston travel of 6 in. has a running travel of 7½ in., a difference of 1½ in., while the clasp brake with a standing travel of 6 in. has a running travel of 6¾ in., a difference of ¾ in. With the decreased piston travel a higher equalizing pressure is attained, which increases the efficiency of a given diameter of cylinder.

The location of the shoe of the clasp brake in relation to the wheel — namely, on the horizontal center line — prevents the rapid accumulation of slack, and consequently piston travel, while the old-style brake beam with the single shoe located some distance below the center line caused rapid wear and accumulation of slack, which was reflected in the piston travel.

With the single-shoe brake beams on the heavier trains on long grades, when the brake shoes were wearing thin the shoes have been found fused to the heads due to the heat generated by the long application of brakes. This condition does not obtain with the clasp brake.

With the clasp brake, journal bearings wear more uniformly and do not crowd to the side, as in the case of single-shoe brakes. It has also been found with the clasp brake that there is less wear on the pedestal jaws and the journal boxes and the shoes, and when starting out from a terminal shoes can be permitted to go on a run worn as much as $\frac{1}{4}$ in. thinner than in the case of the single-shoe application. We feel that on the heavier-passenger-equipment cars the clasp brake will shortly be found in almost universal use.

The reduction in the number of hot boxes on passenger-train cars which has taken place since the application of clasp brakes is not claimed to be due wholly to the clasp brake, but it has contributed its share to the improvement.

C. B. Smith[1] (written). The subject has been so thoroughly covered by the author, who is the "father of clasp brakes," as to leave little either to criticize or add. To the advantages which the author has stated it might be added that the clasp brake has accomplished more successfully that which was previously sought by the adoption of the brake-slack adjuster, *i.e.*, taking up slack due to shoe wear as well as other lost motion in the brake rigging and trucks. The slack adjuster now becomes a necessary adjunct to the clasp brake and will perform its functions as intended.

Our limited experience with the clasp brake confirms all that the author claims. No new passenger equipment will be constructed without this valuable improvement. Moreover, with its adoption upon important train equipments the way is opened for one of the next desirable improvements in brake mechanisms, *i.e.*, that which will justify producing simultaneous application as well as release of brakes on all cars of the train. This can be done with the electric-control device. The use of the clasp brake on all the cars in an entire train will not completely eliminate possible rough handling, because of the time interval between head- and rear-end application.

[1] Mechanical Engineer, Boston & Maine R. R., Boston, Mass.

B. P. Flory (written). The conclusions which are arrived at in the paper are borne out by practical experience of roads which have cars equipped with clasp brakes in service. On the New York, Ontario & Western Railway we have 12 all-steel cars weighing 106,000 lb. with four-wheel trucks equipped with clasp brakes since May, 1914. There has been no trouble experienced in the operation of the brakes on these cars, and we have noticed all of the things to which Mr. Burton calls attention. These cars run in train with other cars which have the single brake, and the difference in riding of the cars when stopped is very noticeable.

As to hot boxes, we find that for a period of 30 months, the cars making in that time a mileage of 1,873,500, there have been 10 hot boxes mostly due to new wheels and axles. During the same period the balance of our passenger cars had 88 hot boxes with a mileage of 11,507,200, or an average of 130,760 miles per hot box against an average of 187,350 miles for the cars equipped with clasp brakes.

Mr. Burton's conclusion concerning the use of clasp brakes in order to stop trains in a short distance should be in the mind of every railroad official when the question of new equipment is raised. This is recognized, for in 1915 the Master Car Builders' Association adopted as recommended practice the rule that all passenger cars with four-wheel trucks weighing 96,000 lb. and over, and all passenger cars with six-wheel trucks weighing 136,000 lb. and over, should be equipped with clasp brakes.

The Author, in his closure, pointed out that one of the most important advantages of the clasp brake is with low-speed trains, where it reduces, if not eliminates, slack action. If there is anything second in importance to stopping trains in a desired distance it is in stopping them smoothly from low speed.

What Mr. Young said on the importance of properly fitting a brake design is another explanation for the absence in the paper of illustrations of the design. He had had occasion to supervise if not to make a great many clasp-brake designs, and it was the rarest thing to find one design suitable for two similar but different designs of cars. With respect to the application of the brake to heavy-equipment freight cars, he said he had just completed designs for some coal cars of 240,000 lb. capacity, weighing about 315,000 lb. when loaded.

No. 1565

MECHANICAL DESIGN OF ELECTRIC LOCOMOTIVES

By A. F. Batchelder, Schenectady, N. Y.
Member of the Society

The purpose of this paper is to bring to the attention of the Society some of the important features in the mechanical design of electric locomotives, with a view to having a more common understanding of the requirements and the methods of meeting them. These features may be listed in the order of their importance as follows:

1 Safety of operation
2 Adaptability to service conditions
3 Reliability in service
4 Convenience of arrangement as affecting safety and efficiency of operation
5 Power efficiency (affected by mechanical design)
6 Service-time factor (ratio of time available for service to total time)
7 Cost of maintenance of permanent way
8 Cost of maintenance of locomotives
9 First cost.

SAFETY OF OPERATION

2 The steam locomotive has been developed by degrees to such a state of perfection that it is common to see it operate at near 80 miles per hour and with perfect safety; but it is seldom, if ever, seen operating at this speed backward.

3 With the coming of the electric locomotive the railroad operator is not content with single-end operation, but must have a locomotive that will operate equally well in either direction. This does not impose any serious difficulties on the design of locomotives which

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operate at speeds under 50 m.p.h., but with locomotives for the higher speeds it presents new problems, or at least it requires the most careful consideration of the running-gear details to obtain the most satisfactory results as to tracking and the effect on the rails and roadbed.

4 The steam locomotive has what now seem to be natural characteristics to allow high-speed operation in one direction. These characteristics are low center of gravity at the front end carried on the center pin of a two-axle guiding truck tending to prevent rolling over, and with a high center of gravity on the rear end and with inside journal bearings allowing the locomotive to roll, thus increasing the time element. The latter thus reduces and distributes the lateral pressure against the rail over a longer distance, and increases the vertical pressure on the rail, thus holding it more firmly in place. These same characteristics can be obtained in electric locomotives by the sacrifice of double-end operation.

5 The advantages gained in operating the electric locomotive in either direction are so important that means should be provided for satisfactory double-end operation. One way of doing this is by using a four-wheel guiding truck at each end of the locomotive. With the use of the extra truck, however, the importance of a high center of gravity largely disappears. The lateral pressure against the rail at the rear end now appears at the truck flanges rather than at the flanges of the driving wheels. The high center of gravity no longer provides the same increased vertical pressure on the outer rail at the point of the maximum lateral pressure. The lateral stresses from guiding the main frame being taken at the center pin of the guiding trucks, the additional vertical pressure on the outer rail is dependent upon the height of these center pins rather than upon the height of the center of gravity of the main frame above the rail head, thus leaving less advantage to be derived from a high center of gravity.

6 To demonstrate more clearly, it is well to see what happens to a locomotive when entering a curve, which is also illustrative of its action on tangent track when oscillating from one side to the other inside of the normal clearance between gage line of track and the wheel flanges.

7 A locomotive having a high center of gravity and with two driving axles guided by a two-axle swivel truck, will serve to illustrate the action. As the locomotive enters the curve, its tendency is to continue on in a straight line, but the flange of the leading wheel gradually comes in contact with the outer rail. This gives the guiding

truck an angular motion about its outer rear wheel and exerts a lateral pressure against the center pin, thus giving the main frame an angular motion around its outer rear wheel.

8 The lateral pressure tending to displace the rail at the leading wheel is the amount required to slip the two inner wheels and to accelerate the truck around its outer rear wheel, plus one-half the amount required to slip the two leading drivers and the rear inner driver and to accelerate the main frame around its rear driving wheel, plus its relative portion of the centrifugal force of the whole locomotive.

9 The lateral pressure tending to displace the outer rail at the rear wheel of the leading truck is the amount of reaction from slipping the two inner wheels and the angular acceleration of the truck, plus one-half of the amount required to slip the two leading drivers and the rear inner driver and to accelerate the main frame around its rear outer driving wheel, plus its relative portion of the centrifugal force of the whole locomotive.

10 The lateral pressure tending to displace the outer rail at the rear wheel of the main frame is the amount of reaction from slipping the two leading drivers, the inner rear driver, and the angular acceleration of the main frame, plus its relative portion of the centrifugal force of the whole locomotive.

11 The greater weight being concentrated at the drivers, and the distance of the truck center pin from the main truck wheels being greater, and the fact that there is but one wheel to take the strain, it follows that the point of the greatest concentrated lateral pressure is at the rear outer driving wheel.

12 The above disregards the important factor of time in the accelerating and centrifugal forces due to the rolling, governed by the height of the center of gravity above the wheel hubs, which tends to reduce the lateral pressure at the rear outer driving wheel. With a high center of gravity above the wheel tread the accelerating and centrifugal forces also tend to tip the locomotive up on the outer driving wheels, relieving the weight from the inner wheels and thus lessening the force required to slip them, and at the same time increasing the adhesion between the outer rail and tie by the additional weight. On good roadbed and rails the locomotive described is capable of being run at above 80 m.p.h. without any apparent bad effect on the track.

13 If this locomotive is operated in the opposite direction, the lateral stresses at these wheels are of the reverse order, the guiding

force now being applied at the driving-wheel flanges and the reaction taken through the center pin to the truck-wheel flanges. The swivel truck, now trailing, is free to oscillate from one side to the other, and the reaction from the force of turning the main frame may be applied at the center pin when the truck-wheel flanges are tight against the inner rail. This allows the force to accelerate the truck as well as the main frame through the gage clearance to the outer rail, thus adding momentum, the value of which depends upon the lateral distance through which the truck is moved; and as the vertical pressure on the rail is limited to the normal weight at the wheels plus the vertical component of the force applied only at the height of the center pin of the truck, the relative lateral to the vertical pressure at the wheels of the truck may be greatly increased. A number of observations have appeared to confirm the fact that the action of the trailing truck above described is one of the most important in producing excessive lateral pressures against the rail in a symmetrically built electric locomotive with similar trucks at both ends.

14 It will be seen, therefore, that while the swivel truck is desirable as a guiding agent at the front end, it is not as desirable at the rear end, and means must be provided to prevent oscillation of the truck and to accomplish the same results that the high center of gravity does in a single-end locomotive.

15 To accomplish these results, it is necessary to reduce the momentum effect and to reproduce the equivalent of the time-element factor and of the increase of vertical pressure on the outer rail that is characteristic of the high-center-of-gravity single-end locomotive.

16 The momentum effect can be reduced by introducing resistance against swiveling, thus restricting the truck from oscillating from one side to the other of the track, the amount of this resistance to be determined by the allowable amount that can safely be applied to the truck when leading. To reproduce the time-element factor, lateral movement can be given to the truck center pin by any of the several methods for giving lateral movement to the leading-truck center pins on locomotives. However, the writer has obtained the best results with the method that is the nearest to constant pressure and dead beat, as it also tends to prevent oscillating.

17 To increase the vertical pressure on the outer rail the center bearing of the truck can be made wide, thus allowing the vertical component of the lateral pressure at the center of gravity to be transferred through the bearing to the wheel; or with the narrow center

bearing the height may be made such that the lateral pressure at that point will result in an increased vertical component independent of the height of the center of gravity.

18 It is the writer's opinion that the double-end locomotive, while its characteristics are different, can be designed for high speed with safety equal to the single-end locomotive, and this regardless of the height of the center of gravity.

ADAPTABILITY TO SERVICE CONDITIONS

19 The electric locomotive, besides being required to operate in either direction, is often also required to be adapted for operating high-speed passenger trains and heavy low-speed freight trains over main-line tracks, to negotiate sharp curves, and to be easy on light track and bridge structures.

20 With locomotives having geared motors, the requirement of operating the passenger and freight trains can often be met by changing the gearing to obtain the proper speed and drawbar pull.

21 The running gear can be made with trucks of short wheelbase coupled together, the number of trucks depending upon the required weight of the locomotive for its maximum drawbar pull, and also on the allowable weight per axle. With such a design curves of very short radius can be operated over and the weight per axle can be such as to allow operation over light structures.

RELIABILITY IN SERVICE

22 When the design is such that it is safe to operate at the required speeds and is proper for the curves and other service requirements, and a liberal factor of safety is provided for the parts subjected to strain, the reliability in service affected by the mechanical part of the locomotive depends mainly upon the bearings, their lubrication, and the method of power transmission from the motors to the drivers. It is necessary therefore to provide effective lubrication and as few bearings and as simple a driving mechanism as the design of the motors will allow.

CONVENIENCE OF ARRANGEMENT AS AFFECTING SAFETY AND EFFICENCY OF OPERATION

23 After providing all the safety appliances recommended by the Interstate Commerce Commission, it is important to arrange for the most convenient location of the operator to allow him an unobstructed view of the track and signals, to place within his easy

reach the air-brake valve and locomotive-signal-device handles, as well as the reverser and power-controller handles, keeping in mind the importance of making them so free from complication that the operator will require the least amount of thought to manipulate any of the devices and be left free to respond to signals and look out for emergencies.

24 The arrangement for housing the electrical apparatus and its position in the cab must be governed largely by its design, but it is important to arrange it so that its operating parts are accessible and easy to inspect, and at the same time are protected against persons coming in contact with any live parts.

POWER EFFICIENCY

25 The power efficiency as affected by the mechanical design is governed largely by the type of the traction motors.

26 It is apparent that the gearless motor mounted directly on the axle allows the design of the maximum efficiency on account of its few bearings and its absence of gearing and moving parts.

27 The gearless motor which is mounted on a quill and driving through springs to the wheels may be considered second in its possibilities for high-efficiency design, it having additional bearings and a greater number of moving parts.

28 The single-reduction geared motor with its additional bearings and gear losses can be given third place in its possibilities for high-efficiency design.

29 The single-reduction geared motor driving through gears and side rods to the wheels may be placed fourth.

30 The gearless motor driving through side rods and jackshaft to the wheels should be placed fifth.

SERVICE-TIME FACTOR

31 The service-time factor is dependent upon the ability of the locomotive to operate under all its service conditions and without undue strains, which requires a liberal design of its wearing parts. In addition to this it depends on the simplicity of its design and the ease with which its parts can be inspected, adjusted, repaired, or replaced.

COST OF MAINTENANCE OF PERMANENT WAY

32 The cost of maintenance of the permanent way is a very important item and can be increased or reduced by the design of the locomotive, and the lowest cost is obtained when the locomotive

meets its service requirements without undue strains, when the rotating parts are balanced, the weights per axle are suitable for the structures, a suitable equalizing system is provided to maintain the proper weight distribution, and when provision is made to protect against flange wear.

COST OF MAINTENANCE OF LOCOMOTIVES

33 The cost of maintenance of the locomotive is dependent upon its safety of operation, its adaptability to service conditions, its reliability, its convenience of arrangement, and the same items that enter into its service-time factor. It is also governed by the same conditions that affect the maintenance of the permanent way.

34 The care with which the material is selected, the quality of workmanship, the ease with which the parts can be inspected, adjusted, repaired or replaced, and the simplicity of the design, are the most important features that govern the maintenance cost.

FIRST COST

35 The first cost of a locomotive will depend largely upon the design chosen, but its importance, except at the time of purchase, becomes of little moment when taking into consideration the eight foregoing features. With two locomotives designed for the same service, the cost of the difference in the efficiency and in the locomotive maintenance alone for one year may, when capitalized, amount to a sum representing a considerable proportion of the first cost of one of the locomotives.

36 The writer feels that too much importance cannot be given to developing to the utmost the mechanical part of the electric locomotive that is the simplest in design and is the highest in efficiency. From the present outlook, the locomotive for high-speed passenger service with the gearless motor, its armature being mounted directly on the axle, and the locomotive for freight and switching service with the single-reduction geared motor mounted on and geared to the axle, lend themselves best to simple design and low cost of maintenance.

37 In making these comments the writer does not wish to be understood as criticizing the work of any other designer. The conclusions he has reached are drawn from experience with his own designs of the various types referred to.

38 In order to verify the conclusions of this paper it would be of interest to see a complete investigation of the different types of loco-

motives that are now operating in the same kind of service at the different railway terminals in New York City, comparing the same on the basis of their first cost, efficiency and cost of maintenance. The differences in the annual operating expense due to these items when capitalized might show a sum that would present a stronger argument for simplicity of locomotive design than can be done in a general and purely verbal discussion such as the writer has presented.

DISCUSSION

C. H. Quereau [1] (written). In the main I agree with the conclusions reached. However, it seems to me very unfortunate that this paper, as well as others recently presented, is a general statement of the conclusions of the author rather than a statement of accomplished results. There should now be available accurate figures based on actual operation, which should not be held back because of mistaken ideas on the part of those who have the facts or those who have the authority to make them public. That the author appreciates this, is shown by the last paragraph of his paper.

In the list of features given in Par. 1 and arranged in the order of their importance, I would add "Maintenance" under Item 4. It would then read "Convenience of Arrangement as Affecting Safety, Efficiency of Operation and Maintenance." It is quite probable that convenience of maintenance was in the author's mind, but it seems to me of sufficient importance to be mentioned specifically.

In the discussion under the heading Safety of Operation, I must confess that I cannot quite follow, probably because I do not clearly understand what was written, or have not the information on which the author bases his discussion. For instance, in describing the action of a locomotive having two driving axles guided by a two-axle swivel truck when entering a curve the statement is made that "The flange of the leading wheel gradually comes in contact with the outer rail, giving the guiding truck an angular motion about its outer rear wheel." I have believed that the angular motion of a four-wheeled truck was about its inner rear wheel, the fact being that when a truck is on a curve the outer-forward-wheel flange and the inner-rear-wheel flange are against the rail, the other two flanges not touching the rail. I do not see, however, that this difference is

[1] Superintendent of Electric Equipment, N. Y. Central R. R., New York.

of particular moment, and bring it up only that the matter may be discussed and clearly understood.

I believe it will be generally agreed that the operating advantages gained by having electric locomotives designed to operate in either direction is of so great importance that means must be found to provide satisfactory designs to meet this condition, notwithstanding the fact that the effect of the trailing engine truck contributes to unstable riding of the engine. The paper proposes to prevent this oscillation by the introduction of resistance against swiveling. This is practicable and has been so demonstrated, but results in increased flange wear, at least when the center of gravity is low.

As to reliability in service, we have had a number of papers describing the operation of electric locomotives on steam railroads, the design of the locomotives and the description of power plants and transmission lines, but reliability in service usually has been overlooked. In eastern territories, especially around the large cities, a delay of a very few minutes will upset the smooth operation of the railroad for hours and the effect of it reach back on the line for 150 miles. The prevention of such delays is worth considerable increase in first cost, and the maintenance methods should be such as to prevent delays practically regardless of cost; it is decidedly poor policy to reduce maintenance costs if by so doing the result is increased delays.

Mr. Batchelder very wisely considers the cost of maintenance of permanent way of more importance than cost of maintenance of locomotives. It is extremely difficult to state definitely what, if any, effect the electric equipment has on the cost of maintenance of permanent way. If the cost of maintenance of way is no greater under electric than under steam operation, such a condition would undoubtedly not be used as an argument against electrification.

As to the cost of maintenance of electric locomotives, the difference in cost-of-maintenance charges at the rate of 3.5 cents a mile and seven cents a mile may be safely figured as not less than $1000 per engine per year. This saving, capitalized, represents a considerable sum, and would warrant an appreciable increase in first cost to secure it.

In discussing a paper on the mechanical design of electric locomotives, we must recognize the fact that there is no common fund of experience or knowledge from which to draw evidence in reaching conclusions as there is concerning steam locomotives. Opinions and theories are essential in designing radically new electric equipment, but only conclusions based on extended service results are authoritative. No one has had the advantage of experience with more than one type, therefore one's conclusions as to other types are based on opinions and theoretical considerations rather than on actual results as shown by service records.

So far as I know, the published data on service results are very limited, either as to first cost or maintenance or reliability in service as shown by train-delay statistics, and therefore each person will place the emphasis on some particular feature, rather than considering results as a whole. If it were possible for this Society, as a neutral, to obtain statistics covering the main points as to results in service, extending over several years, and make them public, it would be of very decided value.

The New York Central electric locomotives are all equipped with bipolar gearless motors mounted directly on the driving axle. The operating results have been completely satisfactory to the officials of every operating department affected. This statement does not include the net financial returns from the investment, which must take into account the item of fixed charges. With the usual maintenance these locomotives ride satisfactorily and without undue effect on the track structure, and are perceptibly more comfortable than steam locomotives. In order to secure these results it is necessary to keep the total lateral motion, both in the boxes and center pins, within three-quarters of the allowable lateral motion on steam locomotives. Table 1 contains statistics which will permit a personal conclusion as to the reliability of these locomotives in service, which will probably be more satisfactory than any expression of opinion.

In this connection I wish to enter a strong plea for the use of Miles per Detention instead of Miles per Minute Detention as the unit in the preparation of statistics by which to judge the reliability of equipment in service and the efficiency of the organization responsible for maintaining it. Including the time element leads only to confusion.

The figures in Table 2 include the cost of inspection and maintenance of all the electric locomotives, both road and switch. In 1912 and 1913 approximately half the total engine mileage and in 1914 and 1915 approximately one-third was that of engines used in switching service. The cost of maintenance of engines in switching service is about twice that of those used exclusively in road service. It follows that the cost of maintaining the road locomotives has been about 2.5 cents per mile and that of the switch engines about 4.8 cents per mile. These engines were not designed for switching service; bearing this in mind, they have given remarkable results.

TABLE 1 TRAIN DETENTIONS DUE TO DEFECTS IN ELECTRIC LOCOMOTIVES[1]

Year	Miles per Detention — All Locomotives		
	Mechanical	Electrical	Grand Total
1912	48,271	103,967	32,965
1913	27,873	86,716	21,093
1914	35,625	57,395	21,981
1915	53,720	107,440	35,813
	Type S Locomotives		
1915	59,583	187,260	45,201

[1] All detentions of two minutes or more included. In 1913 and 1914 there was a total of sixteen Class T locomotives placed in service. In 1912 there were 47 locomotives in service. Since the middle of 1914 there have been 63. Detentions due to man failure, or delays to following trains, not included.

For the first ten months of 1916 the average cost of maintenance of all the electric locomotives has been 2.73 cents per mile. This gives a cost of approximately four cents per mile for the locomotives

TABLE 2 INSPECTION AND REPAIRS OF ELECTRIC LOCOMOTIVES[1]

Year	Cost, Cents per Mile		
	Labor	Material	Total
1912	1.888	1.460	3.348
1913	1.982	1.454	3.436
1914	2.155	2.134	4.289
1915	1.901	1.379	3.280

[1] The above statistics were compiled in accordance with the requirements of the Interstate Commerce Commission. In the year 1914 it was necessary to replace all driving-wheel tires because of unsuitable material, regardless of the extent to which they had been worn. The costs of maintenance have been essentially as above since 1907, omitting 1914.

in the switching service and approximately two cents per mile for those in road service. I expect these costs will not be exceeded for the entire year 1916, but very much doubt that we will be able to keep the maintenance costs permanently at this level.

C. E. Eveleth (written). When an occasion arises to examine critically different designs of electric locomotives there is almost always a tendency, due to the individual's interest in specific features, to concentrate on particular elements and rather superficially consider the locomotive as a whole. In Mr. Batchelder's paper we are fortunate in having a clearly-brought-out presentation of all the essential elements which should be taken into consideration and properly balanced before judgment is passed in favor or criticism made of the mechanical features of a particular locomotive.

The subjects brought out in the paper furnish a measuring rod for which the individual making the examination can arbitrarily assign values to the subdivisions. If corresponding elements are then measured by these values the summation must yield results which cannot be far astray concerning the mechanical design. A number of the elements are intimately related to common features of design, particularly the subjects of Reliability in Service, Service-Time Factor, and Cost of Maintenance of Locomotives, which are all affected directly by the simplicity of parts.

Disregarding other features, the bipolar type of engine with its freedom from all gears, pinions, gear case and motor-armature and motor-axle bearings has as regards these three related subjects a decided initial advantage over all other designs. It also has an unquestioned superiority in mechanical efficiency. In this element there are fundamental general differences which are about of the following magnitudes, the values varying somewhat with the cycle of operation and being generally more adverse to the more complicated structures as the speed increases:

RELATIVE MECHANICAL EFFICIENCIES

	Per cent
Bipolar gearless	100
Quill drive	99
Geared drive (twin gears)	95
Geared to jackshaft and side rods	90
Direct-connected jackshaft and side rods	87

The above list covers practically every type of drive applied to commercial locomotives for heavy duty. The difference in power consumption, due simply to the difference in the mechanical efficiency, can easily be great enough so that the power savings, assuming about 100 miles per day per locomotive, may easily when capitalized amount to from one-third to one-half the original cost of the engines; in other words, to obtain the same overall economic result a material increase in investment in an engine of higher mechanical efficiency is justified, if such investment is necessary to obtain this type of drive.

Mr. Batchelder has been responsibly connected with the design of electric locomotives since their first application to heavy traction. The arguments and conclusions of his paper are based solely on his own personal experiences not only in design but in construction and test. To be more specific, the types of locomotives which have been designed, built and tested under his supervision are as follows:

Quill drive, Baltimore & Ohio	1895
Bipolar locomotive, New York Central	1904
Twin-geared drive, Great Northern	1905
Diagonal side rod, direct-connected drive (experimental)	1911
Geared side rod (experimental)	1912

It is interesting to note that Mr. Batchelder's locomotives were built and tested previous to any existing commercial applications of any of these designs.

All that has been said and all that his paper infers cover only the most desirable type of locomotive considered from a purely mechanical standpoint. If a system of electric application, due to motor-design limitations, necessitates the utilization of side rods or combinations of side rods and gears, this in itself constitutes a material disadvantage for the system considered and must be taken into consideration in an unbiased analysis of the relative values for the various ways of doing the work electrically.

In conclusion, it appears that, considered from the mechanical-design standpoint, Mr. Batchelder's claim for superiority of the bipolar gearless design for high-speed service is founded on the incontrovertible facts that the engine is:

Safe in operation

Superior as to reliability and availability for service, requiring no overhaul periods and requiring minimum inspection time; and that it has the

Lowest cost of maintenance, on account of the elimination of gears, gear case, jackshaft, pin and motor bearings; and

Maximum mechanical efficiency, insuring minimum power consumption.

With Mr. Batchelder's suggestion of the use of a truck center pin located in a well-elevated position, all of the advantages of high center of gravity, so far as effect on rail displacement is concerned, can be obtained. On the other hand, with ordinary leading-truck designs it appears that the high-center-of-gravity designs will give a low-center-of-gravity effect by the action of the rear truck on the track unless the high-center-pin arrangement suggested by Mr. Batchelder is adopted on the trucks. Apparently, if you have high center pins on the leading trucks, the location of the center of gravity is of comparatively little importance. These remarks of course refer to symmetrically designed locomotives intended to run in both directions.

These features do not seem to have had general recognition, as they should place the bipolar gearless locomotive distinctly in a class by itself, and superior on account of these features to every other design. It is therefore to be expected that where the system of electrification will lend itself to the use of this type of engine its application will become very general.

E. B. Katte thought it was obvious that the only difficulty in designing an electric locomotive to ride as well as a steam locomotive was the added requirement of operating at high speed in both directions. The tendency to oscillate and spread the track is due to the fact that a double-ended locomotive is naturally designed symmetrically about the center and by means of cross-equalizing the effect of dissimilar ends is created. He thought the addition of springs over the journal boxes, called hub springs, had done more to establish the easy riding of a symmetrical locomotive than any other one thing. The effect of these springs is to keep the wheels always in contact with the rail. When a slight irregularity in the track is encountered, one wheel leaves the rail and, because of the turning effect due to the motor, the tendency is to skew that axle, which in turn sets up an oscillation of the whole locomotive. Because of the sensitive springs on the top of the journal boxes, the wheel is almost immediately forced back to the rail and the tendency to oscillate is broken up.

This effect was very noticeable on one of the late types of New York Central locomotives. Before the springs were added, it was possible when running at 60 or 65 miles per hour to see the equalizers work, but after the application of the hub springs, the equalizers acted so quickly that the eye could not follow them. Mr. Katte had always believed that it was the application of these springs that created the particularly easy riding of the late type of New York Central locomotives.

L. S. RANDOLPH took up the question of the guiding action of trucks, which he thought would repay study, especially in connection with electric locomotives.

GEORGE L. FOWLER questioned the author's statements regarding the safety of operation. He gave results of his investigations in regard to the lateral thrust of engines and cars upon the track. On the blackboard he made diagrams to show why he disagreed with the author's statement that the rear driver puts an excess pressure on the rail on the outside above that of the other wheels. He also demonstrated by diagrams what thrust is put on the track on a curve by the wheels of various types of engines.

As to the effect of the height of the center of gravity, Mr. Fowler gave his reasons for thinking that the thrust is quite as dependent upon the character of the wheel and of the vehicle as it is upon the center of gravity. In making some investigations in which he had occasion to measure the thrust on trains running from 50 to 60 m.p.h. over an 8-deg. curve, with the track elevated for a speed of 24 m.p.h., he found that the locomotive did not begin to put the thrust on the rail that a sleeping car at the back end of the train did, and yet the height of the center of gravity of the locomotive and of the sleeping car only varied about three or four inches.

The thrust on the tangent track seemed to be a function of the track rather than of the locomotive, for if there was a blow upon any particular part of the track it seemed to make no difference whether the engine was running at 30, 50 or 60 miles an hour, or what type of a locomotive was running over the track — the thrust was invariably at the same spot.

GEORGE GIBBS (written). I take it that Mr. Batchelder's paper is not intended to be more than a very general statement of certain features which must be taken into account in the design of an electric locomotive, together with certain conclusions which he is led to make from his study of the problem. It may therefore be in order for me to give briefly some opinions of my own, and in so doing I am sorry if I must disagree with the conclusions in some respects.

A complete presentation of the subject of electric-locomotive design has yet to be made; all we can do at present is to chronicle experience with various types and classes in different services. Unfortunately, the total number actually in service is quite limited and the period over which our experience runs is also in many cases short.

In the first electric locomotives built not so many years ago the chief consideration was to get a machine which would run and pull a train without continually breaking down through some electrical defect; this is the first step in a young art.

When these conditions were satisfied, it then became a question of obtaining the best design from the standpoints of safety, efficiency and low maintenance cost. This second stage is still under way and the end is by no means yet attained, nor can this be expected any more than has been the case with the steam locomotive, in which there has been a tremendous improvement during the last ten years in increased capacity and efficiency.

Many years' experience in testing work for railroads has brought me into very close touch not only with the design of locomotives but of track. I early became impressed with the importance of high center of gravity in steam locomotives and with the importance of reducing the dead weight below springs to the lowest consistent amount. The track structure does not present a perfect and unyielding plane surface; it has defects, both in alignment and surface, and it is elastic. Its elasticity is a saving characteristic as regards safety and the low maintenance cost of track and equipment. Therefore, a locomotive in running over the track has set up in it oscillations and movements, the effects of which become important not only to the locomotive structure but to tracks, as regards safety and cost of maintenance.

These conditions are well understood in steam-locomotive practice but, unless we are careful, are likely to be lost sight of in designing electric locomotives, where the radical difference in the application of the motive power suggests or permits a variety of wheel arrangements, weight distribution, etc. I cannot here go into a full discussion of this important point, but I can say generally that the elimination of reciprocating parts from the locomotive, a result accomplished in electric locomotives but impossible with steam, does not warrant us in abandoning some of the very important principles well demonstrated in steam practice, namely, that for safe and successful operation under average track conditions, high center of gravity, least dead weight below spring supports and an unsymmetrical wheel and weight distribution give best results. The above applies especially to high-speed operation; for low speeds similar arrangements are also desirable but not so essential.

A word more about the height of center of gravity. There seems to be a tendency to consider that the center of gravity of the machine as a whole is the only consideration of importance, and some electrical designers have been content to secure a fairly high center of gravity by the combination of heavy apparatus in the cab of the locomotive (this being above the springs), with heavy weights carried on the axles or below the tops of the wheels. This arrangement does not give the equivalent of a given height of center of gravity in a steam locomotive (where the bulk of the weight is above springs), as regards the effect on track. A steam locomotive has dead weight only of the driving wheels, axles and boxes; all other weights are spring-borne and thus are not only eased from vertical shock on the track but by the rolling of the mass on the springs tend to effectively relieve lateral rail pressures and convert them into vertical, a condition which conduces greatly to safety and reduction in maintenance both of locomotives and track structure.

The writer, a number of years ago, in taking up the design of electric locomotives for the Pennsylvania Terminal in New York, suggested a method of making an extensive series of experiments to determine the riding qualities of different types of locomotives. The procedure took the form of tests on an experimental stretch of track which was made movable transversely by having the rails mounted on rollers and the transverse motion restrained by stops consisting of hardened-steel points or balls set up against steel strips which, by indentation, register the tendency to displace the track laterally. It was intended by this means to measure the tendency of certain wheel arrangements in locomotives to set up rhythmic side motion, and also to register the throw sidewise by defects in track surface. As the result of these tests we obtained much valuable information regarding design of electric locomotives confirmatory of the general principles I have mentioned above, showing the value of an unsymmetrical wheelbase for an electric locomotive with a high spring-borne center of gravity and low dead-weight component. I believe a more extensive series of these tests, and an acquaintance with the results by electric-locomotive designers, would accomplish a valuable purpose in clearing up many existing differences of opinion.

Mr. Batchelder speaks of wheel arrangement and height of center of gravity as affecting locomotive design; I am unable to follow his reasoning and, if I understand him, I do not agree with his conclusions. He appears to be trying to make out the case that high center of gravity is unnecessary if you have leading trucks for the locomotive, and that the presence of leading trucks makes it immaterial where the motors are mounted; with these conclusions I disagree. Furthermore, he appears to conclude that a leading truck is essential for high-speed operation by stating that steam locomotives cannot be safely operated backward at high speed. It is common in steam practice abroad, and not unusual in suburban practice in this country, to run steam locomotives backward without leading trucks, *i.e.*, the American type of locomotive. In such service steam locomotives are frequently run at quite high speeds on roads having much curvature. Abroad, especially in England, it is not unusual practice to so operate them over long distances at high speeds. The real objections to the operation of steam locomotives backward are, in the first place, that the view of the enginemen is not good; secondly, that the tender is a short-wheelbase structure having a variable and shifting load; and thirdly, that coal dust which is thrown about becomes very disagreeable to the enginemen. However, I believe that leading trucks are useful in locomotive practice, and favor them, but I simply mention the above to indicate that they are not a necessity for safety.

Mr. Batchelder appears to conclude that simplicity, adaptability, reliability and efficiency require that electric locomotives must be standardized in design in the direction of placing the motive power on the trucks, either geared or gearless motors, and would obtain the requisite power by coupling trucks with short wheelbase together, also that this type of locomotive can be used in either high- or low-speed service by simply changing the gear ratio. This procedure is doubtless desirable from the manufacturer's and also from the user's standpoint, if it can be done, as it results in one type of locomotive for any service; but there is no indication, from my experience, that electric motive power puts us any nearer this desirable end than does steam motive power. It certainly seems a mistake for engineers in this early stage of heavy electric-railroading development to come to the hasty conclusion that one standard type of electric locomotive must be advocated for every service, especially when experience in the operation of steam as well as electric locomotives seems to indicate the desirability of combining the elements as I have before indicated to obtain best tracking results.

It must be remembered that track maintenance is a very large item in the cost of operating the railroad, and that track on the average railroad is not always in the best physical condition; therefore the locomotive designer should coöperate with the track department and not proceed independently in developing the best type of locomotive for any particular local conditions. I should put the prime considerations in determining the serviceability of an electric locomotive as follows:

It should have capacity for the given service
Electrically and mechanically it should be operative
It should be least destructive to track, especially when track is not usually kept in perfect surface
Simplicity and low cost of maintenance
Low first cost.

For high-speed service especially I would reverse the order which Mr. Batchelder gives under the heading Power Efficiency, for the different manners of mounting motors and connecting them to the driving axles.

In concluding, Mr. Batchelder suggests that it would be interesting "to see a complete investigation of different types of locomotives that are now operating in the same kind of service of different railway terminals in New York City. . . ." This certainly would be interesting and should be valuable. I have repeatedly urged such a comparison and suggested that it include not only a comparison of the actual upkeep of the locomotives but should be made to bring out the relative overall upkeep of the railroad. The experimental form of track I have before alluded to could probably be made available for experiments by the coöperation of the various railroads interested if we could secure some concerted action looking to the inauguration of a series of tests. We know now, from figures obtained over a number of years, that there is practically no difference in the cost of locomotive upkeep between such widely divergent types, for instance, as the New York Central and the Pennsylvania Terminal locomotives, and it would therefore be useful to bring out clearly by test what other differences, if any, there are in overall adaptability to influence the future design of a form of motive power of growing importance.

The Author. One point I want to bring out is the desirability of having more weight on the rail at the place of the thrust. That place, by actual test, is at the rear truck. That is not from guesswork, or from theory, or from calculation, but is an actual fact, tested and observed for 50,000 miles of running, for the particular purpose of finding out the effect of a double-end locomotive on the track. I have witnessed rails displaced three-quarters of an inch by the rear truck when the leading truck did not appear to move the rail at all. The speed at this time was in the neighborhood of 80 m.p.h.

The double-end locomotives I have seen run did not oscillate from side to side when running on curved track, but would hug one side or the other and ride steadily; on tangent track, however, they would oscillate from side to side, except when the truck was held from swiveling in the track, in which case the locomotive would run steadily just as it did on the curves.

Regarding Mr. Fowler's tests, they seem to agree with the paper, as the steam locomotive with its rigid frame carried directly on springs and with its center of gravity the same height as the car carried on low center pins, did not give as great a thrust. The car would give the effect of a low center of gravity because the side thrust was taken first against the center pin which was relatively low, resulting in less vertical pressure, while the locomotive has a greater vertical pressure due to the side thrust being taken directly and down through the side springs to the rail.

I have never seen any trouble on a curve with double-end locomotives except where too large resistance was used against the trucks curving. A double-end engine, according to my experience, is absolutely smooth and good running on a properly elevated curve.

In regard to Mr. Quereau's remarks as to the pivot point of a truck, I would say that I have not made tests to determine this, but as the outer wheels are heavier, due to the centrifugal and curving forces applied at the center pin, I have assumed that it would pivot about the heavier wheel.

I am gratified to note that Mr. Quereau has furnished figures which bear witness favoring locomotives of the simple and efficient design, and hope that these figures and what he has to say about the operation and maintenance of these locomotives will have careful consideration by those interested. He has said that the introduction of resistance to prevent the guiding trucks from oscillating results in increased flange wear, and I believe that he is right. With the engines he has in mind, there is a frictional resistance of an amount that once the truck has taken a position some external force is required to put it into another position, and it might therefore run cock-eyed on the track and cause flange wear. In that particular case flange wear is much preferable to oscillation.

Mr. Katte has spoken of coil springs on the journal boxes, and I would add that my experience with them has been gratifying.

Mr. Young has understood me to say that a locomotive will not run backward safely. I would rather modify that statement and say that it is not desirable to run it backward. For that matter, I do not know of any locomotives of the American type operating at 80 m.p.h. backward.

Mr. Eveleth has been associated with me throughout a large portion of his experience with electric locomotives, and his discussion favors the conclusion of the paper.

Replying to Mr. Gibbs's written discussion, I would say that Mr. Gibbs seems to agree with me as to the importance of the high center of gravity in steam locomotives; however, it seems to be an inherent condition in steam locomotives, and if it were not desirable it would be difficult and expensive to reduce it any considerable amount.

It is also desirable to have the dead weight below the springs as little as possible, especially in steam locomotives where the unbalancing, due to reciprocating parts, adds relatively large values to the vertical impact. It is not uncommon in well-designed high-speed steam locomotives to have the dead weight on some axles 13,000 or 14,000 lb., and to this is added the effect of the unbalancing, due to the reciprocating parts. A careful analysis of this will show that these values are considerably in excess of the dead-weight values on high-speed electric locomotives of the design suggested in the paper as best adapted for this service which has been built up to the present time. The last locomotives of this type built for the New York Central R. R. to handle 1200-ton trains at 60 m.p.h. had a maximum of 6310 lb. dead weight per axle.

Again Mr. Gibbs agrees with me in that we should not abandon any of the important principles that have been found good in steam practice. I would add, however, providing it is possible to retain the principle in its desirable form. I believe the paper has made it clear that the high center of gravity cannot be used to the same advantage in double-end-operating locomotives with trailing trucks as with a single-end steam locomotive unless special provision is made in the design of the truck, in which case it matters not whether the center of gravity is high or low.

Relative to the unsymmetrical wheel and weight distribution in double-end-operating locomotives, I am unprepared to agree or disagree with Mr. Gibbs, but, offhand, I would say that within the possibilities of locating the weights, the effect would be slight and could be easily overcome by the design of the guiding trucks.

As to Mr. Gibbs's "a word more about the high center of gravity," and how electrical designers secure the high center of gravity, I wish to repeat that it does not make any difference whether the center of gravity be high or low, or how it is obtained, if the locomotive is provided with guiding trucks at each end (the locomotive is guided at its ends and not in the middle), and, as stated in the paper, the trucks can be designed to obtain the results required.

The testing device that Mr. Gibbs describes is extremely interesting and very valuable in determining certain characteristics, such as the transverse movement required in the trucks and measuring the transverse pressures tending to displace the rail, but it does not determine whether or no the locomotive is destructive to the track or safe to operate, as it makes no record of the vertical pressure which affects the adhesion between the rail and tie, which Mr. Gibbs apparently deems important and which I believe is one of the most important features to be obtained.

It is not the intention of the paper to convey the idea that it is necessary to have leading trucks for the successful high-speed locomotive, but to say that with leading trucks the desired results can be obtained.

Relative to operating single-end locomotives backward, I am quite content to leave this with those who have had experience in operating.

Relative to gearing locomotives to obtain satisfactory results for high- or low-speed service, I would say that this is done very effectively on several roads, and the results indicate that it is very good practice.

No. 1566

PULVERIZED FUEL FOR LOCOMOTIVES

By John E. Muhlfeld, New York, N. Y.
Member of the Society

The American interstate railways, as national highways, exercise the right of public franchise and eminent domain. As common carriers their officers and agents are dedicated to the service of the people and obliged to move all of the traffic offered, under identical rates, rules and regulations, and to transport it with safety and as expeditiously and economically as conditions will permit.

2 It therefore becomes essential that these railways shall provide adequate and suitable facilities and equipment, all of which should be maintained in safe and effective working order and utilized to the best advantage for the mutual benefit of the public who pay for the service and the owners who invest their money in the property. During the past decade the cost for money, labor and material entering into the financing, new general construction and equipment, and the maintenance and operation of railways, including taxes, has increased enormously; while the gross operating revenues per passenger and per ton-mile have decreased. For this reason the credit of many properties, which is dependent upon the net profits and the probability of earning capacity, has seriously depreciated.

3 To continue or establish satisfactory credit in order to provide adequate capital at reasonable cost, a steam railway must preserve the proper ratio between gross operating revenues and expenses; and this ratio is largely contingent upon the effectiveness of its developed means for moving traffic.

4 As next to labor the largest single item of cost for transportation is the fuel used in locomotive operation, and as in the final analysis the cost per revenue passenger or per ton-mile is largely conditional upon the capacity, effectiveness and economy of the unit of motive power per hour, it is easy to realize to what extent the credit of a steam railway is controlled by its locomotive performance and expense.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

5 In order to set forth how the use of pulver
in improving steam-railway operation, the followin
sions resulting from a number of years of investig
and of development work in connection with th
subject, may be of immediate interest.

FACTS AND CONCLUSIONS

a The expenditure for locomotive fuel for the
the United States now approximates $300,000,0
which from $75,000,000 to $100,000,000 represer
that is expended to kindle, prepare, clean, and main
when locomotives are standing, drifting or othe
using steam to move themselves, either light or wi

b Changes in the domestic and foreign suppl
well as in the regulations, methods and labor gover
tions, cause a progressive increase in the price per

c Necessity for conserving the limited supply o
exhausting fields, for other than railway fuel p
eliminate this fuel from locomotive service; while
and shortage in supply of the larger sizes and bet
fuels that are in demand for the commercial trade
the use of the less salable general by-product of
verized form, by the railways and industries adjac

d The extraordinary expenditure required for f
charges as well as for combined maintenance and
with the necessity for reliability and flexibility in
clude the general use of electricity either from an
operating standpoint, for the movement of heav
distances.

e Steam locomotives will be equipped to ap
service by the use of pulverized fuel, which in t
smoke, soot, cinders, sparks and fire hazards; red
dispatching at terminals, and stand-by losses; and
mileage by producing longer runs and more nearly
between general repair periods.

f In order to provide the maximum drawba
hour, the steam locomotive must be so improved
tinuity of working steam pressure, greater sustain
increased boiler efficiency, reduced cylinder back
highly superheated steam, all of which can be prod
pulverized fuel.

g The large quantity of steam required by the modern locomotive necessitates excessive rates of evaporation, such as can only be effectively and economically produced by the burning of pulverized fuel, in suspension in order to utilize the heat units that now go out of the stack and into the ashpan when solid fuel is fired on grates.

h By mechanically feeding and burning pulverized fuel, arduous labor on the part of the fireman, such as is now required to shovel ahead and supply coarse coal to grates and to rake and clean fires and ashpans, is replaced by the more skilled manual control of combustion and assistance to the engineer in the operation of the locomotive and observation of track and signals.

i The future steam locomotive, on account of its track and bridge weight and tunnel and overhead clearance limitations, will be required to produce the maximum possible hauling capacity per unit of total weight. As the cylinder horsepower available is entirely dependent upon the boiler horsepower and temperature of superheated steam produced, the use of pulverized fuel to increase the heat value per cubic foot of firebox volume and provide a higher average and more uniform firebox temperature in combination with a reduced front-end or waste-heat temperature, appears to be the most logical means for the solution of the problem.

j The opportunity for reducing the non-productive time of existing locomotives and for relieving terminal congestion that is now caused by the necessity for cleaning fires, ashpans, flues, and smokeboxes; inspecting and repairing draft, grate and ashpan appliances and for firing up and supplying firing tools and equipment to locomotives burning coal on grates, makes the use of pulverized fuel one of the most effective and economical means for increasing the net earning capacity of present single- and double-track steam roads.

HISTORICAL

6 The experimental use of coal dust for fuel is recorded as far back as 1818, but its applications for industrial purposes in the United States commenced in cement kilns in 1895, when the high price for fuel oil brought about this change.

7 The Manhattan Elevated Railroad in New York City, the Swedish Government Railways and various other steam roads have done some experimental work during the past 20 years in the burning of coal and peat dust for locomotive purposes; but, so far as is known, it has not resulted in regular train-service operation such as has been performed by the equipment described in this paper.

COMBUSTIBILITY OF PULVERIZED FUEL

8 As with all mediums now used in the most advanced and progressive engineering practice for producing mechanical power, such as naphtha, gasoline, kerosene, crude and fuel oils, compressed air, storage batteries and electricity, there is a certain element of danger in the use of pulverized fuel that does not obtain with the more ineffective coarse coal.

9 However, there are now certain established rules and regulations governing the manufacture, storage, handling and use of pulverized fuel, which make it comparatively easy to avoid trouble, where ordinary care is exercised, as confirmed by the records of industrial-plant operations.

10 Where pulverized fuel has been stored in dry, air-tight and fireproof bins and kept below a temperature of 150 deg. fahr., no cases of spontaneous combustion are known to have occurred.

SPECIFICATIONS FOR PULVERIZED FUEL

11 From investigations to date, any solid fuel that, in a dry pulverized form, has two-thirds of its content combustible, is suitable for steam-generating purposes.

12 Domestic and steam sizes and qualities of anthracite, bituminous and semi-bituminous coals, and lignite and peat, as well as the inferior grades such as anthracite culm, dust and slush, and bituminous and lignite slack, screenings and dust, are all suitable for burning in pulverized form.

13 To produce the best results these fuels should be mechanically dried and milled so that they will be of about the same dryness and fineness as portland cement; or so that the moisture will not exceed 1 per cent, and that 95 per cent of the total will pass through a 100-mesh screen and 85 per cent of the total will pass through a 200-mesh screen. This applies to anthracite as well as to bituminous coals.

EQUIPMENT AND COST FOR PREPARING PULVERIZED FUEL

14 As over 8,000,000 tons of pulverized fuel are now being used annually in the United States for industrial kilns and furnaces, it is not thought that the equipment or process for preparing pulverized fuel requires any comment.

15 The total cost to prepare pulverized fuel properly in a suitably equipped drying and pulverizing plant will range from 15 to 45

cents per ton, depending upon the capacity of the plant. For a railway coaling station of average capacity this total cost will be less than 25 cents per ton, an item which will be more than offset by the difference in the cost on the tender of the grades of coal purchased for pulverizing and those that would be required for burning satisfactorily on grates.

STORAGE AND HANDLING OF PULVERIZED FUEL

16 The raw fuel should be dried, pulverized and stored in metal or other fireproof-material containers in such quantities as are needed to supply the demand. After pulverizing it should not be exposed to open lights or the atmosphere, and the production of light "dust clouds" should be avoided. Pulverized fuel should be handled with the same care as fuel oil.

EQUIPMENT FOR SUPPLYING PULVERIZED FUEL TO TENDERS

17 This consists of one or more overhead storage bins equipped with suitable means for supplying 15 tons of pulverized fuel to a tender in 3 or 4 minutes without exposing the fuel to the atmosphere during its conveyance, or producing any dust clouds.

18 Complete control of the fuel flowing to and of the air exhausting from the tender must be maintained, and this can only be done by installing special equipment suitable for the requirements.

EQUIPMENT FOR BURNING PULVERIZED FUEL

19 The illustrations at the end of this paper convey a general idea of the equipment that has been found essential for burning pulverized fuel in a steam locomotive.

20 The particular factors that have been kept in mind in the development of this apparatus are:

a To produce equipment that will be readily applicable to either new or existing steam locomotives of standard designs.

b To simplify and standardize the various details and make them interchangeable for the different types and sizes of locomotives.

c To apply all possible operating equipment, in a self-contained manner, to the tender fuel tank; eliminate complicated mechanism for conveying fuel from the tender to the engine, and remove all special apparatus except fuel and air-supply control levers from the cab.

d To eliminate the necessity for any manual handling of fuel, fire or ashes in the operation.

e To insure positive control over the fuel feed, in order quickly to meet all conditions of road or terminal operation, and to provide for quick firing up, free steaming, perfect combustion, regularity of boiler pressure, uniform firebox temperature and maximum capacity of boiler with the minimum heat loss.

f To place the entire regulation of combustion under three hand-control levers in the cab, *i.e.*, fuel feed, air supply, and induced draft (the latter when locomotive is not using steam).

g Provide an arrangement of refractory furnace that will insure ready accessibility to all parts of firebox for inspection and maintenance.

h To insure a supply of dry fuel under all conditions of weather.

i To eliminate the necessity for firing tools, such as scoops, rakes, hoes, slash bars and grate shakers, as well as the glare, heat effect, and lowering of firebox temperature and draft from furnace-door openings.

j To minimize the noise and dust in the cab.

k To reduce engine-house facilities and delays and expense incident to building, preparing, cleaning and dumping fires and hostlering locomotives.

l To make the pulverized-fuel-burning and storage equipment on the engine and tender readily convertible for the use of fuel oil.

PROCESS FOR BURNING PULVERIZED FUEL

21 The process whereby pulverized fuel is burned in locomotives equipped with the described apparatus may be briefly stated as follows:

a Suitable conveying, feeding, commingling and delivering apparatus for the fuel and air.

b Induction of the fuel and air mixture and generation of combustible gas in the preliminary chambers.

c Induction of combustible gas and perfection of complete combustion in the furnace proper.

d Induction and diffusion of the products of combustion through the furnace and boiler.

OPERATION OF PULVERIZED-FUEL-BURNING EQUIPMENT

22 For firing up a locomotive the usual steam blower is turned on in the stack, a piece of lighted waste is then entered through the

firebox door opening and placed on the furnace floor, just ahead of the primary arch, after which the pressure fan and one of the fuel and pressure-air feeders are started. From 45 to 60 min. is ordinarily sufficient to get up 200 lb. of steam pressure from boiler water at 40 deg. fahr.

23 After firing up, the regulation of the fuel and air supply is adjusted to suit the standing, drifting or working conditions, the stack blower being used only when the locomotive is not using steam.

24 The operation of the fuel-burning equipment is as follows:

25 The prepared fuel, having been supplied to the enclosed fuel tank, gravitates to the conveyor screws, which carry it to the fuel and pressure-air feeders, where it is thoroughly commingled with the pressure air and carried by it through the connecting hose to the fuel and pressure-air nozzles and blown into the fuel and air mixers.

26 Additional induced air is supplied in the fuel and air mixers, and this mixture, now in combustible form, is induced into the furnace by the smokebox draft.

27 The flame produced at the time the combustible mixture enters the furnace obtains its average maximum temperature (from 2500 to 2900 deg. fahr.) at the forward combustion zone under the main arch, and at this point auxiliary air is induced by the smokebox draft to finally complete the combustion process.

28 The liquid ash runs down the under side of the main arch and the front and sides of the forward combustion zone of the furnace and is precipitated into the self-clearing slag pan, where it accumulates and is air-cooled and solidified into a button of slag which can be readily dumped.

29 As each of the fuel and pressure-air feeders has a range in capacity of from 500 to 3000 lb. of pulverized fuel per hour, and as from one to five of these may be easily applied to the ordinary locomotive tender, there is no difficulty in meeting any desired boiler and superheater capacity.

30 The uniformity with which locomotives can be fired is indicated by the fact that the regularly assigned firemen can maintain the steam within a variation of 2 lb. of the maximum allowable pressure, without popping off.

CHEMISTRY OF THE COMBUSTION OF PULVERIZED FUEL

31 Fuel as ordinarily burned on locomotive grates consists principally of bituminous and anthracite, a percentage of soft coal usually being mixed with the latter.

32 Owing to the limited firebox space, which frequently necessitates the burning of as high as 75 lb. of anthracite and of 150 lb. of bituminous coal per square foot of grate surface per hour, the strong draft (from 10 to 18 in. of water in the smokebox) necessary to supply sufficient air through the grates for combustion causes enormous losses through unburned gases and fuel that are exhausted from the stack and of fuel that is carried into the smokebox and ashpan.

33 For the best results coal should be sized to about 3-in. cubes for burning on locomotive grates, but as this is now quite impracticable, due to the methods of mining and the cost, a mixture of fine and large coal is usually supplied, which tends to burn irregularly and results in a reduction of boiler capacity and efficiency.

34 Generally speaking, it is necessary to break up any fuel to such uniform size that the oxygen in the air can unite perfectly for combustion. A deficiency in this respect results in some portions of the fuel passing off as unburnt hydrocarbons and other portions being left as incompletely burned coke. It is equally important that the proper quantity of air should be admitted to the furnace, as any insufficiency or excess lowers the efficiency. For example, the preventable fuel loss with 8 per cent of CO_2 in the stack gases will be about 10 per cent, with 12 per cent of CO_2 it is reduced to about 3 per cent, and with 16 per cent of CO_2 there is practically no waste.

35 As a 1-in. cube of coal exposes but 6 sq. in. of area for absorbing oxygen and liberating heat, but when pulverized to the proper fineness will expose from 20 to 25 sq. ft. of area for oxidation, the first essential for complete combustion is the breaking up of the fuel into dry, minute and uniform particles. Then by diffusing these so that each may be surrounded with the right quantity of air for complete combustion it will be possible to burn practically all of the available combustible, regardless of the percentage of non-combustible.

36 The principal fuels adaptable for use in pulverized form in locomotives are anthracite, semi-anthracite, semi-bituminous and bituminous coals and lignite and peat. (Pulverized-fuel-burning locomotives are also readily convertible for the use of fuel oil.)

37 These fuels differ more in physical characteristics than in chemical composition, but as the carbon and hydrogen content are the most valuable elements and determine the calorific value, they are usually taken into account for classification purposes.

38 Many fuels also contain a considerable percentage of oxygen,

which diminishes the calorific value, while sulphur which is frequently present gives out heat during the combustion process.

39 The union of the carbon, hydrogen or sulphur of the fuel with the oxygen of the air is accompanied by the evolution of heat, and the products of this combustion are water, carbon monoxide, carbon dioxide, carbon in the form of soot, and sulphur dioxide, all of which are gaseous. The remaining constituents of the fuel are left as ash, having combined with more or less oxygen.

40 The value of a fuel is determined by the amount of heat that it will generate when burned, and this in turn is dependent upon how much combustible carbon, hydrogen, hydrocarbons and sulphur and how much non-combustible moisture, ash, oxygen and nitrogen it contains.

41 Of the non-combustibles, ash, which usually contains a mechanical mixture of silica, alumina, iron, lime, potassium, sodium, and magnesium, is the most detrimental. Like moisture, it is anti-calorific, and furthermore it acts as an obstruction to the flow of air and gases, reduces boiler capacity and efficiency, and incurs heavy expense and delays in the cleaning of fires and ashpans and in the final disposition of the ash.

42 The "clinkering" and "honeycombing" of ash is one of the worst troubles to be contended with in the combustion of coal, and its formation may be either chemical or by fusion.

43 Clinker is of two kinds, "hard" and "soft." "Hard clinker" is formed by the direct melting of some of the ash content. It hardens as it forms and usually gives but little trouble. "Soft clinker" is formed by the slagging of the ash and is either pasty or fluid and steadily grows in size. "Honeycomb" or "flue-sheet clinker" is formed by the condensation or coking of tarry matter or vapor as it strikes against the firebox sheets, and results in the accumulation of a relatively soft, light, ashy substance that grows or spreads over certain of the refractory or metal parts of the furnace.

44 With the use of pulverized fuel the usual difficulties resulting from the formation of hard and soft clinker on grates are eliminated, but with fuels containing certain intrinsic combinations of ferrous silicates which fuse at comparatively low temperatures (2000 to 2300 deg. fahr.) the honeycomb formation will result when the proper air-supply and combustion conditions do not obtain to produce ferric silicates, which fuse at relatively high temperatures (2500 deg. fahr. and above). For example, during the process of combustion, ferric sulphide (FeS_2), commonly known in fuel as iron pyrites, is reduced

to ferrous sulphide (FeS) as the result of the chemical reduction illustrated by the following formula:

$$FeS_2 = FeS + S.$$

45 As ferrous sulphide (FeS) melts at a comparatively low temperature (2138 deg. fahr.), it may surround itself with fuel and ash and form a pasty mass which may act as a binder to collect other ferrous sulphide (FeS), fuel and ash, all of which may tend to collect on and adhere to the hottest portions of the firebox sheets, such as staybolt heads, flue beads, and like parts which are higher in temperature than the melting point of ferrous sulphide (FeS) and the surrounding metal surfaces, while the temperature of the latter may be lower than the melting point of the ferrous sulphide (FeS).

46 The following formula shows the result of *incomplete combustion* owing to *insufficient air:*

$$2\,FeS + 2\,O = 2\,FeO + 2\,S.$$

47 By providing *sufficient air* through an excess supply, the following formula shows the result of *complete combustion:*

$$2\,FeS + 9\,O = Fe_2O_3 + 2\,SO_3.$$

48 For this latter process an oxidizing atmosphere must at all times obtain in the firebox to prevent the reduction of ferric sulphide (FeS_2) to ferrous sulphide (FeS), as expressed in the first formula.

49 The ferrous sulphide (FeS), as has been shown, is the direct cause of honeycomb, for the reason that it produces ferrous oxide (FeO), which unites with the silica to form a honeycomb that is very fusible at temperatures over 2400 deg. fahr.; whereas by the production of ferric oxide (Fe_2O_3), in combination with the silica present, a highly infusible clinker is formed.

50 As a general rule an increase in the percentage of silica, alumina and magnesium in the fuel matter will tend to decrease, while an increase in the percentage of iron, lime, potassium and sodium in the fuel matter will tend to increase, the fusibility of ash, but in every case a relatively high percentage of ferrous oxide (FeO) resulting from an insufficient supply of air for combustion will be accompanied by honeycomb formations that will tend to adhere to various parts of the firebox.

LOCOMOTIVE PERFORMANCE

51 For the fiscal year ended June 30, 1914, the Interstate Commerce Commission reports a total of 64,760 locomotives of all classes

in the United States having made a total of 1,755,972,325 miles. This gives an average for each locomotive owned of about 27,115 miles per annum, 74 miles per day, or but little over 3 miles per hour.

52 From the foregoing figures it is easy to imagine that over one-half of the time of locomotives is now spent at terminals in the hands of the transportation and mechanical departments, and that most of this delay is due to the necessity for cleaning fires, ashpans, flues and smokeboxes; inspecting and repairing draft, grate and ashpan appliances; and for firing up and supplying firing tools and equipment. Frequently the delays to locomotives waiting to reach ashpit tracks and to rekindle fires exceeds the time required to do this work, and during the interim much fuel is needlessly consumed and the boiler subjected to excessive contraction and expansion.

53 With pulverized fuel a locomotive having the boiler filled with cold water may be brought under maximum steam pressure within an hour and the fuel feed then stopped until it is called for service. When standing or drifting at terminals or on the road the fuel feed can also be discontinued, as the steam pressure can always be quickly raised. After the trip or day's work the locomotive can be immediately stored or housed, the usual ashpit delays being entirely eliminated. The possibilities for increasing the productive time of existing locomotives and for relieving terminal congestions that are now brought about by the necessity for cleaning and rebuilding fires on grates, makes the use of pulverized fuel one of the most attractive and quickest methods for increasing the earning capacity of present single- and double-tracked steam railways.

RESULTS OBTAINED FROM USE OF PULVERIZED FUEL

54 From the actual operation of steam locomotives in regular train service, the use of pulverized fuel has demonstrated in particular the practicability of eliminating smoke, cinders, sparks and fire hazards; increasing drawbar horsepower per hour per unit of weight; improving the thermal effectiveness of the steam locomotive as a whole; reducing non-productive time at terminals; utilizing otherwise unsuitable or waste fuels; eliminating arduous labor; providing greater continuity of service and producing more effective and economical operation and maintenance.

55 The following performances (Table 1) of a ten-wheel type of freight locomotive, rated at 31,000 lb. of cylinder tractive power,

with 69-in.-diameter driver wheels, when used in fast through-freight service on runs of from 91 to 138 miles in length for the purpose of testing various fuels under identical adjustment conditions, may be of interest:

TABLE 1 PERFORMANCES OF TEN-WHEEL-TYPE LOCOMOTIVE

Item	Pulverized		
	1 Bituminous	2 Bituminous	3 Bituminous
Fuel:			
Fineness, per cent through 200 mesh	0.85	0.85	0.85
Moisture, per cent	0.40	0.81	0.59
Volatile, per cent	24.72	36.27	24.36
Fixed carbon, per cent	68.43	58.29	65.05
Ash, per cent	6.85	5.44	10.59
Sulphur, per cent	1.96	0.68	0.84
B.t.u., per lb.	14,739	14,334	13,912
Miles run, total	1,324	426	398
Cars per train, average	61	65	60
Adjusted tonnage per train, average	1,719	1,808	1,759
Speed when train was in motion, miles per hour, average	26	25	24
Boiler pressure when using steam (200 lb.), average	198.3	193.5	194.9
Front-end draft when using steam, in. of water, average	7.15	7.79	6.69
Firebox draft when using steam, in. of water, average	3.50	3.22	3.18
Temperature of steam, deg. fahr.	562	573	555
Coal fired per hour of running time, lb. (average)	3,275	3,063	3,457
Adjusted ton-miles per lb. of coal (average)	12.84	13.97	11.59

56 The locomotive was worked at its maximum capacity on all trips, about 10 per cent more tonnage being hauled than usual for like locomotives burning coal on grates, and at practically fast-freight schedule speed. The exhaust-nozzle opening was about 25 per cent larger than the maximum for hand firing.

57 The general results were excellent, particularly as regards tonnage, speed, combustion, and steam pressure, the latter being maintained at full speed with injector supplying the maximum amount of water to the boiler.

58 With the highest-sulphur coal (No. 1) and the highest-ash coal (No. 3) there was less than 1 cu. ft. of slag in the slag box at the end of each run, and practically no collection of ash or soot on the flue or firebox sheets. In fact, with the No. 3 fuel there was less than 2 handfuls of slag, ash and soot collected on each trip.

59 The steam railways in the anthracite-coal-mining district generally use for their locomotive fuel mixtures which will run from 25 to 50 per cent of bituminous and the balance of anthracite pea and buck sizes which will pass through a $\frac{7}{8}$-in. and over a $\frac{5}{16}$-in. round opening. As anthracite coal is very low in volatile, ignites slowly, and is a poor conductor of heat, the bituminous mixture is used to overcome the trouble this causes when the smaller sizes must be burned on grates, and even then it necessitates the use of unusually small exhaust nozzles to create sufficient draft.

60 In the experiments with pulverized anthracite fuel for locomotives the idea has been to utilize the grade of coal of lowest commercial value, such as birdseye, which is of a size that will pass through a $\frac{5}{16}$-in. and over a $\frac{1}{16}$-in. round opening, as well as the refuse called culm or slush, which passes through the $\frac{1}{16}$-in. round opening and is usually wasted in the washery water or used for back-filling the mines. To reclaim this slush a couple of wooden bins were installed, through which the washery water could be finally passed for the collection of the solid matter.

61 The analyses of the various fuels used may be approximated as given in Table 2.

TABLE 2 ANALYSES OF FUELS USED IN EXPERIMENTS

Item	Pulverized		
	Bituminous Run-of-Mine	Anthracite	
		Birdseye	Slush
Moisture, per cent	0.50	0.50	1.00
Volatile, per cent	29.50	7.50	6.00
Fixed carbon, per cent	60.00	77.00	71.00
Ash, per cent	10.00	15.00	22.00
Sulphur, per cent	1.50	1.00	2.5
B.t.u. per lb.	13,750	12,750	11,250
Fineness, per cent through 200 mesh	86.00	86.00	86.00

62 At the commencement of the development work the locomotive was equipped with an arrangement of refractory baffles and fuel and air inlets for burning 100 per cent bituminous coal, and after this had been properly accomplished successive adjustments were made to burn the following mixtures, the last of which is now being used with as satisfactory results as the 60 per cent bituminous and 40 per cent birdseye:

First: 75 per cent Run-of-Mine Bituminous and 25 per cent Anthracite Birdseye

Second: 67 per cent Run-of-Mine Bituminous and 33 per cent Anthracite Birdseye

Third: 60 per cent Run-of-Mine Bituminous and 40 per cent Anthracite Birdseye

Fourth: 60 per cent Run-of-Mine Bituminous and 40 per cent Anthracite Slush

Fifth: 50 per cent Run-of-Mine Bituminous and 50 per cent Anthracite Slush

Sixth: 40 per cent Run-of-Mine Bituminous and 60 per cent Anthracite Slush

63 Further work along this same line will determine just how great a percentage of anthracite slush can be used to the best advantage, but the evaporative results so far obtained, *i.e.*, about 7 lb. of water from feedwater temperature per lb. of coal, indicates that considerably more than a 60 per cent anthracite-slush mixture may be utilized. This accomplishment not only means a decrease of 25 per cent in the cost per ton for locomotive fuel, but also the release of a large tonnage of commercial anthracite, which is becoming more scarce and in greater demand each year.

64 The principal trouble to be overcome has been on the intermittent runs, as it is more difficult to maintain proper combustion with a slow fire and to re-ignite the fuel after the feed has been stopped for a time, with the low- than with the higher-volatile coals.

65 The same increase can be made in the size of the exhaust-nozzle openings (about 25 per cent) for anthracite as for bituminous coal when burning in pulverized form, as compared with hand firing of coal on grates.

66 The development of sufficient drawbar pull in a Consolidation type of freight locomotive with 63-in.-diameter driver wheels, rated at 61,400 lb. of cylinder tractive power, to haul a freight train of 23 loaded cars (representing about 1562 actual tons) over a ruling grade of 1½ miles of 1.65 per cent grade with a 6-deg. curvature, further indicates the advantages of sustained boiler horsepower in combination with reduced cylinder back pressure, which is only made possible by this method of stoking and burning fuel.

67 The average results of a number of trips made by an Atlantic type of passenger locomotive, rated at 21,850 lb. of cylinder tractive power, with 81-in.-diameter drive wheels, when used in high-speed

passenger service on round-trip runs of 171 miles in length, may be stated as follows:

TABLE 3 PERFORMANCE OF ATLANTIC-TYPE PASSENGER LOCOMOTIVE

Fuel Used	
Kentucky unwashed screenings:	
Fineness, through 200 mesh, per cent	83
Moisture, per cent	2.46
Volatile, per cent	36.00
Fixed carbon, per cent	54.00
Ash, per cent	7.94
Sulphur, per cent	0.79
B.t.u. per lb.	13,964
Locomotive Performance	
Miles run	171
Running time, hours	3.87
Train, number of cars	5.8
Train, tonnage	291
Speed, miles per hour	44.2
Drawbar pull, pounds	2,711
Horsepower per hour	319.5
Fuel used, tons	3.82
Water used, gallons	8,381
Fuel per h.p.-hour, lb.	6.17
Water per h.p.-hour, lb.	56.48
Evaporation, water per lb. of coal, lb.	9.15
Evaporation from and at 212 deg. fahr., lb.	11.1
Boiler efficiency, per cent	77

68 The combustion results may be indicated by the smokebox-gas analysis given in Table 4.

TABLE 4 SMOKEBOX GAS ANALYSIS, TEST OF TABLE 3

Pounds of coal burned per hour	Per cent of		
	CO_2	CO	O
3067	14.5	0.0	4.5
3498	15.2	0.0	2.8
3931	15.2	0.0	4.0
4000	16.0	0.4	2.6

69 This locomotive could be fired for the round trip with a variation of not over 2 lb. in the boiler pressure, and the size of the exhaust nozzle used was 5½ in. in diameter and the temperature of the superheated steam averaged about 635 deg. fahr. for steam of 185 lb. boiler pressure and the smokebox gases about 460 deg. fahr., although maxi-

mum temperatures of 715 deg. fahr. for superheated steam and of 482 deg. fahr. for smokebox gases were recorded.

70 Furthermore, this pulverized-fuel-burning Atlantic-type locomotive, with 21,850 lb. tractive power and 81-in.-diameter driver wheels, performed on these runs the identical service of the regularly assigned hand-fired coal on grate-burning Pacific-type locomotives with 27,900 and 33,700 lb. tractive power, and 69-in.-diameter driver wheels, both types begin equipped with superheaters.

NOTES ON PERFORMANCE

71 From tests made with pulverized lignite having an analysis of about 1.8 per cent moisture, 47 per cent volatile, 41 per cent fixed carbon, 9.5 per cent ash, and 0.75 per cent sulphur, and a heating value of 10,900 B.t.u. per lb., in regular passenger-locomotive service, the same satisfactory results were obtained as with bituminous coals, the combustion and operating being entirely smokeless, sparkless and cinderless, and the steam pressure being fully maintained.

72 With pulverized fuel the control of the fuel feed and thereby of the over- or under-production of steam is nearly perfect. A locomotive can be fired up and the fuel consumption then stopped until a few minutes before starting time. At the end of the run, or when drifting, the fire can be extinguished at will and quickly re-ignited without any special equipment or materials. A locomotive with boiler full of water and 185 pounds of steam pressure, after standing 11 hours, without fire, still had 80 pounds of steam pressure.

73 Comparative tests made between similar locomotives in the same service resulted in the use of 2775 lb. of lump coal, hand-fired, to get up steam and for terminal handling and dead time, as compared with 1569 lb. of pulverized screenings to produce the same result, or an increase of over 75 per cent. The greatest saving is in the firing up alone, this requiring 1700 lb. of lump coal as compared with 750 lb. of pulverized screenings, or an increase of over 225 per cent.

74 In the engine-house terminal handling there is the least possible delay and expense. No more time or facilities are required than for fuel-oil-burning locomotives. A locomotive fired up at 6 A.M. can leave with its train at 7 A.M., and upon arrival at the destination engine house can be immediately fueled, watered and housed, the slag pan being dumped over the engine-stall pit.

75 Delays incident to building and preparing fires and cleaning ashpans outbound, and for waiting to get on ashpits and to inspect

and clean fires, grates, smokeboxes and ashpans inbound are entirely eliminated, as is also the necessity for subjecting fireboxes and flues to the chilling effect of cold air due to standing around and hostlering.

76 When in pulverized form, the preparation and handling of fuel on locomotive tenders is avoided. It is always dry and never mixed with rain or snow. No fuel is lost or wasted, and there is no necessity for sizing, shoveling or pushing ahead on tender, or for the use of extra labor or firing doors, shakers, or tools for handling the fuel, or for manipulating the fire, ashes or cinders.

77 Through the possibility of enlarging exhaust-nozzle openings from 25 to 50 per cent as compared with the areas required for burning coal on grates or fuel oil, the full benefit of expenditures for improved cylinders, valves and valve gears, particularly in connection with cylinders of large volume, can now be obtained. Heretofore the necessity for maintaining relatively small exhaust-nozzle openings to produce the required firebox draft has enabled but little benefit to be gained from improved steam distribution, as cylinder back pressures of from 15 to 30 lb. when operating at maximum capacity of engine and boiler are not at all uncommon in some of the most recently built stoker-fired single-expansion locomotives. As every pound of cylinder back pressure saved is equal to at least 2 lb. added to the boiler pressure when a locomotive is working at its maximum capacity, and further provides freer movement and less wear, tear and fuel consumption, the benefits to be derived are obvious.

78 As the limiting factor of a steam locomotive is, or should be, the ability of the boiler to produce steam, the rate and effectiveness of combustion become the controlling factors. When coal is burned on grates a rate of about 50 lb. of run-of-mine, and of about 60 lb. of lump bituminous coal, per sq. ft. of fire surface per hour is the maximum allowable for the greatest boiler efficiency. However, as this limits the rate of consumption to a total of from 3000 to 6000 lb. per hour for the average modern locomotive of great power, and as the actual coal supplied to the firebox by mechanical stoking frequently reaches a rate of 150 lb. per sq. ft. of grate area, or a total of from 9000 to 15,000 lb. per hour, the boiler efficiencies frequently run as low as from 55 to 45 per cent and even less. Therefore the necessity for eliminating grates if much over 12 lb. of water per sq. ft. of evaporating surface per hour is to be obtained with reasonable efficiency.

79 From results established during the past six months, the quantity of live steam required for the operation of pulverized-fuel-burning equipment when the locomotive is being worked at its maxi-

mum boiler-horsepower capacity, is about 1½ per cent of the saturated steam generated, which is considerably less than what is required for the steam-jet operation of mechanical stokers when firing coal on grates, and very much less as compared with what is used in the generally existing steam-jet practice of burning fuel oil; this latter amount, according to reports made by the U. S. Naval Board, is approximately about 6 per cent of the total steam generated, exclusive of the reduction of efficiency in combustion due to the evaporation of steam into hydrogen and oxygen and back into H_2O in the firebox.

80 Comparing the use of pulverized fuel and fuel oil for steam-locomotive purposes, it may be stated that with pulverized coal at 13,750 B.t.u., costing $2.35 per ton, and fuel oil at 19,500 B.t.u., costing $2.75 per 100 gal., an amount of at least $2.50 must be expended for the fuel oil necessary to perform the same useful work as will obtain from $1.00 expended for pulverized fuel.

81 A diagram showing the thermal efficiency of electric and steam motive power under different operating conditions is presented in Fig. 23. The top portion of this diagram, in which 100 per cent of the total maximum capacity or load factor is assumed as utilized, is obviously an ideal condition and one which never obtains in actual service, the condition in which 50 per cent is utilized being more nearly the average for steam-road operation. With electrical operation this load factor seldom exceeds 35 per cent. It will be noted that under these conditions pulverized fuel appears as extremely advantageous. These figures do not consider any emergency power plant or storage-battery equipment for electrical operation.

PRELIMINARY EXPERIMENTS WITH STATIONARY BOILERS

82 The development work pertaining to the use of pulverized fuel for locomotives has been carried along in direct conjunction with the use of like fuel in one 463-h.p. (nominal rating) Stirling type of stationary boiler, various tests being made for the purpose of determining the best combination of fuel and air admission, flameway, and draft and furnace construction for the maximum boiler capacity and efficiency consistent with minimum renewal of refractory materials.

83 Both bituminous and anthracite fuels have been used, the principal work being in connection with the latter on account of the greater difficulty in maintaining combustion due to the low volatile content. Table 5 will give some idea of the results accomplished in the combustion of the more inferior grades of clear anthracite.

84 The analysis of the anthracite birdseye coal used in pulverized form was from 6.85 to 7.63 per cent volatile and from 18.68 to 21.53 ash. The analysis of the anthracite slush used in pulverized form was from 5.88 to 9.84 volatile and from 22.16 to 46.45 per cent ash.

85 As anthracite slush is not burned on grates, no comparison is available for the pulverized performance. However, comparing the performance of anthracite birdseye hand-fired on grates equipped with forced blast below and induced draft above the fire, with the

TABLE 5 TESTS ON STIRLING TYPE BOILER WITH POWDERED FUEL

Test No.	1	2	3	4	5	6	7
Duration, hours	72	336	24	48	120	240	24
Horsepower rating	463	463	463	463	463	463	463
Horsepower developed, per cent	133	135	147	178	112	118	124
Fuel:	Anth.	Anth.	Anth.	Anth.	Anth.	Anth.	Anth.
Kind	B.E.	B.E.	B.E.	B.E.	Slush	Slush	Slush
Dryness, per cent	0.65	0.65	0.65	0.65	0.8	0.8	0.8
Fineness, per cent through 200 mesh	86.0	86.0	85.0	86.0	88.0	86.0	88.0
Evaporation, from and at 212 deg. fahr., lb.	8.7	8.9	9.6	9.8	7.8	8.1	8.5
CO_2, average per cent	16.6	16.3	15.9	16.6	16.2	16.5	16.7
Vacuum in breeching uptake, in. of water	0.25	0.23	0.22	0.33	0.27	0.28	0.27
Vacuum in combustion chamber, in. of water	0.16	0.14	0.13	0.16	0.17	0.19	0.15
Boiler pressure, average lb.	140	142	141	140	143	144	145
Flue-gas temperatures, deg. fahr., average	518	525	496	603	475	580	576

same fuel pulverized and burned in suspension, there is an average increase of over 40 per cent in the evaporation in favor of the latter.

86 As the percentage of CO_2 indicates that the preventable fuel loss through improper combustion is practically nil, the advantage of this method of burning solid fuel, which has been found to require the minimum amount of excess air to insure complete combustion, is obvious.

87 In general, it may be stated that the use of pulverized anthracite slush will double the steam-generating capacity of boilers now burning birdseye anthracite hand-fired on grates, and at the same time eliminate fire cleaning, greatly decrease the amount of ash to be handled, and reduce the boiler-plant-labor cost about 40 per cent.

88 Furthermore, with the pulverized fuel the boiler pressure can be more readily maintained or increased or reduced to meet the requirements, and when one or more of the boilers are not needed

temporarily the fuel feed can be stopped and started at will, thereby eliminating the necessity for maintaining banked fires and burning fuel when not required in order to have the boilers ready for instant use.

89 An investigation of the culm banks in the anthracite-coal-mining district would undoubtedly disclose many millions of tons of domestic and steam sizes of fuel that can be reclaimed, and in addition, the large percentage of slush that would be produced in this process could all be utilized in pulverized form for power-generating purposes.

ADVANTAGES OF PULVERIZED FUEL

90 The performance of locomotives in regular road passenger and freight service during the past year has demonstrated that the burning of solid fuels in pulverized form in suspension has the following detailed advantages over the burning of like solid fuels in the usual steam sizes on grates:

(1) Eliminates the necessity for electrification to provide smokeless, cinderless, sparkless, and sootless operation.

(2) Saves from 20 to 40 per cent in fuel of equivalent heat value fired.

(3) Increases boiler efficiency from 10 to 15 per cent.

(4) Reduces non-productive time of locomotives and of engine, train and yard crews.

(5) Maintains maximum steam pressure and sustains boiler capacity without popping or smoke at all times under absolute control of the fireman, with minimum fuel consumption, whether the locomotive is working at full capacity, drifting, stopping, starting, or standing at terminals, stations, or on sidings.

(6) Snow, rain, wind, slipping and like factors do not effect steaming or fuel consumption.

(7) Stops the practice of "slugging" in firing, as the feeding of excess fuel has the same effect on steam generation as an insufficient supply.

(8) Eliminates smoke inspectors and avoids fines for violating smoke ordinances.

(9) Eliminates expensive ash and cinder pits and labor and car service incident thereto, and dispenses with the cleaning of fires or ashpans outbound from the engine house.

(10) Eliminates front-end diaphragm, table and deflector plates, nettings, handholes and cinder hoppers, and the inspection, repairs and renewals of these parts.

(11) Eliminates scoops, shovel, rake, hoe, slash bar, sparking bar, squirt hose, firedoor, grate, grate shaker, coal pusher, ashpan, ashpan damper and like firing equipment, and the inspection, repairs and renewals of these.

(12) Insures clean boiler and superheater flues, superheater elements and front end. It prevents cutting out of parts such as result from impinging action of cinders and ashes passing through them.

(13) Heat is equally distributed and diffused over all firebox sheets, which increases their evaporative capacity.

(14) Less excess air is required for combustion. This reduces the amount of non-productive air to be passed through the boiler and results in increased heat value per cubic foot of firebox volume, higher average and more uniform firebox temperature, and reduced smokebox or waste-heat temperature.

(15) Minimum amount of moisture contained in or conveyed by the fuel, as fired, reduces wasteful effect from that heat in the products of combustion which is below the temperature of the water or steam in the boiler.

(16) Higher average temperature of superheated steam, and superheat temperature is more quickly raised after starting the locomotive.

(17) Increase of from 25 to 50 per cent in exhaust-nozzle area reduces the cylinder back pressure and noise and results in smoother working of the locomotive.

(18) Reduced front-end vacuum and the automatic induction of combustible mixture into the firebox and of products of combustion through the furnace and boiler toward the stack opening eliminates any destructive impinging action on firebox sheets or brickwork.

(19) Elimination of the ashpan and grates permits of placing large-diameter driving wheels below the firebox and of adequate heating surfaces, superheater elements, and firebox volume for the production of maximum tractive power and boiler capacity without the necessity of using trailer wheels for freight locomotives.

(20) Eliminates the 10 to 30 per cent in weight of combustible that is found in the ash and clinker from grate-fired locomotives.

(21) Permits the selection of firemen for skilled rather than for arduous labor, thereby ultimately raising the standard of both firemen and engineers.

(22) The fireman has the same opportunity as the engineer for observation of track and signals, thereby reducing liability for accident.

(23) Reduces time and facilitates firing up and avoids the necessity for using special fuel or equipment for that purpose.

(24) Permits the utilization of existing refuse and of such grades of fuels as cannot be readily disposed of by mine operators in the commercial trade and which now make up from 35 to 70 per cent of the mine output necessary to produce the domestic readily salable sizes. The possibilities of making use, in pulverized form, of the low-grade coals in the West, which now have to be passed over as unsatisfactory for locomotive use, will be readily recognized, as well as those of using dust, screenings, slack, culm and refuse which accumulate from all mining operations.

(25) A solid cubic inch of coal exposes only six square inches for the liberation of heat, while in pulverized form a cubic inch of coal exposes from 20 to 25 square feet, making possible a much more uniform production of gas.

(26) Permits of operating locomotives for relatively long, continuous mileages or time periods, as there are no grate fires to require cleaning.

(27) As no fuel whatsoever is supplied to the furnace by hand and grates are eliminated, there is no liability for "holes in the fire," "banking" and "clinkering" to cause waste of fuel or steam failure.

(28) The enclosed fuel container prevents spilling and loss of fuel, and it is protected from snow, rain, mud and other unfavorable conditions.

(29) More uniform furnace temperature and distribution of heat reduces the liability for firebox and flue leakage, and the greater retention of heat from the arrangement of refractory material prevents chilling and quick loss of steam pressure when fuel supply is cut off.

(30) As over one-half of the time of road freight locomotives (exclusive of general repair-shop time) is now spent at terminals in the hands of the Transportation and Mechanical Departments, this will be greatly reduced by the elimination of fire, ashpan and flue cleaning, and the time for firing up and inspection will also be reduced.

(31) No special fuel or equipment is required for firing up and fuel will re-ignite from heat in the brickwork within an hour after being turned off.

(32) No cinders or ashes are produced to cement self-draining ballast or to burn out wooden cross-ties and trestles.

(33) Maintains the steam locomotive on its relatively low first cost and expense for fixed-charge basis, and further reduces the cost for maintenance and operation of large units of power.

ILLUSTRATIONS SHOWING THE APPLICATION OF POWDERED-FUEL APPARATUS TO LOCOMOTIVES

Reference to the halftones and diagrams which follow will clearly set forth the principal reductions to practice and results obtaining at this time in the use of pulverized fuel for steam locomotives. Typical milling plants are shown as well as locomotive equipment. Owing to the exacting service requirements, in combination with the multiplicity of qualities and grades of anthracite, bituminous and lignite fuels, it has been found of great advantage to carry along experimental work on stationary boilers, three installations of which are illustrated, equipped for this purpose.

FIG. 1.—RAW-COAL STORAGE BIN (ABOVE), WITH DRYER AND DRYER FURNACE (BELOW), WITH INTERMEDIATE CONNECTION

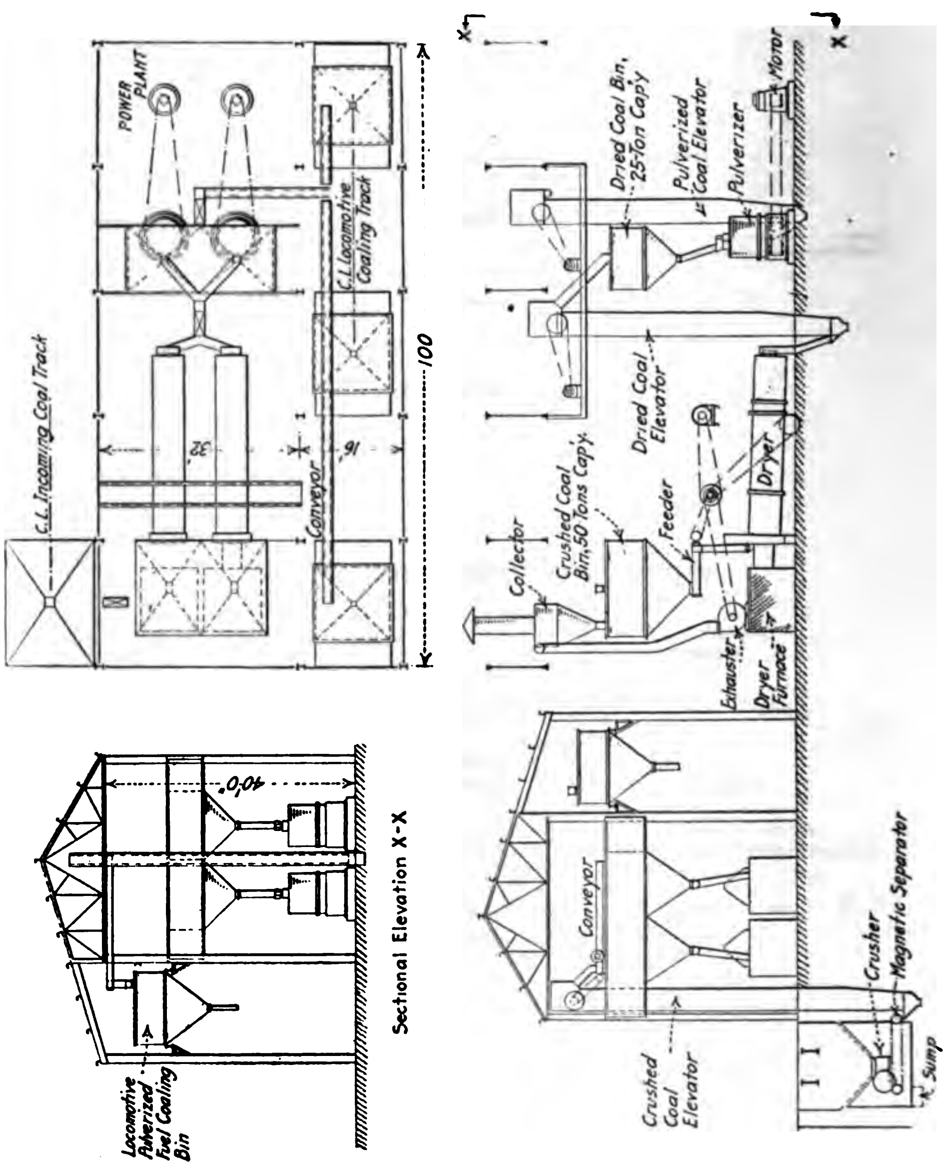


Fig. 4 Double-Unit Milling Plant, 16-Ton per Hour Capacity, and Triple-Bin Locomotive Coaling Station for the Central Railway of Brazil

FIG. 5 MILLING PLANT ANNEX AND LOCOMOTIVE COALING PLANT LOCATED OVER YARD TRACKS TO THE RIGHT

FIG. 6 THE DELAWARE AND HUDSON COMPANY CONSOLIDATED LOCOMOTIVE No. 1200 TAKING PULVERIZED COAL AT OLYPHANT, PA., COALING STATION

A. Exhaust-air line between fuel container and top of fuel-storage bin to permit of exhausted air displaced by fuel entering tender.
B. Fuel outlet from storage bin.

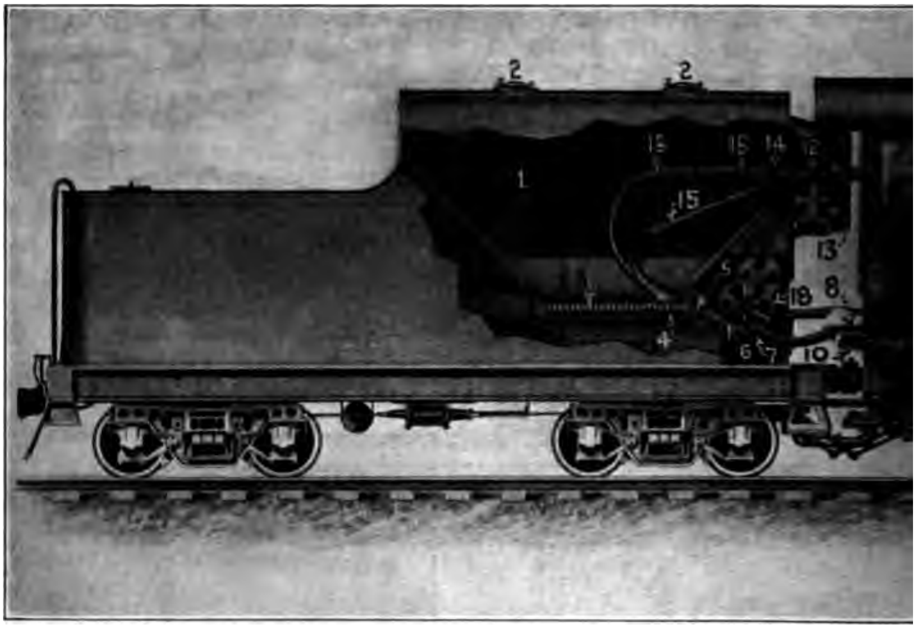
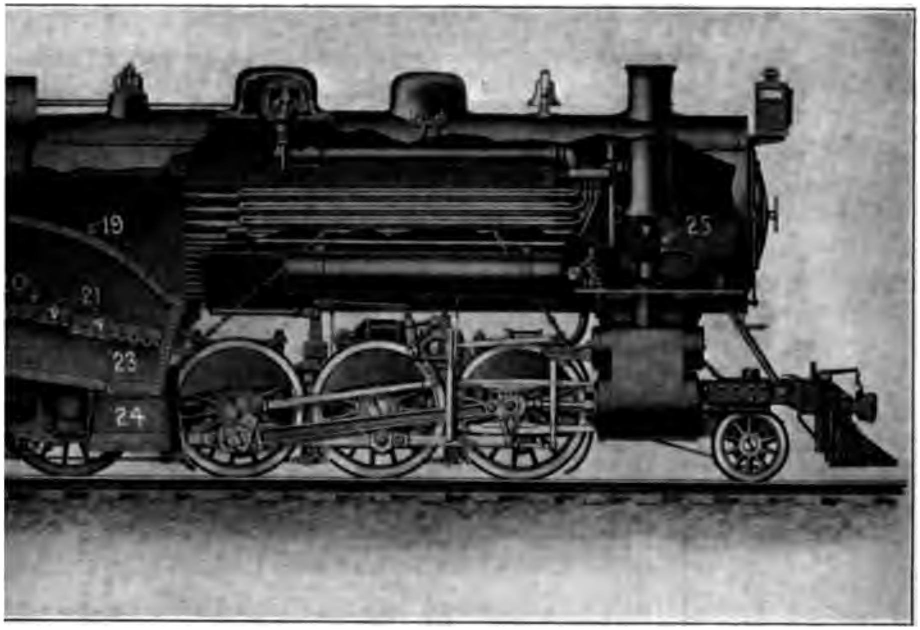


FIG. 7 TYPICAL APPLICATION OF PULVERIZED-FUEL-BU

1 Enclosed Fuel Container
2 Fuel Supply Inlets and Covers
3 Fuel Conveyor
4 Fuel and Pressure-Air Feeder
5 Fuel and Pressure-Air Commingler
6 Fuel and Pressure-Air Outlet
7 Fuel and Pressure-Air Flexible Conduit
8 Fuel and Pressure-Air Nozzle
9 Fuel and Air Mixer
10 Induced-Air Inlet Damper
11 Control for Induced-Air Inl Damper
12 Pressure Blower
13 Constant-Speed Steam Turbine f Pressure Blower
14 Pressure-Blower Manifold
15 Pressure-Blower Conduits



MENT TO CONSOLIDATION TYPE OF STEAM LOCOMOTIVE

Variable-Speed Steam Turbine for Fuel Conveyor, Feeder and Commingler

Control for Steam Turbine for Fuel Conveyor, Feeder and Commingler

Operating Gear, Shaft and Clutches for Fuel Conveyor, Feeder and Commingler

19 Main Arch
20 Primary Arch
21 Auxiliary Air Inlets
22 Primary Combustion Chamber
23 Final Combustion Chamber
24 Self-Clearing Air-Cooled Slag Pan
25 Engine and Turbine Exhaust Nozzles and Stack Blower

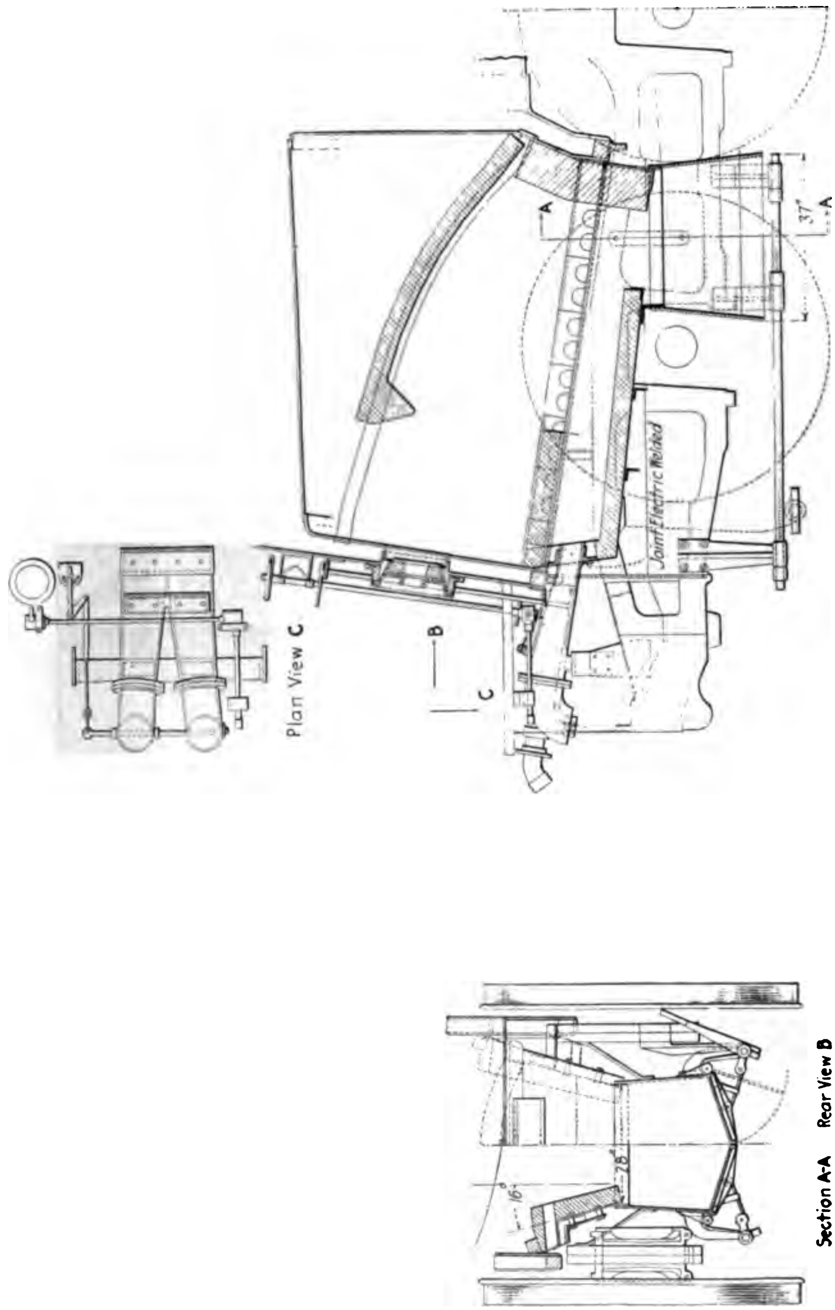


Fig. 8 Double-Burner and Firepan Equipment as Applied to 10-Wheel Type of Passenger Locomotive for the Central Railway of Brazil

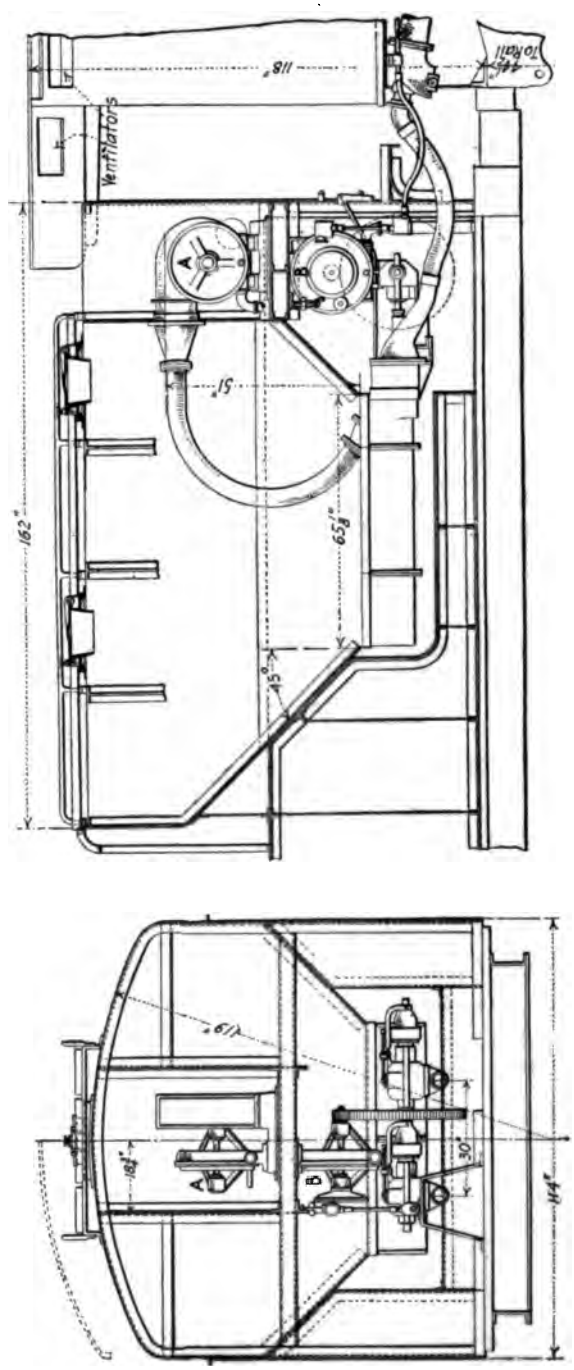


Fig. 9 Double-Feeder Equipment as Applied to Tender for 10-Wheel Type of Passenger Locomotive for the Central Railway of Brazil

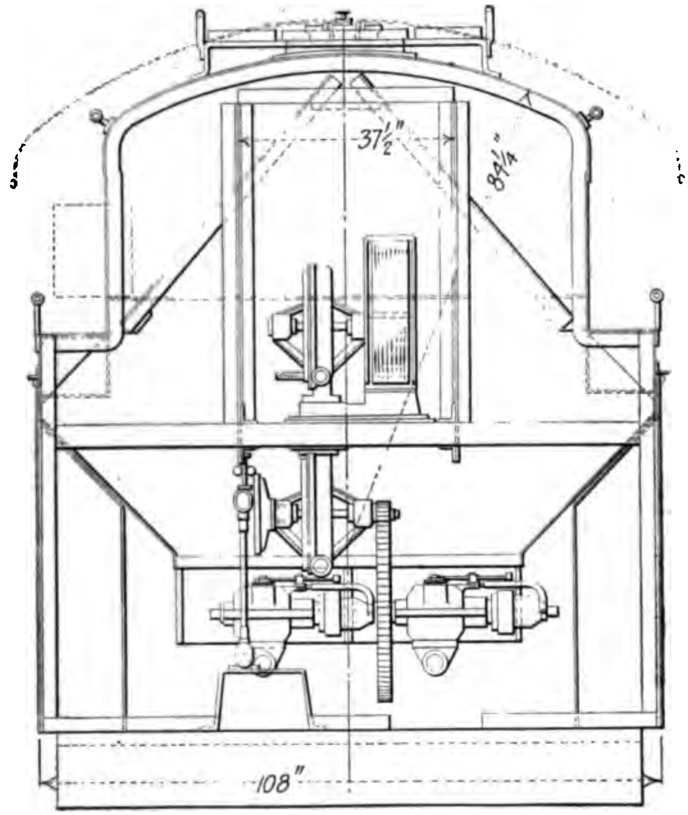


FIG. 10 DOUBLE-FEEDER EQUIPMENT AS APPLIED TO TENDER FOR 6-WHEEL COUPLED TYPE OF SWITCHING LOCOMOTIVE FOR THE NEW YORK CENTRAL RAILROAD

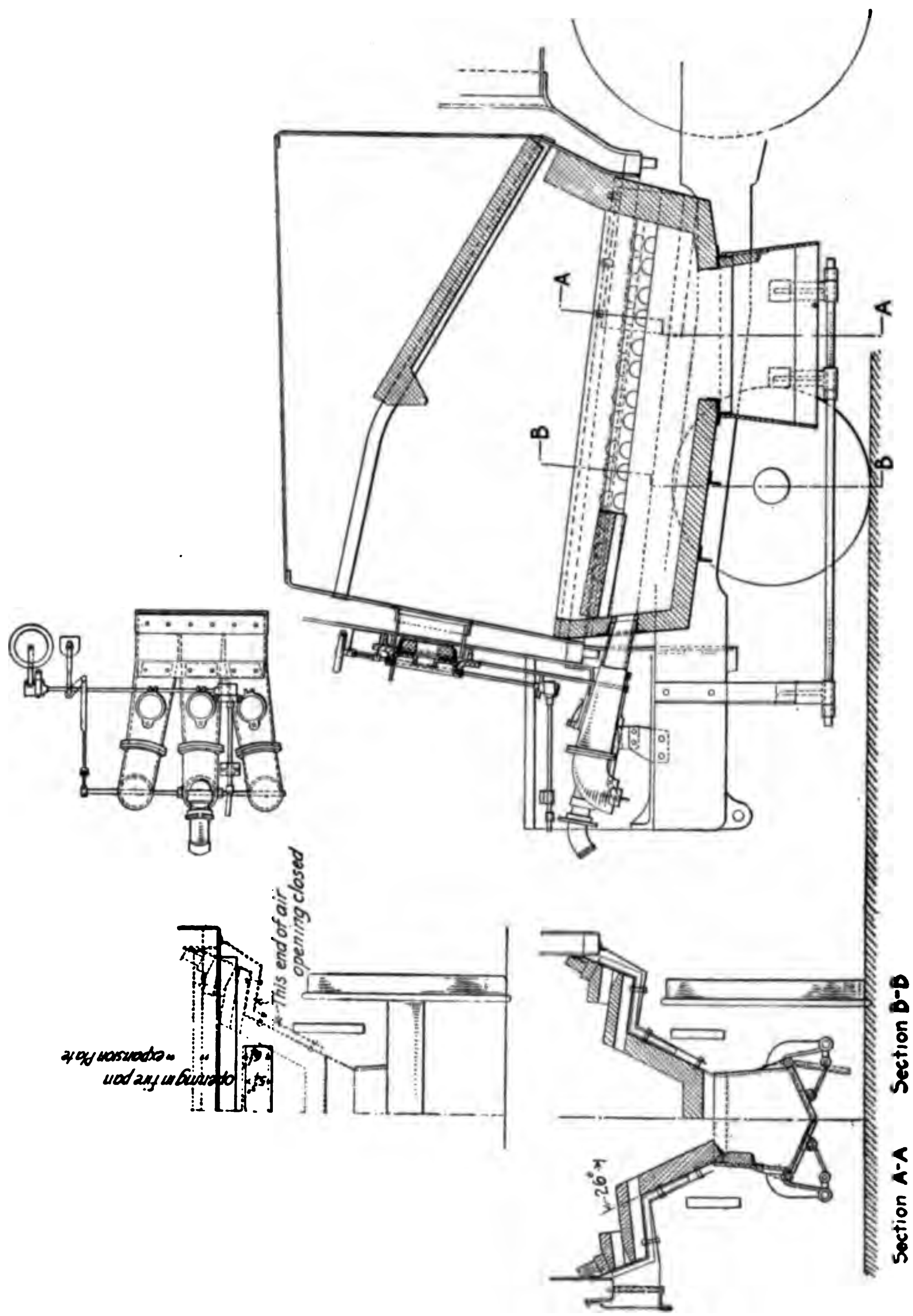
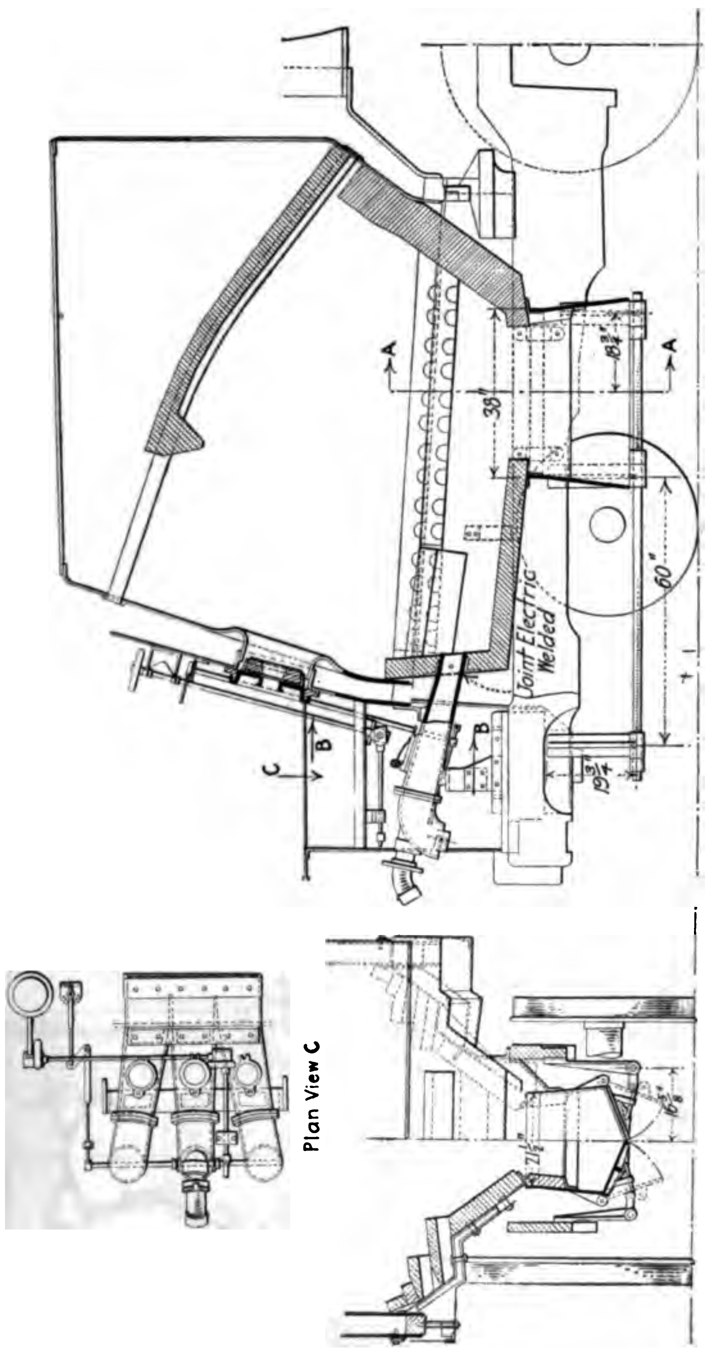


FIG. 11 TRIPLE-BURNER AND FIREPAN EQUIPMENT AS APPLIED TO PACIFIC TYPE OF LOCOMOTIVE FOR THE NEW YORK CENTRAL RAILROAD



Section A-A   Rear View B

Plan View C

Fig. 12 Triple-Burner and Firepan Equipment as Applied to Mikado Type of Locomotive for The Atchison, Topeka & Santa Fe Railway

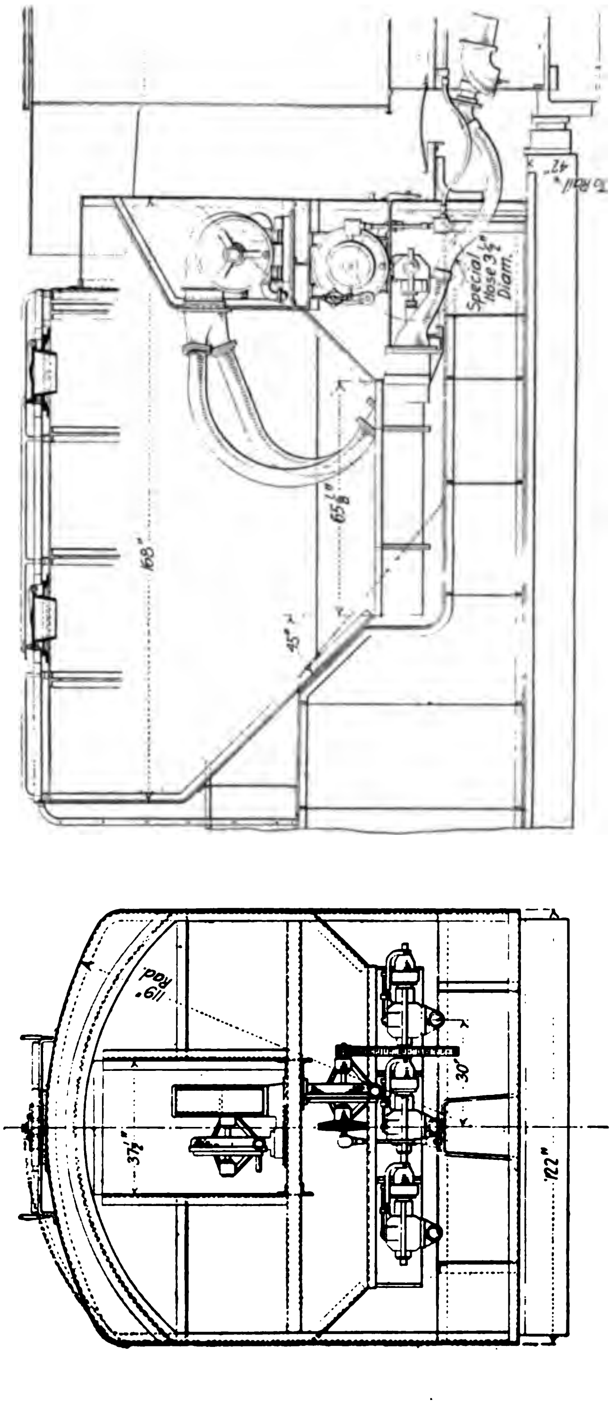


FIG. 13 TRIPLE-FEEDER EQUIPMENT AS APPLIED TO TENDER FOR MIKADO TYPE OF LOCOMOTIVE FOR THE ATCHISON, TOPEKA & SANTA FE RAILWAY

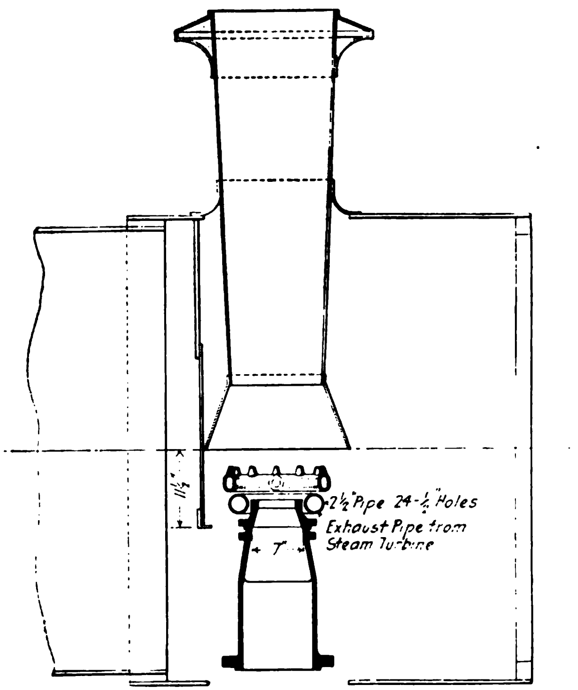
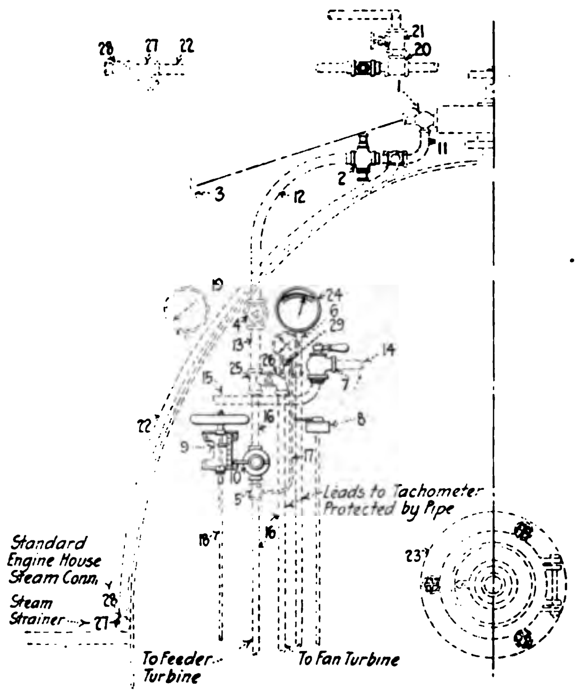


Fig. 14 General Arrangement of Locomotive Front End for Burning Pulverized Fuel



ITEM	NAME
1	1¼" ANGLE VALVE
2	PRESS. RED. VALVE 1¼" INLET-1¼" OUTLET
3	EXTENSION HANDLE
4	1½" GLOBE VALVE
5	1"x1"x¼" TEE
6	300 LB DUPLEX GAGE
7	1½" QUICK OPEN. BLOWER VALVE
8	DAMPER CONTROLLER VALVE
9	FEEDER CONTROL
10	GRAD. PRESS. RED. VALVE 1" INLET-1" OUTLET
11	1¼" DIAM. PIPE
12	1½" " "
13	1½" " "
14	1¼" " "
15	1¼" " "
16	1" " "
17	GAGE CONNECTION
18	¾" ROD
19	BOILER GAGE
20	1¼"x1¼"x1¼" TEE
21	1¼" GLOBE VALVE
22	1¼" DIAM PIPE
23	FURNACE DOOR
24	TACHOMETER
25	1½"x1"x1" TEE
26	1" STREET ELL ¼" OUTLET
27	1¼" STEAM STRAINER
28	STD. ENG. HOUSE STEAM CONNECTION
29	DOUBLE RADIATOR SYPHON AND VALVE

Fig. 15 Arrangement of Cab Equipment Used by Fireman for Operating Pulverized-Fuel-Burning Apparatus

FIG. 16 FINAL COMBUSTION CHAMBER AT FRONT OF FIREPAN AND FIREBOX

FIG. 17 FUEL BURNERS AND PRIMARY COMBUSTION CHAMBER AT REAR OF LOCOMOTIVE FIREPAN

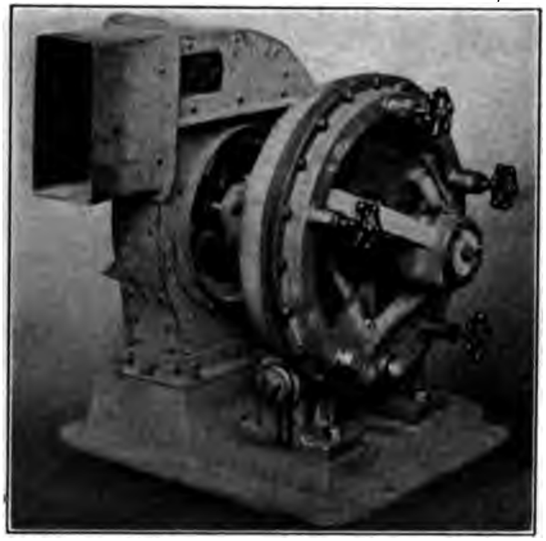


Fig. 18 Direct-Connected Turbo-Blower Equipment

Fig. 19 Variable-Speed Steam Turbine for Control of Fuel-Feeding Mechanism, Showing Special Governor for Varying Speed of Turbine

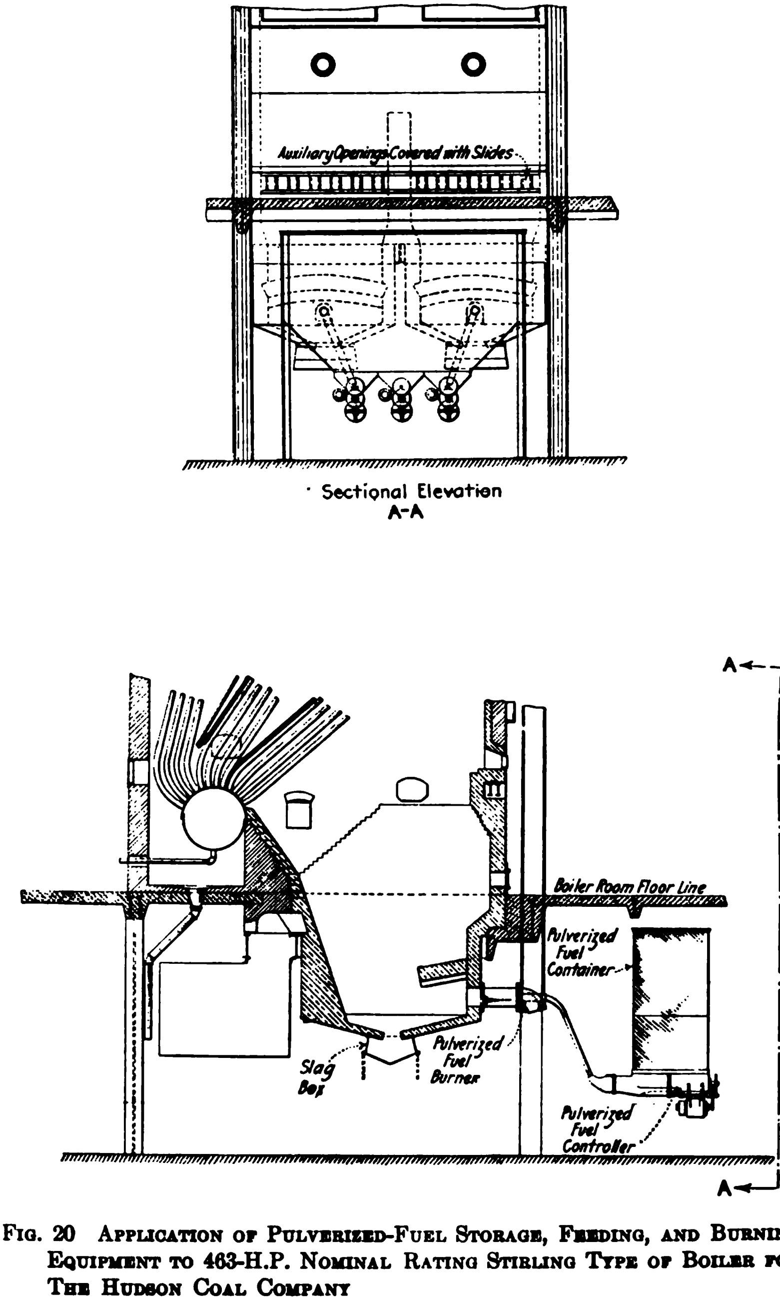


Fig. 20 Application of Pulverized-Fuel Storage, Feeding, and Burning Equipment to 463-H.P. Nominal Rating Stirling Type of Boiler for The Hudson Coal Company

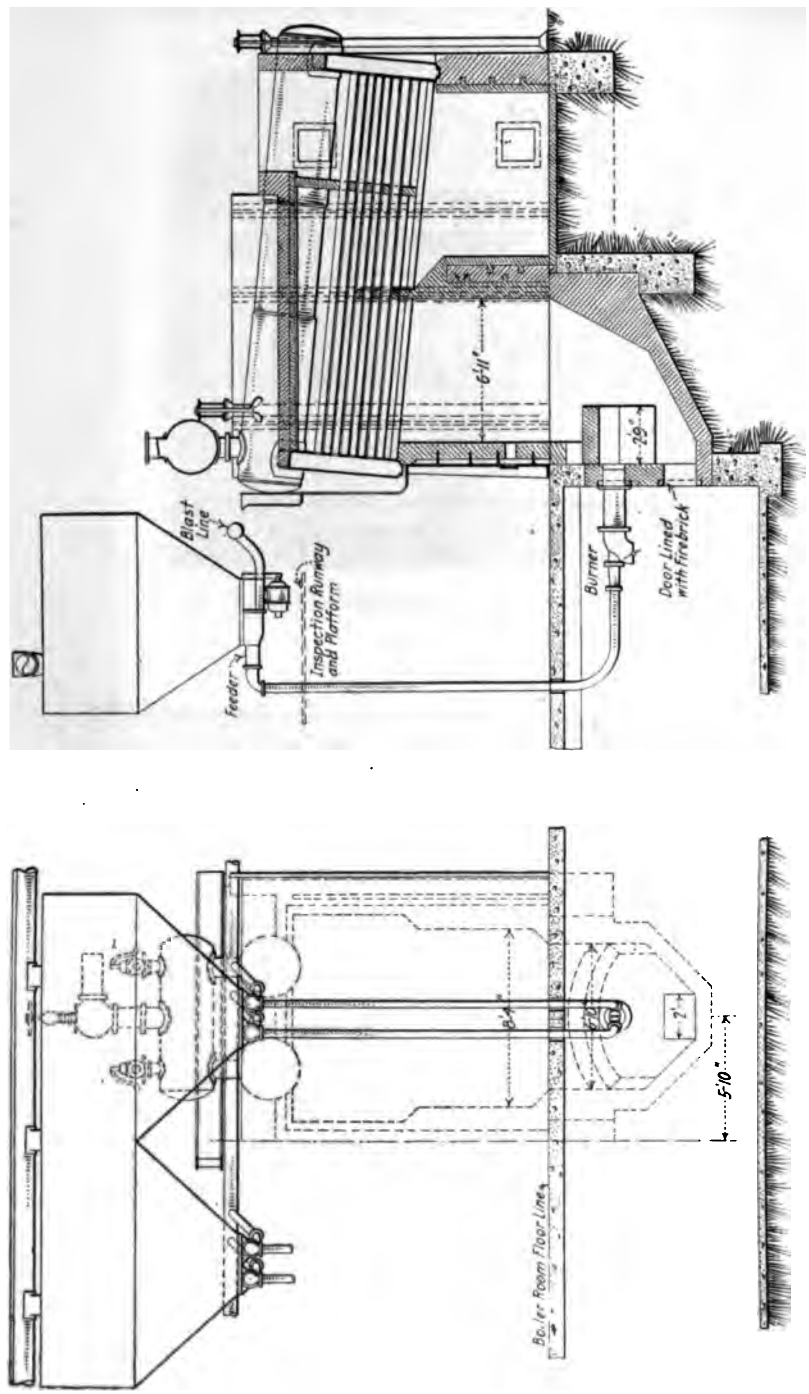


FIG. 21 APPLICATION OF PULVERIZED-FUEL STORAGE, FEEDING, AND BURNING EQUIPMENT TO 250-H.P. O'BRIEN TYPE OF STATIONARY BOILER FOR THE MISSOURI, KANSAS & TEXAS RAILWAY

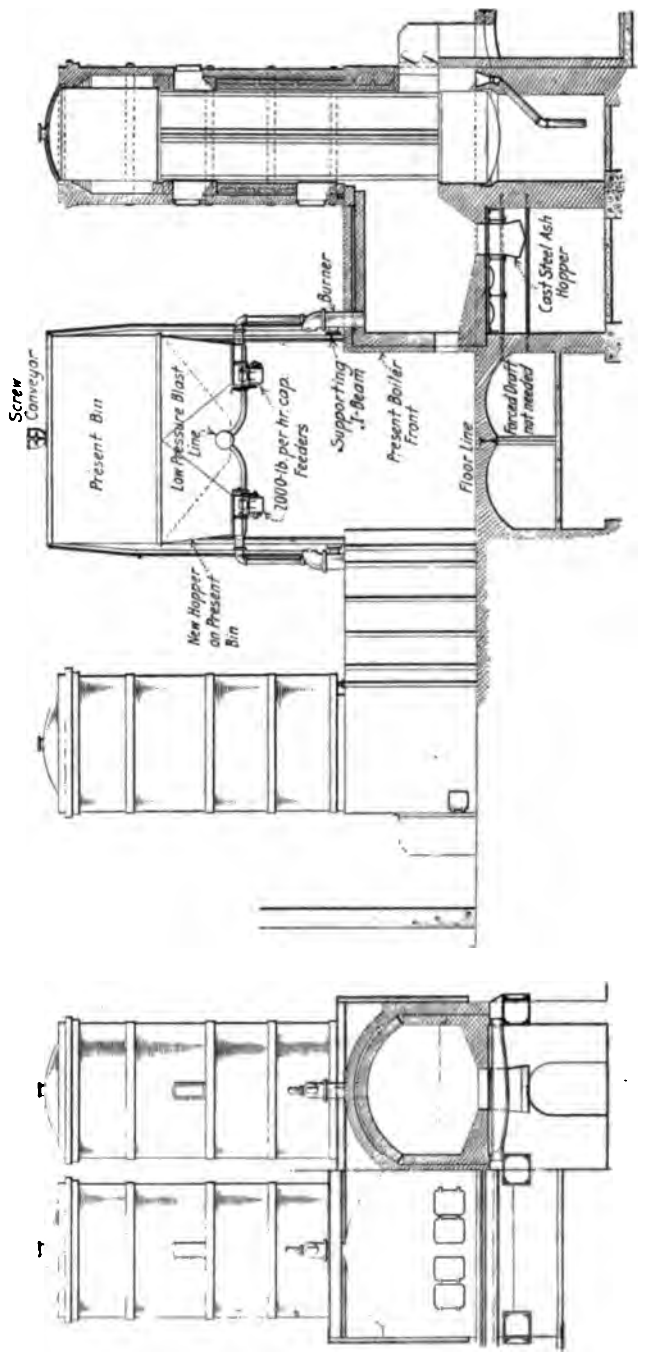


FIG. 22 APPLICATION OF PULVERIZED-FUEL STORAGE, FEEDING, AND BURNING EQUIPMENT TO 305-H.P. NOMINAL RATING WICKES TYPE OF VERTICAL BOILER

DISCUSSION

Angus Sinclair. It seems to me that the progress made lately with the combustion of pulverized fuel has been a greater progress than has been made in railways for many years, or in any other mechanical department.

There is a great tendency among railway men, and always has been, to leave things as they are. After they had got the ordinary

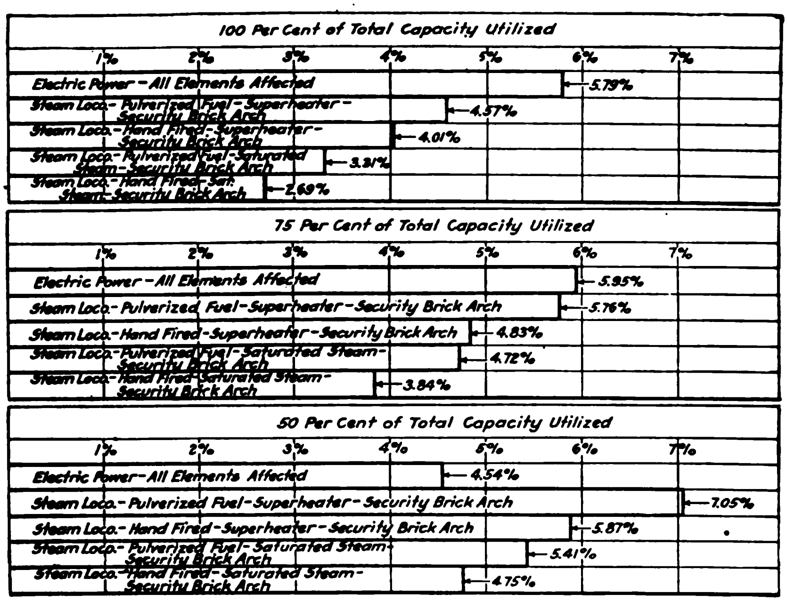


Fig. 23 Comparison of Thermal Efficiency of Electric and Steam Motive Power, Showing Percentage of Power Delivered at the Rail to 100 Per Cent B.t.u. in the Coal and Relative Value of Pulverized Fuel for Firing Steam Locomotives

steam locomotive working successfully, everybody was satisfied to let it go at that, and hooted at any one who proposed to make any improvements.

Nearly all the movement towards improved traction has come from outsiders, and I think that it is not very creditable to railway men that this has been the case, for they have had a very much better opportunity of knowing what was needed.

That has come about in quite a number of other lines than in the combustion of fuel, but I think the waste of fuel that has gone on in this country, ever since mining began, I might say, ought to be an object lesson to make people attempt a greater utility than what has been carried out until now. I think the using of pulverized fuel is the most promising improvement that has been made since steam engineering came into practice, and I have no doubt but what it is going to make wonderful changes in the mechanical operation of power.

W. L. ROBINSON.[1] The constantly increasing demand for slack grades of coal due to the extended application of mechanical stokers to power plants and locomotives, has brought about a condition where it is difficult to obtain sufficient of such fuel at anything except exorbitant prices, and the use of pulverized fuel would therefore appear to be particularly attractive.

From a study made on a road consuming about 6,000,000 tons of coal annually, the figures given in Par. 5*a* for fuel required when locomotives are standing would appear to be approximately correct.

On one of the larger railroads, the average mechanical delay at the heaviest maintaining station runs between 10 and 11 hours, and the lowest delay at dispatching stations is usually from 3½ to 5 hours, or an average of something over six hours for all stations. On account of the labor shortage, serious difficulty has been experienced at many points in retaining competent ashpit labor, and in many instances delays have been excessive. With the elimination of the ashpit delays through the use of pulverized fuel, the delays to locomotives at terminals could undoubtedly be reduced to half what they are at the present time.

In Par. 5*h*, Mr. Muhlfeld has brought out a feature that appealed to me particularly while riding on the Chicago and North-Western pulverized-fuel-burning locomotive, hauling a fast passenger train between Chicago and Milwaukee. The fireman and engineer were operating the locomotive for the first time. For the entire trip the fireman was not off his seat and the steam pressure did not vary more than three pounds. Upon arriving at Milwaukee, the fuel feed was cut off. The engine remained at Milwaukee 1 hour and 45 minutes, and a few minutes before the train was to leave, the fuel-feeding apparatus was started, the fuel igniting from the hot brickwork. On the return trip the engine was handled in the same manner as on the outbound trip. Before leaving the engine at Chicago, I asked the engineer what he thought of the engine and he replied,

[1] Supervisor Fuel Consumption, Baltimore & Ohio R.R., Baltimore, Md.

"No one has to do any work on the engine but me — I do not see where I have been relieved of anything." I called his attention to the fact that he had been in a measure relieved in observing the track and signals — that which is supposed to be responsible for the greatest strain upon an engineer — and that there were four eyes watching ahead instead of two.

E. B. Katte. While this is not a discussion of the relative merits of Steam versus Electric Operation, nevertheless, I do not feel that I can let pass that paragraph of Mr. Muhlfeld's paper pertaining to electric operation without at least expressing a difference of opinion. Referring to the combined cost of maintenance and operation, I do not think it will be difficult at the proper time to show that the electric locomotive with the complete electrical installation behind it can be operated and maintained cheaper per train-mile than can the equivalent steam-locomotive service. The complete cost, including fixed charges, can in some cases be shown to be in favor of electricity, while in others the special benefits to be derived from electric operation must be taken to justify the increased fixed charges.

As to reliability, there are typical electrical installations where the service has been shown to be more reliable under all conditions of traffic than the superseded steam service. The greater flexibility of the electric unit is demonstrated by the fact that in some cases one electric locomotive has replaced two steam locomotives, and in others the proportion is two electric locomotives to three steam locomotives. This is so for the reason that the electric locomotive does not have to be sent to the ashpit, turntable, or to be coaled and watered.

I believe that both from an engineering and operating standpoint it can be shown that the electric locomotive is entirely reliable and an economic unit. The one drawback to the more general use of electric traction on steam railroads is the first cost of the installation, and this at the present time prohibits the use of electricity except under special conditions.

George L. Fowler. There is one point that has not been touched upon, and that is the effect of the use of pulverized fuel on firebox stresses. During the past two years it has been my privilege to have made some investigations as to the action of the sheets of locomotive fireboxes in service for the purpose of determining the relative movement of the inner and outer sheets. And when I take into consideration the great sensitiveness of these sheets to variations in temperature, the use of pulverized fuel makes a strong appeal to my sense of what is good and proper.

I found that apparently, if a uniform temperature could be maintained, there would be no stresses set up beyond those of construction. For example, in making my determinations I used a beam of light reflected from a stationary and a rotating mirror, and at the start the two reflected beams struck the screen in line with each other. When set with a cold and empty firebox, these beams of light remained in line indefinitely. Nor did filling the boiler with cold, lukewarm, or hot water change their relative positions. But let the fireman open the blower and throw a piece of lighted waste into the firebox to kindle his fire, and it was impossible to get a reading that did not show a movement of the sheets, even though it were done within 10 seconds of the time of kindling the fire.

When steam is being raised there is a steady backward movement of the tubes; and then, when the throttle is opened, they give a jump backward, increasing the backward motion by from 30 to 50 per cent.

The opening of the firedoor will drop the temperature of the fire side of the sheets next it by from 200 to 250 deg. fahr.; and the closing of the door reverses the conditions.

Tubes expanded through the sides of the firebox for the admission of steam jets for smoke prevention, when not in use will admit a stream of cold air that will keep the fire side of the sheet colder than the water side.

All of these things indicate to my mind the desirability of keeping the firedoor closed and maintaining a uniform temperature within the firebox itself. And anything that will do this, as pulverized fuel will, will be of tremendous advantage in reducing the stresses to which the sheets would otherwise be subjected.

C. D. Young gave his experiences extending over four years in attempting to burn powdered fuel in a stationary locomotive firebox. In his own words:

"The result of our work up to this time indicates very definite limitations in the use of pulverized fuel. The first limitation we have found is that, when we attempted to burn the quantity of fuel per hour that would justify the expense of putting the device on the locomotive, we melted up most everything that we had in the firebox. Our combustion chamber gave us trouble and annoyance that would make it impracticable for us to apply the device to road operation.

This can be controlled, at least we have controlled it, but as soon as we attempt to control it we offset the efficiency of combustion; in other words, we reduce the temperature of the firebox by an excess of oxygen so as to protect the firebrick in the box, and then have an excessive amount of fuel for that burning rate. However, those who are working on the problem should continue their efforts toward a solution, as the use of powdered fuel is a logical step in the present evolution of steam power."

Mr. Young then took up various points in the paper and discussed them in the light of his own experience, saying in part:

"I would like to point out that the Atlantic-type locomotive referred to by Mr. Muhlfeld was a small one and the test results indicate that it was run under a very light load; and we have found that powdered fuel is quite successful on light loading. There were only six cars in the train, and under such conditions, by proper supervision, I believe you can produce practically smokeless combustion without resorting to the use of powdered fuel.

"Where the train consists of twelve or thirteen steel cars the smoke problem becomes a serious one in the restricted territory in large cities such as Chicago and Pittsburgh, and if you are making the experiment for the purpose of eliminating smoke it would seem more logical to make it on a locomotive with which you had great difficulty in producing smokeless combustion with hand firing."

Mr. Young summarized his conclusions with the statement that if powdered-fuel stoking could be developed so that it would successfully fire locomotives of the larger sizes and increase their power, the elimination of stand-by losses would give a margin of economy to offset the cost of maintaining the powdered-fuel apparatus and the increased cost of preparing the fuel.

L. S. Randolph. The paper compares the electric locomotive with the steam locomotive, but it fails to mention what is, in my estimation, the most important point, viz., the limitation of horsepower in the steam locomotive. I am informed that it is possible to get as much as 9000 or 10,000 h.p. from a single trolley wire of 1000 volts, whereas the steam locomotive is limited at present to 3000 to 4000 h.p. I am informed that the Norfolk and Western is hauling freight at fourteen and twenty-eight miles an hour, whereas a 2–8–8–2 or Mallet compound will drop down to two or three or four miles an hour with the same load. Five electric locomotives have, simply on account of their speed, due to increased horsepower, been enabled to displace seventeen Mallets. The author's paper gives the first indication that this terrific handicap of the steam machine may be overcome.

Another point which the author mentions is the dust clouds. I rather regret that he did not enlarge on that, because one of the difficulties I am sure we are going to have is the danger of dust explosions; and from the information I can obtain it is very surprising what a large explosion a small dust cloud will make.

George M. Basford. I believe that improvements in combustion will be the foundation of the most important future development of the steam locomotive. It has been stated that pulverized fuel held its greatest promise for the future because it presented the possibility of increasing the intensity of combustion per unit of firebox volume, that being the starting point for the greatest engineering development of the steam locomotive in the future.

It is extremely easy in locomotive design to run to weight and to more complicated wheel arrangements. Long before the Atlantic type of locomotive had been brought by the Pennsylvania Railroad to its present high degree of development, ten-wheel engines were very common. I don't think anybody would say today that the ten-wheeler has ever reached its conclusion, or been developed to the point of showing its limitations. The same may be said of the Prairie type, the Pacific type, and also of the more recent Mountain type.

In the matter of freight locomotives, I don't believe anybody can say today that the Consolidation type has ever reached the point of development of which it is capable. Pulverized fuel, however, has given this type a new lease of life.

To me, two statements stand out above all the other good ones in this paper. One is the first sentence in Par. 5 *i*, wherein the author says: "The future steam locomotive, on account of its track and bridge weight and tunnel and overhead clearance limitations, will be required to produce the maximum hauling capacity per unit of total weight."

We have not yet gotten fairly started on that principle, and to have that question raised in connection with the subject of improved combustion or pulverized fuel is a very important thing to come before the engineers and the railroad people at this time.

The other statement of equal importance is Par. 70, wherein the work of an Atlantic-type locomotive burning pulverized fuel is compared with the work of a Pacific-type locomotive. Many Atlantic-type locomotives are in use in this country today, and the Atlantic type of locomotive really represents, in this case, a large number of small engines in service in every part of the country. This statement becomes so important because the Atlantic type with improved combustion has been made to do in one case the work of a Pacific-type engine. The next thought naturally is, How many Atlantic types may be made to do the work of Pacific types today? Application of this pulverized-fuel principle to new designs will have the effect of prolonging the life and putting back into service a good many outclassed engines on the one hand, and we have for the future the possibility of using it from the ground up in new designs of great power when the designer starts with a clean sheet of paper. That has never yet been done with the exception of the consolidation locomotive on the Delaware and Hudson. I mean that pulverized fuel will enable the designer to produce greater power per unit of weight than would otherwise be possible.

It seems to me fair to state, since the subject of electric locomotives has been mentioned, that the engineering development — the real engineering development — of the steam locomotive is only just beginning. The progress of the past ten years has been wonderful, as revealed by figures given in papers which have been presented before the Society. In pulverized fuel the designer of locomotives has a factor that he never had before. It will enable him to secure sufficient steam-making capacity. The boiler has always been the limiting feature. Pulverized fuel opens the way for greater improvement made possible by improvements which enable the firebox to make more heat per pound of weight of the locomotive.

Charles Whiting Baker was impressed with that part of the paper wherein it was stated that the pulverized-coal equipment would permit the use of low-grade fuels. In some of the special locations where such fuels are obtainable at low cost, a large reduction in operating expenses may ultimately be brought about, even when the cost of powdering the fuel is taken into account.

He thought that the possibility of using this development for switching locomotives in yard service was of immediate interest. If with powdered coal a locomotive could be operated without smoke, a big step had been taken toward putting an end to the agitation against the switching locomotive and the public demand for electrification of railway terminals.

It would be very desirable if the author, in his closure, would state if it were possible to regulate the burners so that the amount of fuel burned could be made to correspond with the demand for steam — whether it were possible, for example, to operate a switching locomotive and not have it blow off and waste heat while standing.

He also raised the question whether the amount of heat stored in the brickwork would not cause a switching locomotive to waste steam from the pop valves in the frequent intervals when it is standing idle.

C. W. CORNING[1] (written). I am familiar with many of the results from the use of pulverized fuel as stated in Mr. Muhlfeld's paper, having for several months had charge of an Atlantic-type locomotive in first-class passenger service burning this form of fuel.

Of the many things which contribute toward the lightening of the enginemen's cares in the discharge of their duties, probably the two most essential are the proper working of injectors and the free steaming of the engine.

In all of the runs made by the engine mentioned, it never failed to deliver all the steam pressure required (in the language of the fireman, it is "two o'clock" all the time by the steam gage). In the event of the failure of the injectors it is a simple matter to shut off the supply of fuel until such time as the matter can be remedied and the fire relighted.

A very prominent feature of the pulverized-fuel engine is the fact that the draft appliances need not be changed for different grades of fuel or climatic conditions of the various seasons of the year. The locomotive has been operated in all kinds of weather, in very heavy rain storms, snow storms, extremely hot and dry weather, and when the temperature was several degrees below zero, and there never was any noticeable change in the steaming qualities of the engine.

Last, but not least, of the many good qualities stated in Mr. Muhlfeld's paper is that of the possibility of enlarging exhaust-nozzle openings. The area of the exhaust-nozzle opening on the C. & N. W. engine has been increased about 40 per cent. In summing up, what is nearest the heart of an engineman is a free-working engine, and this is obtained by burning pulverized coal.

W. A. EVANS (written). Only two of those discussing the proposition suggested what has always been the one difficulty in obtaining

[1] Chief Smoke Inspector, C. & N. W. R. R., Chicago, Ill.

powdered-fuel combustion under boilers, viz., the difficulty of maintaining the brickwork with its necessity of large combustion space for this form of burning. One of the men stated frankly that his company had experimented with powdered coal, and found when forcing the boiler, as is necessary in locomotive practice, that the brickwork was difficult to maintain. It is hoped that Mr. Muhlfeld will give a full account of brickwork difficulties and means by which he would overcome them.

To those who have watched powdered-coal development there is much satisfaction in the apparent success of burning it under a Franklin boiler at the American Locomotive Works in Schenectady, N. Y. That installation seems to indicate the necessity of large combustion space and slow velocity of fuel and air entering the furnace. These demands are quite contrary to the limited possibilities in the illustrations shown in Mr. Muhlfeld's paper on the application to locomotive boilers. After operating for over two years, to the writer's knowledge there is no evidence that results have been so desirable as to have the other boilers in the same plant equipped with the same fuel apparatus, and two years of experience ought to attract more attention than is evidenced by the lack of further development in the stationary line in this time.

There is further cause for caution on the part of those considering the use of powdered coal in the fact that of the many concerns claiming to produce powdered-fuel equipment and soliciting opportunities to undertake the entire installation, none, to the writer's knowledge, will give a definite guarantee of results.

Lawford H. Fry (written). Mr. Muhlfeld has shown the economy and advantage of applying pulverized fuel to locomotives designed with a grate for burning lump fuel. It would be interesting if he could tell us what further advantages could be obtained if the locomotive were to be designed from the outset without the restrictions imposed by the necessity of a grate. To see what changes would be involved, consider the difference in the two processes of combustion. In firing lump coal, the fuel is introduced intermittently, being thrown on to a bed of incandescent fuel. The volatile matter is distilled off and burnt in the volume of the firebox. The fixed carbon, or at least the greater part of it, remains on the grate. In order to burn this, a strong current of air through the grate is necessary, since, owing to the comparatively small surface of the lumps, it is necessary that they be scrubbed vigorously by the air, using an amount in excess of that which combines with the carbon during combustion. This process of combustion gives two requirements in locomotive design: sufficient volume for the combustion of the volatile matter, and sufficient grate area for the combustion of the fixed carbon. It is well known that the higher the percentage of fixed carbon, the larger must be the grate, hence the large grate areas necessary for anthracite-burning locomotives.

With pulverized fuel the conditions are, as Mr. Muhlfeld has pointed out, entirely different. The fuel is introduced continuously as it is burnt, not intermittently. Owing to the very large surface which each particle of fuel has in the pulverized form, the scrubbing action of the air is not required, and complete combustion can be obtained with the theoretically necessary amount of air, provided the fuel floats with the air for a sufficient time at the temperature necessary for combustion. It is obvious that with the pulverized fuel no grate area is necessary, but there must be sufficient firebox volume and a sufficiently long travel for the flame to allow complete combustion to take place before the gases enter the flues and are cooled below the combustion temperature. It would be very interesting if Mr. Muhlfeld could give us some information as to the volume and length of firebox required for a given rate of combustion.

Now coming to the effect which this novel process of combustion may have on locomotive design: So far as the lump-fuel locomotive is concerned, C. D. Young in discussing locomotive proportions before the Master Mechanics' Association last June, pointed out that the basic steps in designing a locomotive are, first to determine the grate area necessary to give power, and then to settle on the depth of throat sheet which will give sufficient firebox volume for efficient combustion. The depth of throat sheet in conjunction with the diameter of the drivers fixes the diameter of the boiler which can be used with a given loading gage. This all means that the design for the whole lump-fuel locomotive is built up around and is controlled by the grate area and the depth of firebox.

Now with pulverized fuel it would seem to be possible to eliminate both grate and throat sheet, and to use a boiler of larger diameter and special design which would give even better results than can be obtained by adapting an existing lump-fuel boiler. I have not attempted to work this thought out in its practical details, but it would be interesting if Mr. Muhlfeld could tell us whether he has considered this phase of the question of pulverized fuel. Of course, a special design of boiler along the lines indicated presupposes complete conversion of the road using it to pulverized fuel, but no doubt Mr. Muhlfeld has sufficient confidence to look thus far into the future.

E. H. Stroud[1] (written). The results reported by Mr. Muhlfeld show an important advance upon the practice of burning lump coal upon locomotive grates, hand-fired, and a worth-while saving to the railroads in many ways. Still better results can be had, however, and a greater saving of coal and money be made by abandoning the use of stack draft and making all the air necessary for the most complete combustion possible carry into the firebox all the coal that has to be burned.

The statement has been made that it would be impossible to find space enough that could be spared upon the front end of a tender for such apparatus as would be required to do this. Such apparatus, however, is available from my firm, in size to fit that space, and it can be built to operate, at variable speed, by either steam or electricity. It is the result of 16 years of effort, and I know of nothing but our method of operation which has been really successful under boilers of any kind. We tried the stack-draft method and abandoned it years ago for the very faults shown by Mr. Muhlfeld's locomotives. They save 15 per cent of coal. Our plan saves nearly 50 per cent in stationary boiler plants, where the usual losses are not so great as in locomotives. By our method we are able also to give furnace temperatures from 1800 deg. fahr. to 3000 deg. fahr. and over, and such a low temperature has not been achieved before with powdered coal, thus showing perfection of control.

It is evident from Mr. Muhlfeld's showing that the use of stack draft renders necessary for the control of the fire of a locomotive a much more complicated mechanism than is needed for firing by the method and device I have mentioned, which latter performs for powdered coal the same service as the device called a carburetor performs for gasoline: namely, it receives the entire quantity of the two fuel elements, coal and air, and mixes them thoroughly together in exactly the right proportions before they enter the combustion chamber.

It makes no material difference, therefore, whether the locomotive is standing or running, or what its speed may be, or whether there be wind or no wind, provided it be fired by such a stoker; whereas, when depending upon stack draft the circumstances mentioned must

[1] E. H. Stroud & Co., 928–934 Fullerton Ave., Chicago, Ill.

necessarily exert a considerable influence, and require the use of a constantly varying quantity of exhaust steam at all times to counteract those influences, thus putting a back pressure upon the cylinders and reducing the efficiency of the locomotive proportionately.

I think it is admitted by most railroad men that the use generally made of the exhaust steam to create stack draft reduces the total locomotive efficiency 25 per cent. Such being the case, the use of the apparatus referred to would give the locomotive one-third more power to use, besides effecting a greater saving of coal and water and the simplifying of the control of the firebox results, because the exhaust steam will not be used to create draft.

J. H. MANNING[1] (written). Mr. Muhlfeld's paper accords with our practical experience with engine 1200, having a boiler the principal dimensions of which are as follows:

Diameter, first course............................86 in.
Firebox, size....................114 in. wide, 126 in. long
Equivalent heating surface.....................5004 sq. ft.
326 2-in. tubes
46 superheater units.

This boiler supplied steam to 27 x 32-in. cylinders, and developed through a medium of 63-in. wheels 60,000 lb. tractive power at the drawbar, carrying 205 lb. of steam.

This company, the Delaware and Hudson, is closely connected with a territory that produces about 80,000,000 tons of anthracite per year. It is not hard to understand that a great deal of extremely fine coal and dust accumulates in the process of marketing. This cannot be burnt on the grates, but, if at all, in suspension in a refractory furnace. For this latter purpose we have available in our neighborhood 550,000 tons per month. This latter and the fact that there were located around us a number of industrial plants successfully burning bituminous coal in pulverized form, encouraged us to build an experimental locomotive of the dimensions stated above, producing approximately 2700 cylinder horsepower. To guard against the possibility of failure, the entire firebox, boiler and locomotive was so constructed that the application of the powdered-fuel mechanism could be readily removed and the firebox, etc., arranged for burning fuel on grates.

We soon found out it would be impossible to burn clear anthracite coal in pulverized form. Due to the low volatile, it would promptly

[1] Supt. M. P., Delaware and Hudson Co., Watervliet, N. Y.

snuff out if the engine should happen to slip or worked extremely hard, and the firebox temperature would not permit it to again flash. We therefore arranged a program and determined to start with 75 per cent bituminous and decrease until it was found that this objectionable feature was removed. This was continued until a mixture of 60 per cent anthracite and 40 per cent bituminous was obtained. We find this gives splendid results; the engine steams freely with very little smoke and is very nicely controlled by the fireman to the extent of keeping the engine within three pounds of the maximum pressure continuously without popping under the different operations necessarily obtaining in a day's work with an engine of this character, experiencing no firebox trouble whatever.

Such difficulty as we have had with the pulverized-fuel mechanism for the introduction of the fuel into the firebox has been satisfactorily eliminated, and the successful burning of pulverized fuel in suspension in a locomotive firebox, to my mind, has passed beyond the experimental stage. It is now a question of economy only, and this depends upon the source of supply in a great measure.

S. S. Riegel[1] (written). In the use of an induced draft for the air of secondary combustion lies possibly the greatest assurance of success, as this overcomes the destructive heat action of the fuel jets against the brickwork of the combustion chamber and furnace linings, and at the same time furnishes a convenient way to secure the necessary air for secondary combustion.

It is particularly interesting to find that pulverized fuel so easily adapts itself to the greatly varying locomotive service conditions. From the viewpoint of overcoming stand-by and firing-up losses of the locomotives, powdered coal is given an opportunity which is not possible in stationary plants. The effects of radiation of heat from the fuel must not be ignored, as its influence in steam formation is exceedingly valuable.

In powdered-coal furnaces the ash and ashy residue are carried along and removed with the gases, otherwise there is liability of clogging the heating surfaces and boiler passages, with a falling-off of performance. If perfect combustion can overcome most of this difficulty, attention should be directed to secure it, and as this ashy residue always carries away sensible heat, some heat losses must be expected from this source.

As it is necessary to separate the fuel particles and surround

[1] M. E., D. L. & W. R. R., Scranton, Pa.

them with sufficient air for perfect combustion, it would seem equally desirable to separate the particles of the crushed materials in the drying process, and the most effective dryer would be the one which best effected this separation. The common dryer does not seem so well adapted for the purpose as the dryers in use in sugar factories for pulp drying; such a dryer is the Büttner, for inorganic materials, which is very efficient and accomplishes the drying at the least cost.

In America, powdered coal as fuel has been known for many years, especially in the cement industry; its first successful use commercially being for metallurgical work at the American Iron and Steel Manufacturing Company, at Lebanon and Reading, Pa. The Midvale Steel Company, at Philadelphia, Pa., equipped its continuous heating furnace for pressed-steel-wheel and other departments. The Burden Iron Works, at Troy, N. Y., has a large powdered-coal installation. This fuel is being applied to over thirty double puddling furnaces, as well as to a number of two- and three-door reheating furnaces; a saving of from 30 to 33 per cent in fuel is not in any way exceptional. The Milton Manufacturing Company, of Milton, Pa., has a similar installation for busheling and heating furnaces, giving similar results. The Fort Wayne Rolling Mills Company, of Fort Wayne, Ind., installed a powdered-coal plant which furnishes fuel for puddling and busheling furnaces, as well as for reheating furnaces for the rolling mills. Here, as an example, the fuel consumption of one of the heating furnaces using producer gas on certain classes of work averaged 560 lb. of coal per ton. After changing the furnace and applying powdered coal to it, the fuel consumption was reduced to 350 lb. per ton on similar materials. The Scranton Bolt and Nut Company, Scranton, Pa., has installed this fuel system and their plant has been in operation for several years. Powdered coal has been applied in continuous billet-heating furnaces as well as for open-hearth work. The Pennsylvania Steel Company, Steelton, Pa., has applied it to continuous heating furnaces, with considerable economies. The Standard Steel Company, Berlin, Pa., is applying powdered coal for work in similar heating furnaces. On open-hearth work the National Malleable Castings Company, Sharon, Pa., has had a very successful installation in operation for several years. The fuel consumption for a ton of steel is less than 500 lb., and the company has extended its use to its two plants in the Chicago district. The Carnegie Steel Company, at its Farrel plant, has been trying it in competition with other fuels, such as producer gas, fuel oil and tar, and the fuel consumption by the powdered-coal furnace is under 600 lb. per ton of steel produced. Powdered coal is being introduced on a very large scale at its Homestead and Clairtown works. In puddling furnaces the fuel consumption runs from 1200 to 1700 lb. per gross ton of muckbar produced, whereas in hand-fired puddling furnaces it runs anywhere from 1900 to 3000 lb., the average being 2500 lb. In hand-fired heating furnaces the fuel consumption runs from 700 to 1800 lb. per ton of finished iron; with powdered coal, from 450 to 650 lb., according to the size of the furnace, character of materials, etc.

Powdered coal is not adaptable to every use, but where it could be successfully applied it has proved to be a valuable fuel, in that the supply is concentrated at one point and distributed to the various furnaces. Its control, as far as temperatures and character of flame are concerned, is as good as that with oil or gas. Its adaptation to locomotives is particularly useful and interesting, and it is hoped that it will meet with unqualified success.

The Author. I am somewhat surprised and somewhat disappointed that Dr. Sinclair should say that all the steam-locomotive development, or a large part of it, has been made by people outside of the railway field. Personally, I have spent about twenty years in active railroad work, and during that time I was instrumental in developing the Mallet locomotive and the Walschaerts-motion gear for American practice. Evidently Dr. Sinclair does not consider that this has amounted to much, although he was one of its strongest advocates out of a very few who favored it when it first came out. But since that time the Mallet has come into its own, particularly on some of the heavy-freight-carrying railroads, such as that which Mr. Katte represents, the Norfolk and Western, Chesapeake and Ohio, Carolina, Clinchfield and Ohio, Western Maryland, Virginian, Baltimore and Ohio, Delaware and Hudson, Santa Fe, Great Northern, Northern Pacific, Southern Pacific, Milwaukee, Kansas City Southern, and many others. In fact, practically all of the relatively low-freight-rate steam roads that have made great progress in increasing their average revenue train load and in reducing their operating costs are now using Mallet locomotives, which indicates that they have done a great deal towards improving steam-railway operation.

With respect to the points that Mr. Robinson brought out, I would say that he has looked at this matter from a strictly practical standpoint, and, in my opinion, the advantages that he names, through ability for the railway to pool the various grades and qualities of coal that they secure from the different mine operations along their line, reduction in fire building, ashpit and other terminal delays, and the elimination of arduous labor on the part of the fireman, are among the most important items with which the railways are contending today. The advantages of pulverized fuel with regard to all of these have already been demonstrated in road and terminal operation.

Mr. Katte brought up the matter of electrification. I was general superintendent of Motive Power of the Baltimore & Ohio R.R. for five years and during that time had supervising control over a short line of electrically operated trackage through the city of Baltimore, which was always one of the troubles of the Mechanical Department. Of course, the development since that time has been very great. That installation was commenced in 1895, and the time I refer to was between 1903 and 1908, when I had the opportunity to follow it closely. Comparing the entire working expense of locomotives of similar tractive power operated in like service, the cost per 100 miles actually run was about double for the electric motive power, exclusive of interest, depreciation, taxes, and insurance. Furthermore, the first cost for the locomotives alone was, on a per pound basis, about twice as much for the electric as for the steam.

Mr. Davis, Electrical Engineer of the Baltimore and Ohio, may have some later data on what has been accomplished in that direction.

Now, in regard to the present-day cost for electric installation and operation, I saw the other day where the Milwaukee Road bought twenty-three electric locomotives at a cost of about $114,000 each. I think that is an unusual price, although generally it was about double the cost of a steam locomotive. With respect to the cost of maintenance of steam and electric locomotives, it must be remembered that the steam locomotive is a self-contained unit and carries its own power plant, while the electric locomotive is not, and due allowance must be made for that fact.

If, as Mr. Katte states, the electric locomotive and the complete electric installation behind it for the movement of heavy traffic over long distances can be maintained and operated cheaper than the equivalent steam unit, then we would like to have some figures to show it. I would assume, taking the New York Central, or the New York, New Haven and Hartford, that the load factor would run 30 to 35 per cent, and that on the Milwaukee it will probably run from 5 to 10 per cent. That being the case, a lot of power is being developed that cannot be used and which must be accounted for in some manner or other. A fair comparison must cover the combined fixed charge, maintenance and operating expense involved per drawbar horsepower per hour.

Mr. Katte felt that the electric installations so far made have demonstrated greater reliability under all conditions than steam locomotives. In reply to this it might be interesting to make a comparison of steam-railway steam-locomotive operation for from 25- to 30-mile runs in the Chicago district with steam-railway electric-locomotive operation for similar distances in the New York district as regards schedule time and regularity and continuity of service, summer and winter. Then again, there are emergency conditions where all electric motive power is tied up at one time. I have had a number of experiences where something has occurred, usually due to an act of Providence, or to a mechanical or electrical failure, where the whole system was tied up, and then the only relief that was left was to get steam locomotives into the electrical zone as soon as possible to clear it up. We had several instances of that kind around the New York terminals last winter.

With respect to flexibility of service, a steam locomotive can be operated wherever the gage and strength of the track admit it, whereas an electric locomotive is confined to the electrified section that fulfills its electric-current characteristics and contact-line requirements. In the January 8, 1916, issue of the *Railway Review* I covered this phase of the subject in considerable detail in a paper entitled The Future of the Steam Locomotive, the majority of the data presented having been obtained from about five years of actual experience with steam- and electric-locomotive operation on the Baltimore and Ohio Railroad.

I still feel quite assured that the self-contained motive-power unit that is independent of any other adjunct in connection with its work except the tracks, is going to be the one that we will have to depend upon for handling heavy freight trains over long distances. Of course, in a terminal proposition we have other conditions, and electrification is bound to be installed.

The point that Mr. Fowler makes about more constant firebox temperature with pulverized fuel is certainly correct, as the firedoor is never opened during the time that combustion takes place, and the liability of a cold shaft of air passing through the firebox, as where coal is burned on grates, is entirely eliminated.

Mr. Young refers to work that the Pennsylvania has done and which, from results obtained with a locomotive in a stationary condition, must have been with entirely different means, methods and processes from what we have developed and make use of in both locomotive- and stationary-boiler practice. Their arrangement evidently concentrates or pockets the heat in connection with the refractory material, and too great a velocity pressure of the products of combustion must obtain in the firebox.

Mr. Young also brought out that the Chicago and North-Western locomotive water-rate performance was exceptionally high. He evidently considered this from a cylinder-horsepower, rather than from a drawbar-horsepower-per-hour standpoint.

With respect to his statement that from 40 to 60 per cent of the fuel consumption of steam locomotives, when coal is being burned on grates, takes place when the engine is not working steam: Our data in this regard, obtained from actual road results, are in accord with his figures. Of course, the more congested the operation on a right of way, as, for example, during the past few months on various steam roads in the Eastern district in the United States, the greater this percentage becomes. Mr. Robinson also brought out this matter of terminal delay, and at this time, when the railroads are saturated with business, it seems to me that the matter of terminal handling of locomotives is one that requires serious consideration. As Mr. Young states, the percentage of stand-by loss in fuel is largely due to terminal conditions. I have many times had the personal experience of going to a large congested terminal and finding from fifteen to twenty locomotives waiting to get on the ashpits, five or six on the ashpits, and each requiring from one to two hours to clean the fires. That is a sort of congestion due to a terminal condition that can now be overcome.

Mr. Randolph brought out the matter of liability of dust explosions. From the fact that about 8,000,000 tons of pulverized coal are now being burned in the United States per annum, it is thought that general practices with respect to the handling, drying, pulverizing, storing and disbursing of the same have been pretty well taken care of, and the general results with the various cement and industrial plants using this kind of fuel indicate this to be the case. Where we have had our designs of fuel-preparing, -handling and -burning equipment installed, up to the present time no trouble whatsoever has obtained.

Mr. Basford's idea of prolonging the life of existing locomotives by modernizing them through the application of pulverized fuel was well taken, and an enormous amount of work remains to be done along this line which will enable the reclaiming of motive power that in its present condition is ineffective and uneconomical.

Mr. Baker brought out the problem of smoke elimination, particularly in the larger city terminals where numerous switching and transfer locomotives are used.

The Delaware and Hudson locomotives operate regularly in practically a switching district. Sometimes they stand for half or three-quarters of an hour, and at other times they are switching back and forth continuously or making a pull of thirty-one miles over a heavy grade, and there has been no difficulty in operation.

One thing that has caused a great deal of trouble is to get an absolute control of the fuel feed. We have only worked this recently in combination with the B. F. Sturtevant Company, of Boston. A special fuel-feed governing control that gives any desired range and makes a very effective device has now been perfected.

The control of not only the smoke but also of sparks, cinders and popping off, as well as the reduction in the exhaust-nozzle noise, is entirely possible and practicable with the development that has already been demonstrated through the operation of the Chicago and North-Western pulverized-fuel-burning locomotive in the city of Chicago.

In regard to Mr. Baker's remark about the ash: We made several attempts to get some of this ash from the smokestack, and we find that it varies almost directly with the fineness of the fuel. Where the fuel is relatively coarse and has a good deal of moisture in it, it is quite perceptible. When the fuel is according to our specifications, which we state should be one per cent of moisture or less, and 85 per cent through a 200 mesh, you cannot find it. Of course, it is there, but only to a very slight extent. I heard of one case where a railroad official rode on top of the back end of a tender for about sixty or seventy-five miles after his cap flew off, and at the end of the run he didn't find a particle of grit in his hair (he still has some!). If there had been any ash coming out of the stack, he would certainly have gotten some of it in his eyes or hair.

The items that Mr. Corning brought up were those which appealed to the practical railway operating official as well as to the engineer and firemen in charge of the locomotive, and the benefits to be derived from flexibility in the operation of the equipment and the maintenance of economical working steam pressure at all times and under all conditions, and further the increased tractive power by the enlargement of the exhaust-nozzle area are most essential in that regard.

Mr. Evans brought up the difficulty in maintaining brickwork, with the necessity for large combustion space in the use of powdered fuel in boilers, and requested data on that subject. The answer is: Reduce the velocity pressure of the combustion gases to the minimum; eliminate restricted areas in the brickwork through which these gases must flow; and bring these gases into contact with heat-absorbing surfaces as quickly as possible after the combustion process has been completed. We have found that owing to the rapidity of oxidation large combustion space and brick area are not necessarily essential to effective results.

Probably the reason for Mr. Evans' not being able to secure a definite guarantee of results from the use of pulverized fuel from the concerns with whom he has taken the matter up, is the fact that until recently very little practical knowledge has been available on which to base such assurances, and that essential means, methods and processes were not really developed along practical lines until the application of pulverized fuel to the most unfavorable condition, *i.e.*, the steam-locomotive boiler, was undertaken.

Mr. Fry brought up the question of volume and length of locomotive firebox necessary for a given rate of combustion. All of our development work has been done with existing standard designs of locomotive fireboxes and boilers ranging from 48 in. wide by 90 in. long to 114 in. wide by 126 in. long, and in no instance has there been any difficulty experienced with burning the requisite amount of fuel to secure economical boiler horsepower under the most extreme working conditions. This applies to lignite as well as to bituminous coal and to a mixture of 60 per cent of anthracite slush and 40 per cent bituminous screenings.

The point that Mr. Fry brings up relative to being required, when coal is burned on grates, to build up a locomotive design around the grate-area and depth-of-firebox dimensions, is largely correct, and the burning of fuel in suspension will enable the use of special designs of locomotive boilers, for example, for longer flameway and return tubes, which will permit of utilizing a much greater percentage of the fuel value than will ever be possible by the burning of fuel on grates.

Mr. Stroud points out the feasibility of securing still better results than what have been obtained by abandoning the use of stack draft. The practical work that we have done along this line has, to the present date, not demonstrated this. While stack draft is not needed to secure combustion results, it is required to produce boiler and superheater capacity and effectiveness, which all-important factors Mr. Stroud has apparently overlooked.

Mr. Manning states that on his road a mixture of 60 per cent of anthracite and 40 per cent of bituminous is now giving splendid results in locomotive service. I desire to elaborate on this and state that the 60 per cent consists of anthracite slush, or heretofore waste by-product of mining, and that the 40 per cent consists of bituminous unwashed screenings, all of which is mixed and pulverized.

This mixture gives a fuel of about 15 per cent volatile as compared with the heretofore generally recommended practice of not less than 30 per cent volatile. Furthermore, this result has been acquired with the second type of furnace refractory arrangement tried out, and we feel that the next change in the refractory arrangement will result in the utilization of a mixture of at least 80 per cent of anthracite slush and 20 per cent bituminous screenings, for the reason that no difficulty whatever is now experienced in burning the straight anthracite slush in stationary boiler practice and obtaining the requisite boiler capacity and maximum efficiency.

Mr. Riegel drew notice to the fact of the paper not bringing out that we had gotten the final and very best results. We do not feel that we have. It requires a great deal of time in the development of any mechanical device to get to that stage. Our preliminary work was done in connection with exceedingly small locomotives, as we felt that there would not be much difficulty in firing a locomotive that had a firebox of say 60 or 70 sq. ft. of grate area with pulverized fuel, but what we wanted to work out first was to accomplish this in the smallest boiler, where the element of furnace capacity was limited. We knew that if we could make it a success there, that we could adapt it to all existing steam locomotives, and that there would be very little trouble in applying it to the modern type of locomotive, and that is the only reason why the equipment used for the preliminary work was confined to the smaller locomotives.

The use of pulverized fuel for steam locomotives is now an accomplished fact, and I will predict that in combination with higher boiler pressure and superheat temperature, refractory-material furnace, feedwater heating and cross-compounding it will put the steam locomotives of the future on an operating basis that will continue its use indefinitely.

No. 1567

AN ANALYSIS OF MARINE SAFETY VALVES, WITH SUGGESTIONS FOR REPAIRS AND IMPROVEMENTS

By E. F. Maas, Puget Sound, Wash.
Member of the Society

The purpose of this article is not to give a complete description of the design and working of all marine safety valves to be found on the market today, but to analyze some of the characteristics of the working parts of the more common types, to indicate their advantages and the difficulties experienced with them, and to point out helpful methods in repairing and improving old valves. The paper presents the results of observations made on valves under working conditions and during tests of repaired valves in the mechanical laboratory. In the case of one valve, not yet upon the market, the statements are based upon experience gained from work on other valves and on a somewhat incomplete report of laboratory tests of this valve. This test was not witnessed by the author.

2 Each of the seven figures gives, not a complete view of a valve, but a partial section only, this section showing such parts as come within the scope of this paper. It is assumed that the working of these safety valves is familiar to every reader, as well as the nomenclature of the valve parts. No general explanation of them, therefore, will be given.

3 Fig. 1 represents a valve of a somewhat obsolete design probably not made by any manufacturer today. This type is still to be found, nevertheless, in many of the older ships, and should be considered. Fig. 2 shows a later type of valve having an attachment for adjusting the blow-down, a feature indispensable in a modern marine safety valve. Fig. 3 is practically the same as Fig. 2, the main difference between the two being in the design of the blow-down adjustment. Fig. 4 represents one of the latest and most successful designs

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on the market at the present time. In Fig. 5 is shown a valve which has not yet fully emerged from the experimental stage, and to the knowledge of the author is not yet offered for sale as a marine safety valve. Figs. 6 and 7 show modifications made to valves of the type represented by Fig. 3, these modifications having been applied during the repair of these valves in order to fit them for further service, after their original working adjustment had been destroyed through long service or neglect in maintenance.

4 The particular features to be discussed in this article are:

a Valve seat, flat or beveled, and its tightness
b Pop of valve, amount of simmering before pop, and height of lift
c Closure of valve and chattering at closure
d Blow-down of valve and methods of adjustment
e Discharge capacity
f Overhaul and adjustment of old valves and improvements made at slight expense.

VALVE SEAT

5 A glance at the figures will disclose two valves with flat seats (Figs. 1 and 5), the other five having beveled seats. While this might indicate that the beveled seat is the more common, the relative advantages of each type are strongly supported by manufacturers of the two types. The experience of the author has been that a flat seat is just as easy to grind in and make tight as a beveled seat, and that the flat seat is more likely to stay tight. On account of the distortion of the valve seat which will result from a slightly uneven expansion or from spring of the material under high pressure, the tightness of a beveled seat is seriously affected, and the more so the greater the bevel, if by bevel we understand the angle between the valve seat and a plane perpendicular to the center line of the valve. On the other hand, a flat seat will remain steam-tight even after a considerable distortion has taken place, provided that the valve has been carefully ground to its seat in the first place. It should be borne in mind that the valve and seat are in contact only along a very narrow strip, in the case of either flat or beveled seats. For equal lifts of the valves a flat seat will give a greater discharge capacity under certain circumstances, as will be discussed later, a fact which will weigh in favor of the flat seat, for some designs, at least. It has been argued that a beveled seat, especially where the bevel is considerable, about

45 deg., will more easily rid itself of particles of scale or other foreign matter between the valve and its seat. In this connection attention should be called to the prevalent custom in marine practice of attempting to tighten the valve on its seat by turning it by means of the valve stem. Such practice usually has an effect directly opposite to that desired. A better way is to lift the valve from its seat by the easing gear and thus blow away the scale or other matter. A steeper bevel than 45 deg. is inadvisable for a steam safety valve, as the tendency for the valve to stick on its seat is too great and may produce disastrous results. It is significant that while only a few years ago

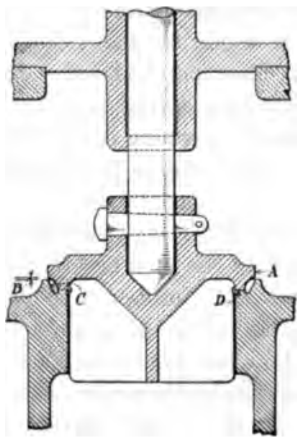

FIG. 1 OLD TYPE OF SAFETY VALVE

most safety-valve specifications called for seats beveled at an angle of 45 deg., specifications of today, notably those of the U. S. Navy Department and various state boiler rules, do not require it, indicating that a transition from 45 deg. to the flat seat seems to be under way. It is the author's opinion that such a change is for the better.

POP OF VALVE

6 By "pop" of a safety valve is meant its instantaneous rise to almost full lift after the first tendency to move from its seat, or what is known as simmering, has taken place. A clean-cut pop is most vitally necessary in a successful safety valve. Without pop there will be a prolonged simmering of the valve as soon as the pressure has been reached for which the valve has been set, and the unduly pro-

longed simmering will soon score both valve and seat, particularly if the steam is superheated, thus producing a leaky valve. The pop of the valve in Fig. 1 will be determined by the diameter of the flange *A*, the amount of the distance *B*, and, to some extent, by the shape of the surface *C*. It is assumed, of course, that the valve has been ground to its seat and made tight. To design a valve of this type that will give a satisfactory pop, therefore, requires previous experience regarding the relations of these three variables, otherwise some experimenting must be done. When this valve is refaced and reground it is also of utmost importance that the original conditions of the working parts be restored. Gages for the original shape of both valve and seat, and the proper distance between the two, should be furnished the machinist repairing the valve, otherwise a misfit is liable to result and an uneconomical valve produced. A valve of this type requires a very long spring in order to give a satisfactory lift, on account of the small overhang of the flange *A* over the seating surface of the valve *D*, which limits the available additional lifting force required after the valve has started from its seat.

7 In the valve in Fig. 2 the pop will depend on the diameter of the flange *A*, the amount of the distance *B*, the depth *C* of the main pop chamber, and the size and number of the holes *D*. Of these four items *A* and *D* are determined by the manufacturer of the valve, and, if rightly proportioned in the first place, need never be changed. Items *B* and *C* are both variable, however, with the wear and refacing of the valve and seat. In this valve it is desirable to have the lip *A* wear down in the same proportion as the valve and seat. As this is very seldom the case, however, there arises the necessity for adjusting these features when the valve is overhauled. It should be noted that the depth of the pop chamber does not influence the pop as much as does the distance between the flange lip and the valve bushing. The former need be checked only roughly, while the distance *B* must be absolutely correct. An approximate value for *B* is 0.01 in. for the average size of marine safety valve.

8 The pop in Figs. 3 and 6 is decided by the three items, size of flange *A*, distance *B* and depth of pop chamber *C*. In these respects the valves are similar to the valve of Fig. 2, the remarks about which will apply to Figs. 3 and 6. One difference, and an important one, is that the distance *B* can be adjusted after the valves are assembled, by means of the blow-down ring *D*. Thus it would seem as if the necessity for the very close machining of the lip *A* in overhauling these valves would not exist. Such is not always the case, however,

as the adjustment of the distance *B* for a satisfactory pop may interfere with the adjustment for the desired blow-down. This will be explained in more detail later.

9 In the valves of Figs. 4 and 7 the pop is dependent upon the size of the flange *A*, the distance between flange and valve seat *B*, and the width of the ring-shaped opening at *C*. This difference between the two valves prevails, however, that in Fig. 4 the main deciding feature is the distance *B*, by reason of holding the ring area *C* comparatively large, whereas in Fig. 7 the pop will be determined mainly by the distance *C*, which is just large enough to let the valve flange *A* clear the inside of the adjusting ring *D*. Therefore, in overhauling the valve of Fig. 4, strict attention must be paid to the distance *B* in order to bring it back to its original value, by means of a cut from the wider side of flange *A* in case either the valve or its seat have been refaced, thereby lowering the valve and decreasing the distance *B*. In the valve of Fig. 7 only a rough check of distance *B* need be made. A suitable value for this distance in an average-size valve is $\frac{3}{16}$ in. The width of the ring area *C* has been fixed once in making the valve parts, and need never be changed in overhauling this valve.

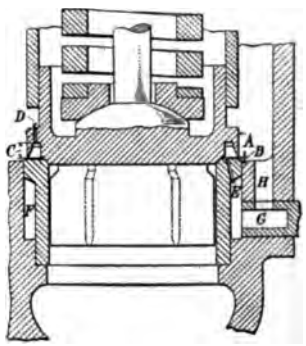

Fig. 2 Later Type of Valve with Adjustable Blow-Down

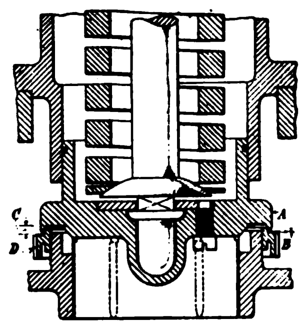

Fig. 3 Modern Safety Valve with Adjustable Blow-Down

10 In Fig. 5 the pop depends upon the distances *A* and *C*, which can be regulated by means of the adjusting rings *B* and *D*. The inner ring should be screwed very close to the valve disk, but not touching it. The outer ring furnishes the greater part of the additional lifting force required after the valve has started from its seat.

SIMMERING

11 The reduction to a minimum of the period of simmering before lift, which is very essential to the successful working of any safety valve and to the length of its useful life, can easily be accomplished in all the valves of Figs. 3 to 7 if the proper care is exercised. In the valves of Figs. 1 and 2, as previously pointed out, the reduction of the simmering is mainly a question of design and original adjustment, unalterable after the valve has been set up.

LIFT

12 It is known that all safety valves will give their highest lift at popping, the sustained lift being less than this by 10 to 25 per cent.

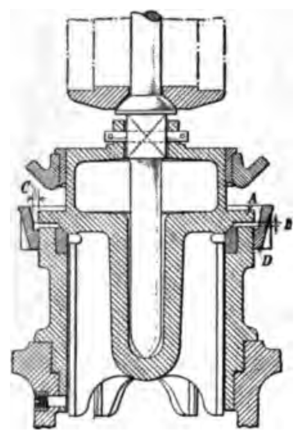

FIG. 4 RECENT TYPE OF MARINE SAFETY VALVE

The average lift of a 3- or 4-in. valve is about $\frac{1}{8}$ in. To increase this lift without interfering with the blow-down or resorting to abnormally long springs is a difficult matter. In order to obtain the maximum lift the valve spring should be made to contain as many turns as possible within the space available for it.

CLOSURE

13 In order to prolong the life of a safety valve it is necessary that its closure be accomplished with the minimum amount of shock. When a valve comes down on its seat each time with a heavy blow, both valve and seat will soon be distorted, and the valve will start to

leak. For obtaining a large discharge capacity a high lift is desirable, but this becomes a detriment if the closure is accompanied by shock. In this respect a valve with a beveled seat is generally inferior to the one with a flat seat, as the flat seat affords more of an opportunity for the steam to form a cushion at the moment of closure. Especially is this the case with the valve shown in Fig. 5, there being an excellent steam cushion between the valve disk and the inner ring *B* for checking the descent of the valve. Another valve which seems to give an easy closure, although having a 45-deg. beveled seat, is the one shown in Fig. 7. It is to be regretted that on neither of these valves are there available data on the length of their useful service without repairs upon which to base a more reliable conclusion re-

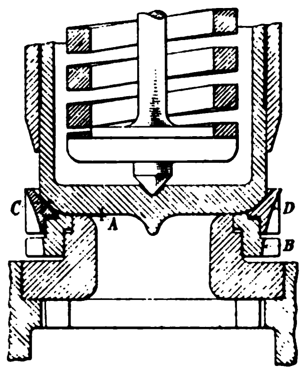

Fig. 5 Experimental Valve

garding this feature of the valve closure. As previously mentioned, both of these are new types of valves and not yet fully tested out in service. The tendency of a valve to chatter at closure is generally caused by too small a blow-down.

BLOW-DOWN

14 For adjusting the blow-down, or difference in steam pressure under the valve at popping and at closure, there are two different systems represented in the valves shown. Before analyzing these two systems it should be mentioned that in the valve of Fig. 1 there is no adjustment of the blow-down. In this valve, therefore, the amount of blow-down desired will have to be decided on in advance, by previous experience from similar valves, proportioning the valve

flange and the spring as well as the shapes of valve and seat so as to give the desired blow-down.

15 The arrangement in the valve of Fig. 2 for adjusting the blow-down is one probably familiar to most steam engineers. When this valve discharges, part of the steam passes through the holes *E* drilled in the bushing and into the chamber *F*, generally termed the secondary pop chamber, but more properly called the blow-down chamber. The only exit from this chamber is through the bushing *G* and the passage *H* communicating with the discharge space of the valve casing. By turning the bushing *G* and locking it in different positions the size of this exit passage for the steam can be restricted, thereby limiting the amount of steam passing through the holes *E* and so regulating the pressure under the valve lip *A*. Of course, the

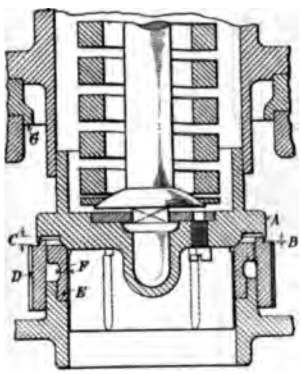

FIG. 6 MODIFIED SAFETY VALVE OF TYPE 3

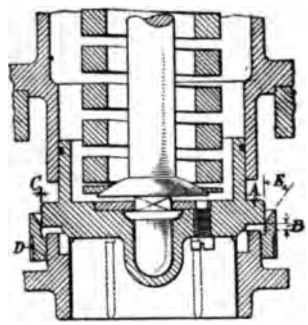

FIG. 7 ANOTHER MODIFIED VALVE OF TYPE 3

holes *D* in the valve flange will have a similar influence on the blow-down. The size and number of these holes are generally proportioned for a minimum blow-down of, say, 4 lb. A smaller blow-down than this should not be attempted in any safety valve as it only shortens the life of the valve and produces a tendency to chattering, as previously mentioned, with no additional advantages in the working of the valve.

16 In all of the valves of Figs. 3 to 7 the adjustment of the blow-down is accomplished by means of the adjusting rings *D*. Screwing these rings up will increase the blow-down and screwing them down decrease it. The shapes of these rings should be noticed. The rings of Figs. 4 and 5 are especially adapted to give a minimum obstruction to the flow of steam and so increase the discharge capacity of the

valve, both being excellent shapes. It has been especially claimed for the ring of Fig. 5 that in conjunction with the lower ring *B* it has the effect of reducing the lifting force at very low lifts and increasing it as the lift increases. This is a most desirable feature in any safety valve.

17 The ring *D* of Fig. 7 has been designed and tried out as an improvement on valves of the type shown in Figs. 3 and 6. It has already been pointed out that in these valves the main feature determining the pop is the distance *B*, also that the position of the ring *D* will determine the amount of the blow-down. Thus it will be seen that there are two factors to be considered in determining the proper position of these adjusting rings, and that each of these factors may require a different setting of the ring. In such a case it is necessary to take out the valve disk and machine its lip *A* until the proper distance of this lip over the adjusting ring is obtained, after first determining the position of the adjusting ring for the desired blow-down. This procedure involves tedious trials. Where gages for finishing the valve disk and adjusting ring are furnished by the manufacturer of the valve, it is a comparatively easy matter to machine these parts when making repairs. When the time comes, however, to overhaul the valve, it is generally found that all gages are lost or unavailable. In order to eliminate the tedious machine work in overhauling and adjusting valves like those in Figs. 3 and 6, the modifications as shown in Fig. 7 were devised. The lip of the valve disk has been cut away and the diameter of flange slightly reduced. The shape of the blow-down ring has been changed entirely, as shown, and made somewhat similar to the one in Fig. 4. The different action of the two rings *D* in Figs. 4 and 7 as regards popping has already been explained. The main feature in the ring of Fig. 7 determining the blow-down is the angle *E*. This angle, 36 deg., 30 min., has been found to give a range of blow-down of from 4 to 9 lb. for the particular valve on which it was employed, which was a 4½-in. valve. The laboratory tests on the popping and blow-down of this improved valve have given very good results.

DISCHARGE CAPACITY

18 It is well known that when steam under pressure is allowed to flow through an opening, the maximum discharge will be obtained when the lower pressure is equal to, or less than, 58 per cent of the higher pressure. When such conditions obtain, the pressure at the throat section, or most contracted part of the channel through which

the steam is flowing, will be 58 per cent of the higher pressure, regardless of how much the pressure in the discharge chamber will fall below this value. The weight of steam discharged can then be calculated from Napier's formula: *Flow in pounds per second = absolute pressure × area in square inches* ÷ 70. If this is applied to the safety valves shown in Figs. 1 to 7, it will be found that the valves of Figs. 1, 2, 4, 5 and 7 undoubtedly will have their throat section at the inner circumference of the valve seat. Therefore, the area of opening at this section will determine the discharge capacity of the valve, provided the resistance to the flow is not made great enough to raise the pressure in the valve casing above 58 per cent of the pressure under the valve seat. Making allowances for reductions of the free opening over the valve seat caused by guide wings, the following approximate formulæ, as given by some manufacturers, are obtained for calculating the discharge capacity:

$$W = 149\,lpd \text{ for a flat seat}$$
$$W = 105\,lpd \text{ for a 45-deg. seat}$$

where W = discharge in pounds per hour
l = lift of valve in inches
p = absolute pressure under valve, lb. per sq. in.
d = diameter of valve seat in inches

19 For equal lifts, then, the flat seat will give a greater discharge than the 45-deg. seat, which is obvious when it is considered how the 45-deg. seat restricts the throat section of the valve as compared to the flat seat. It should be mentioned here that the discharge capacity will be influenced to some extent by the smoothness of the approach to the throat section. On this account the valve of Fig. 5 should give an exceptionally good discharge capacity, which seems to be borne out by the meager test figures available. With valves of the shape shown in Figs. 3 and 6 it is frequently found that the actual discharge capacity will fall considerably below that obtained by the above calculations. This, no doubt, can be accounted for by the throttling of the discharge channel under the valve lips, especially at the lower lifts.

REPAIR AND ADJUSTMENT

20 Bearing in mind that a safety valve, to be successful and satisfactory, must not have too delicate an adjustment nor require too frequent overhauling, it seems advisable to alter the valve in Fig. 3 to the type shown in Fig. 7, where repeated attempts at obtaining

satisfactory operation of the old type have failed. Such alteration can be accomplished easily and cheaply, as the only new part to be made is the adjusting ring *D*, and the only part to be machined is the valve disk. The result will be an exceptionally rugged valve having no delicate parts, and one which can be easily overhauled and adjusted without expert assistance.

21 For grinding a valve to its seat the most satisfactory grinding material is powdered glass and machine oil. It is easiest and most expedient to grind a valve cold, and this will in most cases produce entirely satisfactory results. Only where repeated efforts at making a valve tight by these methods have failed, should hot grinding be attempted. The hot grinding is a trying performance to any workman and will only be an approximation to actual working conditions in any event, as a grinding at the actual temperature under which a safety valve operates is out of the question. Where all other methods have failed to produce a tight valve, the design shown in Fig. 6 will sometimes serve. By inserting the raised seat *E* and taking care to machine this so that its weakest section *F* will be between the old valve seat and the new, there is an opportunity for this raised seat to adjust itself to the valve, regardless of what expansions and contractions may be taking place in the metal of the old valve seat. Frequently this will produce a tight valve. The construction shown in Fig. 6 is an improvement on the valve shown in Fig. 3, but the same, or a similar design, can be applied equally well to most other types. In making this alteration to a valve it is necessary to make a new valve seat *E* and adjusting ring *D*, and to insert a distance piece *G* under the valve bonnet in order to raise the bonnet the same amount the valve has been raised from its former seat. The author has seen this method applied frequently and with great success.

22 Another point, often misunderstood in repairing a safety valve, is the amount of clearance to give to the guide wings under the valve disk in the valve bore. No attempt at a very close fit of these parts should be made, as such would only increase the chances of the valve sticking. A suitable clearance for a $4\frac{1}{2}$-in. valve bore is $\frac{1}{32}$ in. on the diameter, or $\frac{1}{64}$ in. on each side.

DISCUSSION

Charles W. Barnaby asked why all manufacturers of safety valves use a square rod for valve springs when it has been discarded for springs for practically all other purposes.

A. A. Cary answered Mr. Barnaby's question by saying that in pop-valve construction it was desirable to provide a spring which would offer the greatest amount of resistance to compression in the smallest possible amount of space.

A spring made from a wire or bar having a circular section undoubtedly would give the best and safest construction, providing there were no other important controlling requirements.

But, if we were to coil two compression springs to the same diameter and length, making one from round wire of a certain diameter and the other from a square wire with its sides of the same dimension (which springs would both fit into the same-sized cylinder or casing), we would find that the square-wire spring would require a greater force to compress it the same distance than would be required by the round-wire spring.

When a spring of this design was compressed, the principal stress set up in the wire composing the coils (which resisted the spring's compression) was torsional, and therefore in the case of the two springs just described, the difference in their resistance to compression was almost in direct proportion to the respective polar moments of inertia of the cross-section of the wire used.

Another necessary consideration in the design of pop-valve springs was to obtain a spring with the least possible tendency to side bending as the spring was compressed; or, in other words, all coils of the spring should remain equidistant from the vertical axis of the spring at all times during compression. It would be found that springs made from wire having a square section complied with this requirement better than springs made from round wire.

He had been engaged in the manufacture of springs some years ago, and was then called upon to make pop-valve springs from wire having round, elliptical and rectangular sections, as well as from square wire; but experience with these various sections had always brought the pop-valve manufacturers back to the use of square wire for their springs.

No. 1568

THE TALBOT BOILER

By Paul A. Talbot,[1] New York, N. Y.
Non-Member

It is the object of this paper to describe and illustrate the construction and operating principles of the Talbot boiler and to give a summary of results of tests made upon the boiler, with particular mention of certain features of design. The boiler is designed to meet the requirements of marine service.

GENERAL DESCRIPTION

2 This is essentially a boiler of the contra-flow type, water entering near the stack and leaving in the form of steam near the furnace. The circulation is through water tubes and is forced at a high velocity by means of a pump, doing away with a water drum. By preventing gravity circulation, the steam is drawn directly from the lower tubes, eliminating the steam drum. The boiler is internally fired, and hence self-contained. Feedwater and fuel burners are automatically controlled to adjust the boiler to widely varying demands for steam. The boiler should not be confused with "flash" boilers in which the heat-charged tubes furnish steam in direct proportion to the feed supply without regard to the control of the fire. By means of the high velocity, forced circulation and contra-flow principles, the evaporative capacity of the boiler is largely increased.

3 Any or all of the tubes may be readily removed and replaced without disturbing any manifolds, pipes or connections. Official tests recently carried out in the Navy Yard, New York, included a tube-renewal test of a 100-h.p. boiler. A tube was removed from the boiler under 250 lb. steam pressure and replaced by another tube in 61 seconds, the total period between the time of shutting off the fire and feedwater and the time of regaining full pressure and maximum capacity of operation being 3 min. 16 sec.

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CONSTRUCTION

4 An idea of the general construction of the boiler may be obtained from Figs. 1 and 2. Fig. 1 shows the front and Fig. 2 the rear

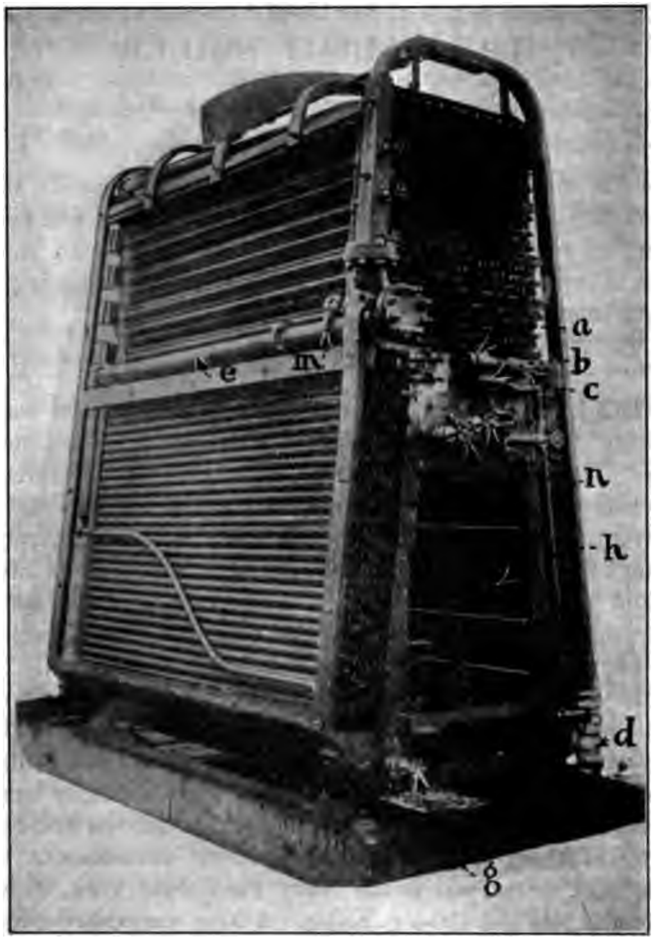

Fig. 1 Front of Boiler, Casing Removed
a Automatic regulator valve, *b* Oil valve, *c* Hand valve, *d* Main check valve, *e* Main steam pipe, *f* Separator, *g* Separator drain, *h* Combustion space, *m* Atomizer pipe, *n* Burner valves

of the boiler with the casing removed. Fig. 3 shows the complete boiler with its casing. The illustrations are of a boiler having an evaporation of 15,000 lb. of water per hour with ½ in. of water draft, 20,000 lb. per hour with 1 in. draft and double the normal capacity at slightly above 2 in. draft.

5 *The frame* consists of front and rear sections, roughly resembling horseshoes in shape, fastened together at top and sides by lateral

FIG. 2 REAR OF BOILER, CASING REMOVED
f Separator, *e* Main steam pipe, *g* Separator drain, *i* Feed pipe, *k* Trap

ties and at the bottom by the ash pan and additional ties. The stack ring and upper panel work are supported by framework bent to conform to the curve of the front and rear frames. For fastening the boiler in position, feet are provided which are bolted to the main frame through the ash pan. The door frame, of angle cross-section, is fastened at the top and bottom to the main frame, and serves as a support for regulating valves shown in Figs. 1 and 3 and described more fully later.

6 *The tubes* throughout the boiler are all alike. As will be seen in Fig. 4, the crucible steel header consists of two sets of overlapping

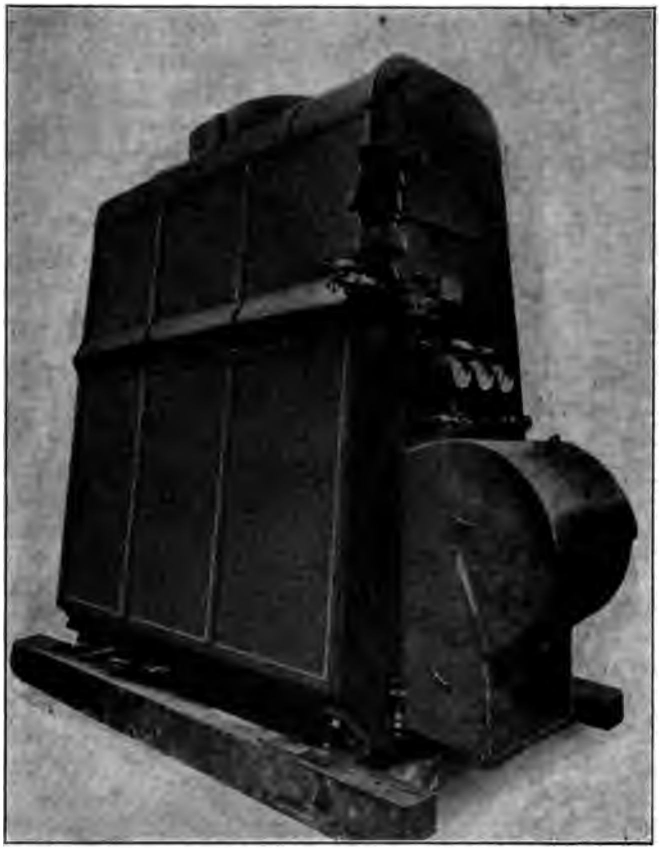

FIG. 3 FRONT OF BOILER WITH CASING

compartments, into one of which is screwed the open-end Field tubes, and into the other the generating tubes, there being an annular space between the two tubes. The end of each generating tube is welded together so as to close it. These closed ends are free to expand and are supported in front by perforated sheets of metal. The method of fastening the tubes into the headers is shown in Fig. 5. Both tubes are threaded with double the standard taper, which makes it easy to remove them. The fit is sufficiently tight to hold a pressure of 1000 lb. per sq. in., using standard-weight pipe.

7 Five sets of tubes with horizontal headers are placed above the combustion space, and on each side of the combustion space is placed another set of tubes with headers arranged vertically. The rear of the furnace is closed by a wall containing water-circulating passages connecting the tubes on the sides, as shown in Fig. 2.

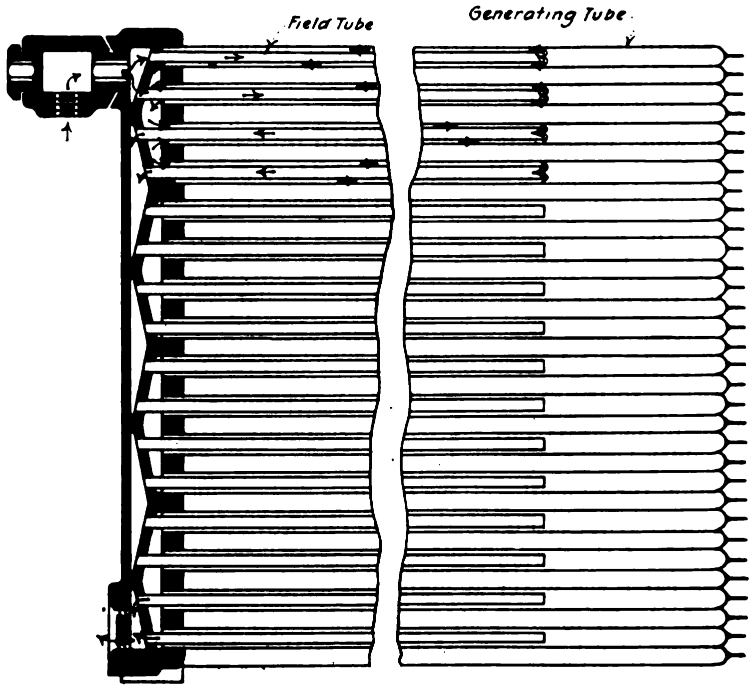

FIG. 4 DIAGRAMMATIC SECTION THROUGH HEADER AND TUBES
Arrows Show Course of Water Circulation

8 Between each set of headers is a *trap connection*, shown in Fig. 2. These traps perform the important function of preventing water from draining by gravity from one header to the next below until sufficient water is forced into the upper header. The formation of steam prevents the water from siphoning when circulation is at rest and the correct amount of water is thus maintained in each header, in readiness for full capacity demands.

9 *Accessibility.* There are three doors in the front of the boiler, which, when opened, give access to the free ends of all the tubes. Any tube can be removed by applying a wrench to its end and unscrewing it from its header.

OPERATION

10 *The circulation of water* is maintained by a pump, delivering the feedwater to the main check valve at the bottom of the door frame, from whence the water rises in a pipe and passes around the door-frame to the regulator valve. The water then passes to the rear of the boiler to the uppermost set of tubes. In order to understand the circulation through the tubes, reference must be made to Fig. 4, which shows a typical header and tubes forming a single section, the path of water through it being indicated by arrows.

11 Having passed through the first set of Field tubes and returned to the inner compartment of the header by way of the annular space in the generating tube, the water returns to the front of the boiler through a similar annular space in the next set of tubes and back to the outer compartment of the header by way of the inner Field tubes. This process is repeated until the water has traversed to the opposite end of the header, where it passes out, through a trap connection, to the header immediately below. Passing from header to header, the water has a general downward direction to the set of tubes which is on the left side of the furnace, as seen in Fig. 2. The circulation through all sections of tubes is the same.

12 Referring to Fig. 2, the steam or water from the set of tubes on the left of the furnace enters a rear wall which is partitioned in order to carry the water at high velocity from one side to the other until it is finally discharged, at the bottom of the right-hand corner, into the set of tubes on the right side of the combustion chamber.

13 During this passage the water has gradually changed to steam, and it now leaves the last set of tubes through the herringbone steam separator at the back of the boiler and enters the steam pipe, crossing to the front of the boiler where safety valve and steam nozzle are located. The separator drain passes to the front of the boiler where the valve controlling it is located.

14 *The length of water passage* from the point of entering the boiler to the point of leaving it varies with the size of the boiler. The passage in a 1000-h.p. boiler is 795 ft. long, 668 ft. of which is through tubes. The areas through which the water flows also vary with the boiler and are restricted to increase the velocity of flow.

15 *The friction of water and steam* by reason of the restricted areas is about 100 lb. at normal load. This friction increases as the load increases.

16 *The velocity through the boilers* in which larger tubes are used is much greater than when smaller tubes are used. Thus the velocity of the steam in the last stages of the boiler varies from 6000 to 12,000 ft. per min., depending upon the size of boiler tubes used. This

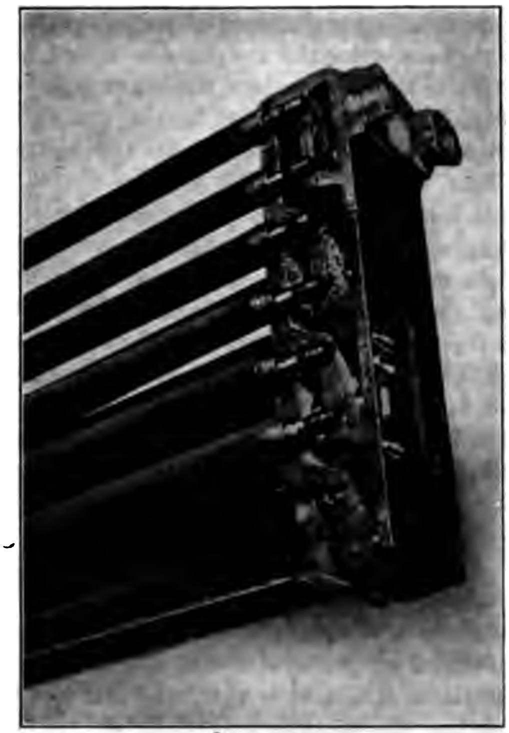

Fig. 5 Inserting Tubes in Header

velocity increases, of course, as steam is formed, so that the velocity in a large boiler is about 500 ft. per min. in the first and 12,000 ft. per min. in the last passages when in the form of steam.

17 *A high rate of evaporation* is obtained because of the high velocity of water through the tubes. The tubes which line the furnace evaporate as much as 30 lb. of water per sq. ft. of heating surface, while the average, including economizer and superheating surface, is 8 lb. per sq. ft. at rated capacity.

18 *The formation of scale* is prevented by the high velocity and consequent scouring action of the water, which keeps the solids in constant motion. The actual velocity great enough to prevent scale formation is not known, but it has been determined that with a velocity as low as 60 ft. per min. and very poor water, mud forms in the economizer section of the boiler. The water tested was alkali of 27 degrees hardness, such as is found in desert regions. With a normal velocity of 500 ft. per min., such as is used, a minimum velocity of 150 ft. per min. can be maintained at partial loads. It has been found that at very low velocities the deposits have been confined to the water portion of the boiler where the temperatures are between 100 and 150 deg. fahr. As the entire heating surface is active and subjected to these high velocities, feedwater purifiers are not necessary for this boiler.

19 *The soot blower.* The soot which forms on the economizer surface of the upper sets of tubes is blown from them by longitudinally directed steam jets situated near the ends of the tubes. Steam is supplied from the main steam pipe, and valves for the control of the jets are on the front of the boiler. The jets have proved a satisfactory method of keeping the surface relatively free from soot. The tubes at the sides of the furnace are practically free from soot, and it is probable that if any soot collects on them it eventually ignites. Exterior as well as interior surfaces of the boiler, therefore, are always clean, eliminating the periodic overhauling and cleaning.

THERMOSTATIC CONTROL

20 The expansion of one of the tubes of the last set at the side of the furnace under the varying temperatures of steam contained within it is multiplied by means of levers, and is utilized as an automatic thermostatic control of both water and fuel supply. Arrangement is made so that the operator may easily and quickly adjust the thermostat to meet the demands. The slightest rise in temperature of the outgoing steam causes sufficient expansion of the tube to open the regulating valve on the front of the boiler and admit a greater quantity of water, thus lowering the temperature of the steam. In case the operation of the thermostat has not prevented the rise in temperature, as might happen under partial load or insufficient water supply, it will cut off the supply of fuel to the furnace. At low temperatures, the supply of fuel is increased by this same thermostat.

21 *Details of the thermostat.* The expansion and contraction of the tube and the movement of the regulating valve are shown by

a gage which is arbitrarily graduated for convenience in indicating the relative temperatures, and may be calibrated by means of a mercurial thermometer so that the operator may judge from it the temperature of the steam. Thus it might be determined that with the gage reading $2\frac{3}{4}$ the temperature of the steam is 650 deg. fahr. Having determined the point for the temperature he desires, the operator can regulate the thermostat to maintain the gage at this point. This reading is checked occasionally, as changes in the frame, header or tube, due to wear or seasoning may make corresponding changes in the calibration of the gage, so that some other point registers the 650 deg. which the operator is trying to maintain. This temperature gage serves the purpose of the water column of the ordinary boiler. The temperature of the steam delivered tells the operator whether or not the supply of water is adequate, as it is obvious that with too little water, at full capacity, the temperature of the steam will rise and indicate this condition on the gage. Alarms may be attached, if desired, but are unnecessary, as the fire automatically is shut off before the boiler is damaged. When the steam temperature drops, the normal supply of water is attained.

22 *Hand valve.* The action of the automatic valve is fairly close, but due to the fact that the regulator tube is not instantly affected there occur periodic changes in temperature and pressure. A hand valve is provided for eliminating this action under normal loads. This valve is arranged to shut off completely the supply of water when closed, and is marked with arbitrary graduations from which the operator may determine at what point it should be set for the best results. When the hand valve is closed too much, it is obvious that the supply of water may be so limited that the temperature will rise to a point where the fuel valve will close. When the valve is opened too much, there will be a periodic variation in the temperature of the steam due to the lag of the controlling thermostat. The point should be determined, therefore, which will allow the regulating valve to be opened all of the time, preventing the periodic variation in temperature due to too frequent shutting down of the fuel valve. The accuracy of adjusting this valve determines the accuracy of temperature control under full-load conditions. Experience has shown that on a 25 per cent load the valve does not require adjustment, but the temperature of the steam will be much less due to an oversupply of water to a reduced fire. By readjustment of the valve the full-load temperature can be maintained at fractional loads. At less than 25 per cent capacity a readjustment is necessary to prevent the periodic

variation in temperature. The ordinary fireman can easily maintain a constant pressure and temperature by means of this hand valve.

OIL BURNER

23 A steam-atomizer oil burner is arranged at the front of the boiler near the top of the inflow of air for combustion. The air is heated and carried in a general downward direction by the air duct, a sheet-metal addition to the front of the boiler having an S-shaped channel. (See Fig. 3.) The quantity of air can be adjusted by air dampers in this duct. Highly superheated steam is used to heat and vaporize the oil and to promote the velocity of the hydrocarbon mixture which issues from the burner tips. Very heavy Mexican oil, 11 Baumé, can be used in this burner. A number of burner tips shoot the flame downward into a firebrick target at the lower rear end of the furnace. This refractory target or oven is designed to distribute the heat uniformly over the entire heating surface. Temperatures of 3800 deg. fahr. are obtained within this oven, but as few bricks are used, the upkeep cost is small.

24 The intensity of the fire is easily controlled by increasing or decreasing the volume of steam used in atomizing the oil. The flame has the effect of a large blow torch, and as the velocity is carried to an extreme, considerable noise is experienced. This is not a detracting feature in marine practice, as it is confined to the fire room.

FURNACE JACKET

25 As the heating surface surrounds the furnace on all sides, the insulation of the jacket is easily accomplished. The insulating material is applied in plastic form and fills the entire space between the sheet-metal casing and the tubes themselves. While this system is preferred, the various methods of applying insulating material in boilers having sheet-metal jackets may be used, or brick settings may be installed where space, weight and vibration are not points for consideration.

EFFECT OF FORCING

26 The thermal efficiency of this type of boiler is affected in much the same manner as in boilers of other types, although the drop in the efficiency curve is not so pronounced at higher draft pressures

and higher evaporative rates, due to the fact that when forced the economizer surface in the upper part of the boiler is correspondingly increased and the stack temperatures are relatively lower. More water is introduced, and the superheating surface is reduced nearly in proportion as the load is increased.

REPORT OF TEST ON TALBOT 75-H.P. BOILER

27 The following is from a report of a test made at the New York Navy Yard to determine the suitability of the Talbot boiler for submarine use.

Weights and dimensions. The steaming weight of the boiler was 4000 lb. and the overall dimensions 54 in. long, 36 in. wide and 71 in. high. The inside and outside heating surfaces were respectively 290 and 350 sq. ft.

Hydrostatic test. The boiler was subjected to a hydrostatic test of 1500 lb. for a period of one hour. The boiler leaked considerably at the fittings between the headers. The fittings were tightened and the leakage was practically reduced to a negligible amount.

Preliminary test. This test was run for a period of two hours in order to test various appliances used in conjunction with tests on the boiler.

Starting-up test. The boiler was started under the conditions which would prevail on shipboard. A quantity of wood saturated with fuel oil was ignited in the firebox. Steam began to form 4¾ min. after the ignition of the wood, and 1¾ minutes later the burner was ignited. Three minutes 19 seconds later the steam pressure in the boiler was up to 300 lb. per sq. in. The time required to bring the steam pressure up to 300 lb. in the boiler when starting up with a wood fire was found to be 9 min. 49 sec.

Test of automatic fuel and water valves. This test was conducted by shutting off the water gradually and noting the effect upon the fire, steam pressure and temperature and then noting this effect when the water was gradually increased. The test was conducted under steaming conditions.

Tube-changing test. At a steam pressure of 250 lb. per sq. in., temperature 720 deg. fahr., the throttle valve open and the boiler delivering approximately 80 boiler horsepower, a boiler tube was replaced in a period of 61 seconds. The minimum steam pressure reached during this operation was 25 lb. and at the end of the change was 35 lb. In 2¼ min. after the tube had been replaced the steam pressure was back at 250 lb. with the throttle valve open and the boiler delivering approximately 80 boiler horsepower. This short period of time required to change the tube can be attributed to the fact that all necessary preparations were made prior to the removal of the tube. It is probable that without any special preparation it would require a period of about 15 minutes.

Reserve-capacity test. The boiler was operated under normal conditions and after operating for a period of one-half hour, the throttle valve open and the fire shut down and feed valves under running positions, the automatic regulator valve was opened 4½ min. after the fire was shut down. The following readings were taken:

Time in Minutes	Steam Pressure	Steam Temperature	Stack Temperature
0	250	640	725
1	110	630	460
2	99	580	400
3	60	395	350
4	46	330	310
5	35	302	275
6	27	290	245
7	22	275	224
8	14	260	205
9.45	5-0	240	....

Docking test. The boiler was brought up to 250 lb. pressure and operated at this pressure for one-half hour. The throttle and oil valves were then closed, as would be the case in a boat approaching a dock. The pressure immediately rose 365 lb. At the end of ten seconds the throttle and oil valves were slightly opened and the steam pressure dropped to 250 lb. Five seconds later the throttle and oil valves were opened wide and the steam pressure dropped immediately to 130 lb Twenty-two seconds later the pressure was again at 250 lb.

Summary of results:

1	Draft, in. of water	1
2	Duration, hours	12
3	Steam pressure, gage, lb. per sq. in	250
4	Steam pressure, absolute, lb. per sq. in.	264.7
5	Steam temperature by thermometer, deg. fahr	548
6	Steam temperature, saturation, deg. fahr	406.1
7	Degrees of superheat	141.9
8	Specific heat of steam	0.59
9	Temperature of feed water, deg. fahr	64.7
10	B.t.u. in saturated steam	1202.3
11	B.t.u. absorbed	1254.1
12	Factor of evaporation	1.29
13	Weight of condensed steam per hour, lb.	2332.7
14	Equivalent evaporation, lb.	3016
15	Boiler horsepower delivered	87.5
16	Weight of feed water per hour, lb	2653.1
17	Total equivalent evaporation, lb	3430
18	Boiler horsepower generated	98.5
19	Weight of fuel oil per hour, lb	287.7
20	Actual evaporation per lb. of fuel, lb.	9.23
21	Equivalent evaporation per lb. of fuel, lb	11.94
22	B.t.u. per lb. of fuel (cal.)	19,553
23	Thermal efficiency, per cent	59.2
24	Mechanical efficiency, per cent	87.9

TEST OF TALBOT BOILER, NO. 1, TYPE "I"

28 The following test was made at Olean, N. Y., Sept. 3, 1915.

Boiler data: Length, 54 in.; width, 36 in.; height, 71 in.; weight of boiler and jacket, 3800 lb.; steaming weight, 4000 lb. Heating surface inside of tubes, 290 sq. ft.; outside, 350 sq. ft. Cold-water pressure test, 2000 lb. per sq. in.

The results of the test are as follows:

Time of Readings	Weight of Water in Lb.	Weight of Oil in Lb.	Oil Consumed During Period, lb.	Actual Evap. During Period, lb.	Equiv. Evap. During Period, lb.	Actual Evap. per lb. of Oil, lb.	Equiv. Evap. per lb. of Oil, lb.
3.45 P.M.	1208	650	....	....	....	....	....
4.00 P.M.	830	625	34	378	808	11.06	14
4.15 P.M.	435	593	33	395	530	11.95	16
	1277						
4.30 P.M.	862	561	32	415	557	13.00	17
4.45 P.M.	459	527	34	403	540	11 85	15
	1290						
5.00 P.M.	896	495	32	395	530	12.38	16
5.15 P.M.	502	452	33	393	527	11.90	15
5.30 P.M.	103	428	34	393	527	11.75	15
	1264						
5.45 P.M.	870	395	33	394	528	11.50	16
6.00 P.M.	469	362	32	401	537	12.20	16
Totals and averages		2673	298	3567	4800	12.00	16

Total heat in the steam from 32 deg. fahr., 1317.6 B.t.u. per lb.
Total heat supplied by boiler, 1304.6 B.t.u. per lb.
Superheat, 204 deg. fahr.
Latent heat, 838.9 B.t.u. per lb. (121 lb. absolute)
Factor of evaporation, 1.34

NOTE: The water is considered bad for boilers on account of alkali. Oil, 18,260 B.t.u. per lb. heating value, contained considerable moisture, as it had been standing uncovered in barrels through several rain storms and was very frothy. Draft was fair, due to a 50-ft. stack, no blower being used.

DISCUSSION

ALBERT A. CARY (written). When a new form of boiler is brought to the attention of one who has had an extended experience with various types of boilers, he will naturally recall the construction and operating principles embodied in these older types, in order to classify and understand better the features included in the new production.

After inspecting Fig. 4 in Mr. Talbot's paper, the resemblance of his boiler to the Niclausse boiler occurred to me at once. I had an opportunity to study this boiler when in Europe a few years ago.

The Niclausse boiler has the same Field tube arrangement; it also has front headers placed in vertical sections with front and back chambers in the header divided by a vertical central baffle and the two sets of tubes secured in the rear face of the header and in the central baffle, respectively. The designs of the two headers, however, are different, the Talbot header being more compact.

The Talbot boiler has no steam drum, but substitutes in the place usually occupied by such a water-storage receptacle a bank of tubes which performs the functions of an economizer.

My personal experience has taught me the desirability of having a water storage made a part of the boiler, which provides, to a certain extent, an immediate source of water supply to the steam-generating tubes when the combustion in the furnace is quickly increased to meet a sudden demand for steam.

Such water is usually heated to approximately the temperature of the steam, and with the feed pump quickly started to meet the urgent demand for steam, such high-temperature water, held in storage, has a tendency to counteract the chilling effect of the entering feedwater at a time when the greatest steam-producing effect is demanded. Further, the greater the mass of water held within the boiler, the less will be the fluctuating effect in steam pressure due to a sudden demand for steam followed by the introduction of a considerable amount of feedwater.

In the Talbot boiler we find ingenious means employed to counteract the doing away with water-storage space by the introduction of automatic regulating devices to control simultaneously the feedwater supply and fuel supply, and if both of these are needed for the successful operation of the boiler, it puts this form of steam generator out of the class that can use solid combustibles for fuel. Such means as are provided here to control the fuel-oil supply would be hardly applicable to coal-burning furnaces.

I cannot find that the Talbot boiler has increased its evaporation (even when using petroleum oil) above that obtained by other marine boilers. The author states that a maximum evaporation of 30 lb. of water per square foot of heating surface per hour in tubes next to the furnace has been obtained. The Niclausse boiler has shown an evaporation of 34½ lb. under identical conditions, except that coal was used for a fuel. On the other hand, the Talbot boiler occupies considerably less space per unit of power produced than does the Niclausse boiler.

As all the steam generated in the Talbot boiler is taken from a lower, submerged part of the boiler, my experience in studying the action of steam and water in the interior of boilers does not allow me to believe that solid water exists within *all* of the tubes of that boiler when it is in operation.

When in England a few years ago I had an opportunity to study conditions existing within a Belleville boiler, which also has a backward and forward circulation through its superimposed sets of tubes somewhat like the Talbot boiler, but uses return bends at each end of its tubes instead of Field tubes. By test it was found that only about 50 per cent of the interior of the Belleville boiler, while in normal operation, was filled with steam.

Judging from this and further related tests which I have made with other boilers, I think it very safe to assume that a considerable lower portion of the Talbot boiler is filled with steam during operation, and such steam-bathed surfaces can hardly be considered good water-contact heating surfaces, *i.e.*, efficient for the production of steam.

After a very long series of carefully conducted tests I have been brought to the conclusion that the contra-flow principle in operating boilers is excellent and highly desirable practice, providing it can be accomplished by simple means and without too great an expenditure of energy or by the use of complicated or inaccessible moving parts.

My experience in testing many different types of boilers and in studying the circulation occurring within them has not prepared my mind so that I can agree with those who argue the advantage of forcing all the steam generated in a boiler to its lower heating surfaces, against the natural upward flow of the steam after it has been formed.

Extended experimental work has taught me that the highest results for heat transmission in boilers are secured by removing the steam from the heating surfaces almost as rapidly as it is formed and thereby permitting the greatest possible area (within such heating surfaces) to be bathed and wetted with solid water.

When we have steam in contact with the interior of our heating surfaces and the much hotter furnace gases on the opposite side of these heating surfaces, we certainly are not generating much more steam at such positions.

In this type of boiler that prevents the steam formed in the interior of the boiler from escaping from the heating surface and accentuates the evil of this condition by forcing it to travel "backward" through the entire lower heating surface to the very lowest tubes (the volume of steam increasing constantly as it continues in its downward course), we certainly are not improving conditions required to facilitate rapid heat transmission through this lower and most valuable area of the heating surface.

I know from actual experience that such conditions — forcing the steam to the lower heating surfaces — are not necessary to secure the advantage due to the contra-flow principle of operation.

I have tested water-tube boilers where the comparatively cool feedwater is admitted to the upper part of the boiler, and this feedwater, gradually heated as it advances along its path of travel and being deprived of all steam as rapidly as it is formed, flows continuously through simple open passages, downward to the bottom heating surfaces, while the removed steam passes freely and directly upward to the overhead steam drums.

Under such conditions I secured the highest rate of heat transmission that I have ever been able to obtain; and further, when such boilers were forced far above their rated capacity, the loss in efficiency was small.

We know that when water is evaporated into steam, the dissolved solid matters which it contains will be precipitated along with such mud or other floating matter as is carried into the boiler with the feedwater. As no mention is made of a mud drum or other place of deposit for such solid matter, I would like to ask what becomes of such precipitated solid matter in the Talbot boiler.

I know from experience that as long as a high velocity is maintained through the tubes of a boiler there will be but little deposit made on the interior of the tubes; but unfortunately there are times when the boiler is not in operation, such as during noon hours, nights, Sundays and holidays, and at such times the floating matter will deposit on the tubes and frequently hold so fast upon these surfaces that the renewed circulation will not sweep it off, and this is followed by the baking action of the furnace gases against such heat-insulated surfaces, and the hardened scale will soon be formed.

C. A. Carr[1] (written). The first trials of Talbot boilers for the Navy were made about three years ago at the Navy Yard,

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Puget Sound. In these cases particularly good results are said to have been shown by a reduction in the quantity of oil burned and an increase in speed of the boat in the case of the installation on a small steamer which was used to run between the Navy Yard and the torpedo station in Puget Sound. It is believed that in this case the economy was directly caused by furnishing dry or superheated steam to an engine which had previously been very wasteful through the use of wet steam. The use of the Talbot boiler appears to furnish a ready, reliable, convenient and inexpensive means of supplying steam of any desired degree of superheat for small steam-power installations in which oil is used as a fuel. In such installations this boiler has evidently been very successful, and the field for such installations is a wide one.

About a year ago I witnessed a test of the largest Talbot boiler which has yet been built. It is rated at 500 h.p., and probably can be forced to generate steam for developing more than 1200 i.h.p. at the engines. This boiler has been in service for nearly eight months on a tug in New York Harbor. From my knowledge of the service performed by the boiler, and the repairs and changes which have been made, I am inclined to believe that larger boilers of this type may be built and used successfully.

Perfect control of the feed system for this type of boiler is essential, for if the supply of feedwater stops, the boiler almost instantly stops forming steam. On account of the small reserve capacity of steam and the friction of the steam and water in the boiler, the feed pumps usually furnished with small marine plants will be found unsatisfactory for use. Feed pumps fulfilling specifications supplied by the manufacturer of the boiler should always be used in such plants.

The automatic control valves are intended for use when the engines are running steadily. When this is the case the control given by the automatic feed valve is excellent. When working to bell signal the oil supply valves and the feed valves are necessarily regulated by hand.

I consider the oil-burning system furnished with the Talbot boiler unsuitable for use with large boilers, and wasteful of both oil and steam with small ones. The oil and steam valves cannot be readily regulated, and require constant attention when the engines are not running steadily. It is evident that firemen must have a certain amount of training before they are qualified to manage a boiler which is without a water gage, and the additional problem of managing a new and unhandy oil-burning system is rather too much for most naval firemen. In my opinion the adoption of an improved oil-burning system would hasten the adoption of the boiler for use on the small steamers of oil-burning battleships.

The tests of the 75-h.p. boiler at the Navy Yard, New York, were made for the purpose of determining the suitability of the boiler for installation on the 50-ft. steamers which are being furnished for some of the latest battleships, and for determining its suitability for general naval use and not particularly for the purpose of determining the evaporative efficiency of the boiler. During the test, of which a summary is given, there was considerable trouble with the oil-burning system and with the oil pump, and during the latter part of the trial, adjustments of the oil supply were frequently made by hand. Also, when the test was about half over, one tube blew out and was replaced. The detailed report shows that conditions during the trial were not at all uniform and that data were taken but once an hour. For a test of the evaporative efficiency of a boiler of this type, a run of an hour, with readings every ten or fifteen minutes, should be sufficient.

R. C. Carpenter said that a study of the White boiler plant in connection with the Talbot boiler should prove of interest. He made reference to a paper which he presented in December, 1906, and which appears in Vol. 28 of the Society's Transactions, dealing with his tests of the White automobile boiler.

D. K. Warner[1] (written). I should like to ask Mr. Talbot if any experiments have been made since those conducted by the U. S. Geological Survey on the temperature of the tube metal in a boiler of natural gravity circulation. Those tests showed that the tube temperature remained practically that of the water in the tubes. As the capacity for receiving heat depends only on the temperature difference between the tube and the gases, and the velocity of the gases, I would take exception to Mr. Talbot's statement that "the high rate of evaporation is obtained because of the high velocity of water in the tubes." Inasmuch as with ordinary circulation the tubes keep as cool as the water, it would seem that the high capacity of this boiler is due more to the high furnace temperature, the absence of baffles and other special features. I would also like to ask if the periodic temperature variations mentioned would be great enough to injure a turbine.

[1] Sheffield Scientific School, New Haven, Conn.

Because of its freedom of expansion this boiler seems admirably suited for very high capacities. May I ask the author then why he limits himself to a draft of two inches of water. Using 20 lb. of air per lb. of oil, a draft of 24 inches of water requires by theory less than 1½ per cent of the power developed in the turbo-generator. Electric fans are built of 50 per cent efficiency, so that it would actually require but 3 per cent of the power. High capacities will be essential on the new ships of such great power and the flue temperatures can be kept down by adding a few layers of tubes.

JOHN C. PARKER spoke of the difficulties he had experienced with his early investigations of boilers of the general type of the Talbot boiler, particularly with the scale formed by the insolubles in the boiler water. He showed how Belleville, who attempted for a long time to build a water-tube boiler without any drum, finally installed the drum, later introducing a gravity circulation independent of the pump, and at last succeeded through perfected automatic control devices. If the Talbot boiler succeeds, he believed it would depend on the efficiency of the automatic devices for its success.

THE AUTHOR. I cannot agree that the U. S. Geological Survey test referred to by Mr. Warner is under all conditions correct. When the evaporative capacity gets beyond the limit made possible by such circulation, it is an easy matter to apply enough temperature in the furnace of a boiler to drive the water out of the tubes or to form a stratum of highly superheated steam or gas insulating the heating surface from the water to be heated, with a result that the heating surface is burnt by the intensity of the furnace temperature. Therefore, when evaporative rates arrive at 20 or more pounds per square foot of heating surface, such results are certain. It is believed that parts of the Talbot boiler have a very much higher evaporative rate and that the circulation is responsible for keeping the heating surface lining the furnace from being damaged by a higher furnace temperature, than is possible with the ordinary boiler. The advantage of this is that the high velocity of flame and extreme localized furnace temperatures (provided the boiler will stand these highly localized temperatures) provide the most compact boiler and greatly reduce the volume required for combustion in the furnace. This is due to the velocity and to the fact that the flame in the Talbot furnace is permitted to impinge on the tubes quite violently.

Therefore, the higher evaporative rates, it is believed, are due to high velocity of water and steam circulation in the tubes, and to the fact that the water at such evaporative rates in ordinary boilers will not remain in contact with the tubes or other heating surface, and that this only applies to low evaporative rates in ordinary boilers with a gravity circulation. It is likely that more startling results will be accomplished when draft pressures of 24 in. are experimented with in Talbot boilers.

Despite Mr. Cary's observation, the similarity of the Talbot boiler to the Niclausse boiler is so remote that a comparison is hardly possible. The only similarity is that both boilers have a combination of inner and outer tubes. In fact, the first water-tube boilers made had a similar construction. Many types of water-tube boilers have used the double-tube system of natural circulation. The Talbot boiler in principle, due to its having a single continuous channel and forced circulation at high velocity, overcomes the difficulty of the very slow circulation characteristic of the Field or double-tube system in water-tube boilers, thus making it possible to arrange the tubes in compact horizontal rows and have a certainty of high velocity produced by mechanical means.

There is always a value in a large water storage. However, a large water storage and quick-steaming boilers are the two extremes. The steady action of boilers such as the Scotch marine type and the reserve capacity of this type of boiler, particularly for intermittent loads, cannot be questioned, but it requires time to heat up a large body of water as well as to cool it. Therefore, any boiler with a large quantity of water stored within it cannot be either compact or capable of raising steam quickly. It is a combined generator and storage tank. The Talbot boiler is a highly efficient generator with but little stored energy. Steam is made as it is used. Doing away with large pressure containers makes it possible to carry high steam pressures with safety. Thus it has not only the advantage of steaming quickly but also the advantage of being capable of higher pressures with resultant engine efficiencies, and it is extremely safe against dangerous explosions.

While it is possible that records are shown where boilers of other types may have been credited with higher evaporative rates per square foot of heating surface, our experience with the very largest express-type boilers shows that in ordinary service such boilers have thus far been eliminated from competition in weight and space occupied for a given horsepower. In fact, the comparison is so startling that it is not necessary to go into figures to see the difference between one boat equipped with a Talbot boiler and another equipped with the best and lightest boilers of other types available in the U. S. Navy, the Talbot boiler occupying for a given horsepower one-third the floor space and one-fifth the cubic volume required by such boilers as Yarrow, Bureau Standard, White-Forrester, Normand, and other three-drum types.

The possibility of water preventing the heating surface from being overheated, the water having a gravity circulation, as presented by Mr. Cary, is, I believe, a matter which can be easily proved by experiment; and this will show that no boiler with a gravity circulation can stand the extreme heat possible with an oil-burning fire without a certainty of destroying the boiler's heating surface where this heat is applied direct to the surface.

Mr. Cary points out that in boilers having Field tubes, scale is apt to deposit while the boiler is at rest. It might be added that when the circulation is stopped the heat is also stopped; that the period of time in the handling of a Talbot boiler in which the heat is applied and the circulation at rest would certainly be but a few seconds, surely not a few minutes. The scale that would form in those few seconds is possibly swept away. I do not know just what happens, as no scale has yet been formed, even when using the worst water which scales other boilers quickly.

I agree with Mr. Cary as to the necessity for a perfect feedwater control, as well as fuel control for boilers of the Talbot type. In fact, not only must the control be perfect but the feed system also, as the demands for reliability are much greater on the feed-pumping system than with ordinary boilers, due to the fact that the feed pressure is much higher and that intermittent service is impossible, as there must always be available feed pressure ready for the automatic devices to use in supplying the boiler. I might add that we have had more difficulty with this one particular than with any other in the application of our boilers for commercial use.

Captain Carr stated that the oil-burning system used in the Talbot boiler was unsuitable for larger marine use. I agree in this particular. We use the steam-atomizing type of burner, which is built in our boiler, finding that it appeared to be easier to train inexperienced operators to handle this type of burner successfully, and at the time when these comparisons were made the mechanical burners were probably not as highly developed as at the present time. I do not see any reason why mechanical burners cannot be applied which will give better results than the steam-atomizing burners we have furnished for a number of years past in our boilers. We hope to be able to conduct experiments on mechanical burners at our new test plant at Plattsburg, N. Y.

After reading Professor Carpenter's paper on the White automobile boiler, it would appear that the steam velocities in the last end of the White automobile boiler were possibly much higher than available in the smaller sizes of Talbot boilers, in view of the friction of the steam through the film passage produced by the inner- and outer-tube construction. The White boiler, however, even having an extremely high velocity on the end or steam portion of the heating surface, must have had a very low water-circulation velocity in the economizer portion of the boiler, which, when using alkali water, probably accumulated mud, and perhaps a small amount of scale was deposited between the economizer portion and the superheater portion where the heat is sufficient to bake the mud on to the heating surface. Our experience with our own type of boiler using alkali water and where the velocities of water were too low in the economizer surface, convinced us that the speed used in a boiler having the same areas throughout its entire length resulted in such deposits in the water portion.

The merit of the contra-flow principle, as well as the use of superheated steam, is very convincingly shown in the small unit tested by Professor Carpenter and confirms the performance of the Talbot boiler in marine installations, but I believe when properly designed and working in combination with uniflow engines, a fuel consumption of less than 0.5 lb. will be obtained per horsepower, due to the added efficiency of the engine when using high pressures and temperatures, together with the efficiency of a boiler operating on the contraflow principle.

In reply to Mr. Parker, I would say that apparently his experiments did not go far enough in velocities or he would not have experienced difficulties with the formation of scale in boilers of the forced circulation or Talbot type. The elimination of scale or deposit of solubles and insolubles is a matter entirely of the circulation of both water and steam through the boiler. In fact, the circulation can be carried to such an extreme that even the steel is rapidly worn away, and due to the velocity, many turns, etc., the various solids both carried in solution and in suspension in the water, are a thoroughly consistent mixture throughout the boiler from the time the water enters to the time steam is discharged.

This foreign matter remains in microscopic particles and is carried through the entire system to a point where it is permitted to settle — usually the hotwell in a condensing system. In fact, the only deposits of any kind thus far found after a number of years of use are deposits in the hotwell. No ill results have been encountered in the engine or running parts, probably due to the fact that these fine particles are more than 100 times as rare in the steam as they are in the water passing through the feed pump.

I agree with Mr. Parker in the necessity of reliable devices for controlling the fuel and feed in such boilers as those of the Talbot type. The Talbot feed control is very robust, being operated by one of the boiler tubes which is both sensitive and powerful. My experience is that delicate controlling devices should not be applied to the boiler construction. I am convinced that there is no limitation to the size of unit which can be built and that the demand for higher pressures and also for superheated steam will call for boilers either having very small steam drums or pressure containers, or boilers of the Talbot type in which no large pressure container is used. In fact, a Scotch marine boiler having the same strength as the Talbot boiler would have to have a shell two feet thick! Structural defects in making steam drums and boiler shells to have the same strength as the ordinary Talbot boiler would make such boilers impracticable.

Replying to the query of Mr. Ambrose E. Dean, I would say that the tube referred to in Par. 3 as being removed in 61 sec. was chosen at random by the officers in charge of the test, though all of the tubes were accessible. It is not, however, always possible to change a tube in as short a time as this, especially if the tubes twist, or stick together, or are badly burned.

Mr. Sherwood F. Jeter asked why the design of the header was changed from passing the water through the outer tubes to passing it through the inner tubes. This was done merely as a matter of simplifying the construction, and for no reason which affected the operation of the boiler.

It might be summed up that the following are the advantages and disadvantages of boilers of this type: The advantages being 1, Compactness; 2, Efficiency; 3, Ease and quickness of repair at low cost; 4, Elimination of the use of boiler compounds, periodical cleaning periods, handhole and manhole gaskets, and tube-cleaning apparatus; 5, Light weight and safety in carrying high pressures and temperatures; 6, Quickness of steaming; 7, Continuous operation with a minimum of attention. The disadvantages being 1, Reserve capacity; 2, Necessity for high pressure and reliable feed pumps; 3, The knowledge of the operator or someone available to understand the burning of oil and of the setting of the automatic devices when out of adjustment; 4, The necessity of renewing tubes while under way, due to the elimination of periodical cleaning and repair periods and to the higher evaporative rates employed. This disadvantage can be easily overcome by installing a double-boiler unit which then makes possible an uninterrupted continued service; 5, The necessity of using steel valves and other modifications occasioned by the use of superheated steam.

At the same meeting at which the foregoing paper was brought up for consideration, a paper was presented by John Clinton Parker on The Downflow Type of Steam Boiler, at the request of the Boiler Code Committee. As a description of this boiler and the results of its performance are readily available, the paper is not included in this volume; pamphlet copies, however, may be obtained from the Society.

No. 1569

THE PENCIL ELECTRODE METHOD OF WELDING FOR BOILER JOINTS

By E. A. Wildt, Scranton, Pa.[1]
Non-Member

This paper has special reference to the welding of joints of drums and not boiler shells as the latter term is commonly understood. The trend of the times is towards that type of boiler in which all the tubes are bent, particularly in the large units such as those at the Commonwealth Edison Station in Chicago, the Delray Station in Detroit, the Ford Automobile Factory and the Solvay Process Company. Since the dimensions of the boiler rooms are growing out of all proportion to the size of the engine rooms, and every item making for a decrease in the size of parts so as to reduce the room for the boilers is in demand, much higher pressures will be resorted to — an item for making the reductions required. Drums are to be used up to 60 in. in diameter, and in order to bear a pressure of 300 lb. or 500 lb. the thickness of the plate will be very close to 2½ in.

2 With regard to making the joints in such a drum, is it not more feasible to weld them instead of employing the usual butt strap? There are several methods of making this joint by spot welding, and that which seems to have forged its way to the front is the pencil form of electric welding, which is now fairly generally used in steam-boiler work, although as yet recognized only for low pressures. The weld made by this process is not so hard as others of the autogenous kind.

3 In a weld, two pieces of metal heated to the proper temperature are united into one solid piece. Success of the process depends on bringing the pieces of metal to the proper heat. For this purpose we have the oxy-acetylene torch, the thermit process and the electric arc, the last of which is the form of modern welding particularly referred to here.

[1] Lackawanna Boiler and Grate Company.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

4 Electricity is used only to supply the heat, and in the pencil method only just enough heat is obtained to accomplish the joining of the two metals. No reference is here made to any particular design or set of apparatus: some are built to use a uniform voltage, others to use uniform amperage or uniform wattage. All autogenous welding is accomplished by adding new metal to the joint to be made, and it is only in the pencil electric-arc method that positive incorporation of the added metal with the metal to be joined is secured. The opposite was the case in some recent failures in other forms of electric and gaseous welding wherein fluidity of both the added metal and the pieces to be joined is a necessary condition. By the electric-pencil method fluidity is avoided and only just enough heat is used to make the plate and the electrode plastic, and there appears also to be an action in it which in the direction of the current tends to pull the metal from the electrode to and into the plate when just at the proper heat. This is so much in evidence that welding can be carried on overhead without the metal dropping upon the operator.

5 The temperature in the added metal in the gaseous and electric carbon type wherein fluidity is a condition approaches 2800 to 3000 deg. fahr., while in the electric-pencil method the temperature in the metal being added is not more than 1500 deg. fahr. As a result the added and the adjacent metal in the weld is not rendered so hard as would otherwise be the result. This is also proved by the fact that, while cutting can be done with other methods, no cutting can be done with the metal electrode.

6 This point that the temperature of the arc is so high, so hot, that there is danger of the metal becoming vaporized, is answered by the fact that the conditions surrounding this particular form of spot welding are analogous to and the same as for forge welding as carried on by the everyday blacksmith at his anvil; there he has a fire very much hotter than the pieces to be welded are required to be heated to; in fact, it must be so; there must be a considerable surplus of heat for quick action; the blacksmith watches and if through carelessness the pieces are overheated, he says they are burned and spoiled and has to begin over again.

7 No other form of welding has the characteristic this one has, wherein there is an automatic action which prevents overheating, actually showing that in this regard it is equal if not superior to forge welding.

8 The form of welding approved in the A.S.M.E. Boiler Code, known as forge welding, entails in its operation the production of big

expansion strains, because the whole seam and the seam only is made at a welding temperature, producing an upsetting of the plastic metal by the unexpanded portion of the adjacent metal, so that when the forged welded seam has cooled off, the adjacent unexpanded metal produces tensile strains of very considerable strength, tending to pull the welded portions apart as it contracts, to the extent of close to $\frac{1}{8}$ in. per ft. of the seam. In comparison with this, the metal-electrode pencil method is a great improvement, because due to the very small area of metal heated the expansion strains are but fractional and may be considered negligible. Both the approved forge welding and this method of welding which is hereby submitted to the Boiler Code Committee for approval are exactly alike in the particular that the metal is not heated in either beyond the point just necessary to produce welding; when it comes to expansion strains they are less in the latter, and in both methods these strains in the weld improve with age.

9 Although the electric heat reaches an estimated temperature of 6500 to 7000 deg. fahr., in this process the metal wire does not have time to reach this temperature before it is added to the plate or in the usual groove which is to be filled with the welding metal. As fast as the metal wire becomes just plastic, the pencil must be advanced towards the work, or the arc gap will become too long for the electric arc to maintain its circuit. The distance needed for the arc does not amount to much more than $\frac{1}{16}$ in. because the voltages used are low, rarely exceeding 60 or 70, and failure on the part of the attendant to maintain this distance by constantly advancing the pencil is met at once by the extinguishing of the arc, because the gap becomes too long for it to maintain itself.

10 Only in this process is carelessness practically eliminated, both as to overheating and heating any considerable area, and the heated area is confined to the smallest dimensions of any; therefore the expansion and contraction strains are smallest. The wire forming the electrode only gets red hot at the point, showing the very localized character of the heat, the rest of the wire remaining black; while in the carbon form of electric welding, the carbon gets very hot from the point up to the holder.

11 Tests have shown that for pressures of 500 lb. per sq. in., and with plates of $2\frac{1}{2}$ in. or similar thicknesses, this method of welding makes a better joint than straps and rivets. The maker of such joints can always know by the hydraulic test whether his work is done perfectly or not. Test after test shows there are no leaks; all one

has to do to insure a perfect job is to secure an operator willing to do a good job, pay him well, and it is fair to say that it is then practically impossible to make a defective weld by this method.

12 In electric carbon-arc welding of rolled stock, the metal in the weld cannot have the same properties as that in the original piece; it may have the same tensile strength but it will not have the same elasticity. This is a limitation in *any* welding process, but in this particular process the metal in the weld is changed the least of any, and in fact shows a tendency towards a fibrous condition. The metal of the weld can be controlled by the kind of metal that is added; low-carbon steel will make the weld more ductile, high-carbon steel will

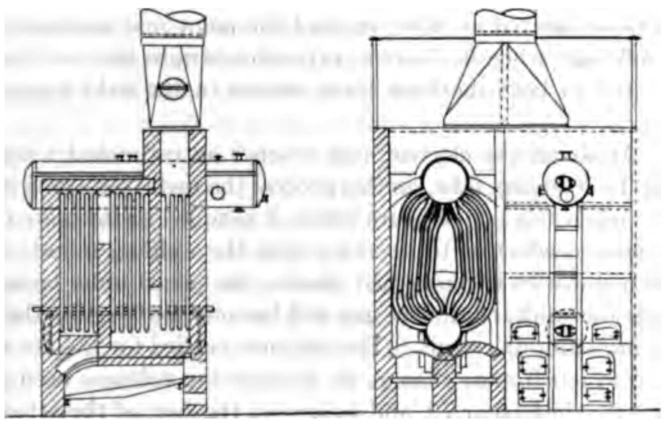

FIG. 1 BATTERY OF TWO 100-HP. WATER-TUBE STEAM BOILERS. THE WELDED-HEAD JOINTS ARE VISIBLE OUTSIDE THE BRICK SETTING

make it higher in tensile strength. A test piece made up entirely of the welding wire showed an elongation of 16 per cent.

13 In order to be sure that a joint so made will be stronger than the plate, and last indefinitely, it is only necessary to keep on adding new metal until the cross-section on both sides amounts to more than the plate itself. This can be carried to extremes, and may as well be, just filling the groove, usually V-shaped, the extra metal to lap over on each side $\frac{3}{8}$ or $\frac{1}{2}$ in., and made in bulged form, both inside and outside of the drum, taking on the form somewhat of a butt strap joint.

There has just recently been put into service, with a view of trying it out in actual practice, a small water-tube steam boiler (Figs. 1 and 2) of the vertical 2-drum type, with all the tubes bent tubes, the drums in which have not a rivet in them. Heavy tests have been applied, and there is not the slightest doubt that the men who have to do with the erection of this boiler do not anticipate any danger to anyone from it. Of course, it is realized that the construction has not been approved, but it is necessary for someone to take a stand and bring it to a head. Without something to show and to test, there will be no basis on which to ask for an approval. The situation is somewhat analogous to the man who wants to obtain a job as a stationary engineer, — not having a license he cannot obtain the job, and not having the job he cannot obtain the license.

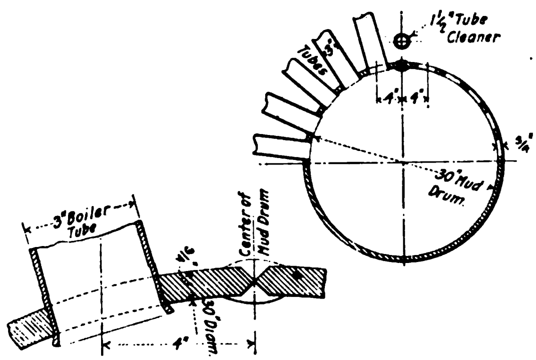

Fig. 2 Cross-Section of Mud Drum, Showing Electric Weld Which is Not in Contact with Fire

DISCUSSION

John C. McCabe thought the temperature of 1500 deg. fahr., rather low, and asked the author how he eliminated the possible stresses set up internally in the weld. As he understood the problem, if there was a differential for each degree difference in temperature between the different plates, there would be an internal stress of about 195 lb. The safety of a vessel could not be determined by the hydrostatic test. It was well known that hydrostatic pressures were seldom allowed anywhere near the elastic limits of the metal, and in the tests and investigations of failures made, he had found the welding material varied, the per cent of elongation running from a fraction of one up to 11, and in view of the bendings or flexures that occurred in the best-formed cylinders, he failed to see how an autogenous vessel, as we understood the problem now, could be considered a safe one.

Christopher H. Bierbaum asked if the great difficulty with all autogenous welding for boiler plates was not the fact that so much of the plate adjacent to the weld was heated, the elastic limit of the plate reduced, and its elongation increased. The test pieces exhibited showed that very fact. There was no break in the weld. We knew that autogenous welds could be made as strong and stronger than the original metal, and especially in the case where the weld was left a little thicker than the body of the metal itself. But did not the internal strains set up and the heating of the plate adjacent to the weld decrease the elastic limit of the metal in the neighborhood of the weld, and induce conditions which should be very carefully determined before any theoretical conclusion can be drawn?

P. A. E. Armstrong.[1] I do not think the forge process of welding is the only process valuable for boiler work; it is not as reliable as could be desired because of the difficulties of thermal disturbance in the metal in the vicinity of the weld. Thermal disturbance is brought about by two things, time and heat. In the vicinity of the weld the grain of the steel is enlarged and the tensile strength of the metal has fallen about one-third. I have conducted over a thousand tests on welds in Sheffield, England, in working up a high-class steel suitable for welding, and found that with ordinary 0.30 to 0.40 carbon steel it was impossible to get more than about 60 per cent of the original strength, yet the breaks did not occur in the weld but some two or three inches away from the hammered area. In every instance the break occurred because of an enlarged grain.

The crystal grain of the metal, providing the thermal disturbance has not reached the point of incipient fusion, can be refined by subsequent heat treatment, but this is hardly applicable for boiler work. I think it is absurd to talk of annealing the shell of a boiler 30 ft. in length. When annealed, this shell would expand and buckle in all directions. If this boiler had riveted joints, the expansion and contraction would be so great that a movement would be set up at the riveted joint, and no amount of calking would give a tight joint; in all probability it would augment the looseness of the rivets.

The oxy-acetylene or gas process generally is a very good one, but the thermal disturbance in the metal outside the weld is very similar to that present during forge welding. The electric carbon-arc welding is worse. The electric bare-wire welding, known as the metallic-pencil welding, overcomes thermal disturbance to a greater degree, but the deposited metal in the weld is distinctly cold-short. This cold-shortness could be improved by annealing, but this is quite

[1] North American Company, 30 Broad St., New York.

impossible under boiler conditions. By duplicating steel-bath conditions, however, we get a fusion process, where the fused metal has a structure which is practically as good as the original steel.

If you take a bare-wire electrode and coat it with a large quantity of slag, it is possible to melt this electrode so that the fusion takes place under the slag and the deposited metal would have all the characteristics of fine-structure cast steel of a given carbon content. Such an electrode has been developed and is extremely suitable for the welding of boilers and pressure tanks generally. The exterior slag coating of this electrode has the effect of localizing the heat. The metallic core is fused so rapidly that there is practically no

Fig. 3 Bare-Wire Electrode

thermal disturbance in the vicinity of the weld and complete fusion takes place.

Fig. 3 shows a bare-wire electrode. A globule of metal is just leaving the end of the electrode, to be passed across the arc and deposited upon the metal to be welded. The incandescent gases immediately underneath the electrode are very nearly neutral; at the outside of the arc flame the burning gases are extremely oxidizing. It is here that the damage takes place. A very interesting experiment can be conducted to prove it. If a bare-wire electrode is fused, a crater is formed immediately beneath the fusing end of the electrode. If the circuit is broken and the arc extinguished, then at the bottom of the crater there is a complete absence of oxide of iron, whereas on the top edge of the crater and right over the deposit a layer of about 0.01 in. of oxide of iron is present, which proves that in the center of the metallic arc there is a neutral place. If the flame is examined spectroscopically, its oxidizing nature can be very quickly traced, and the neutral zone can be discerned in the center of the flame when the outside of the arc flame is slightly disturbed.

The slag electrode in operation is shown in Fig. 4. The end of the electrode is in actual contact; in the bare-wire case there is a space of about ⅛ in. between the fusing end of the electrode and the work. There is a complete absence of the arc flame effect, and the incandescent slag is passing off from the end of the electrode on to the work. The atmosphere of the slag electrode is practically neutral, as the

FIG. 4 SLAG ELECTRODE

vapor is composed of vaporized slag and not of highly incandescent atmosphere. The voltage across the arc is higher than that of the bare-wire system, because the vapor offers a greater resistance to the path of the current, although the arc is shorter and should take only about half the volts to get across, if both arcs were atmospheric.

Fig. 5, 90 magnifications, shows manganese steel of 12 per cent deposited upon forged manganese steel of a like content. There is no thermal disturbance. The diffusion between the original and the added metal is very complete, showing an entire absence of oxide. The cementite in the added metal occurring in the globule formation and very evenly distributed over the mass, there is an absence in the deposited metal of austenitic needles.

Fig. 6, 90 magnifications, shows 0.125 carbon deposited upon 0.65

carbon. There is complete diffusion, and the carbon of the original steel is saturating into the lower carbon of the added metal. There

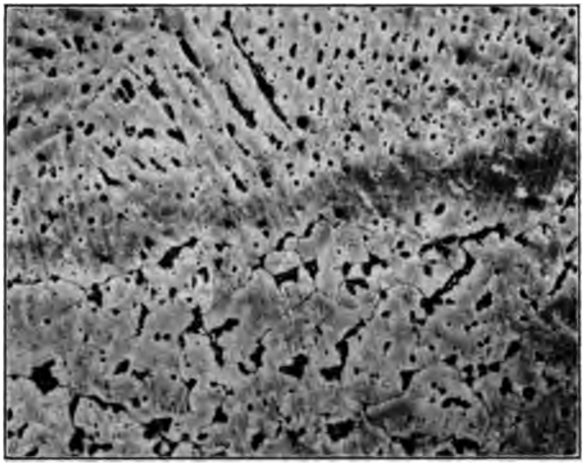

Fig. 5 12 Per Cent Manganese Steel on 12 Per Cent Manganese Steel, Magnification 90

is no sign of decarburization of the original steel at the weld, proving beyond contention the complete absence of oxidation during welding;

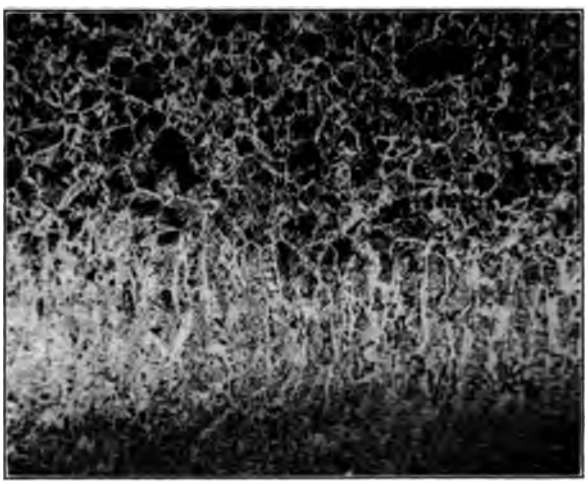

Fig. 6 0.125 Carbon Steel on 0.65 Carbon Steel, Magnification 90

therefore, as before stated, the slag arc electrode is melted in a neutral atmosphere. There is a complete absence of thermal disturbance immediately adjacent to the weld; the grains of pearlite and ferrite are the same size at the area of diffusion as they are half an inch under the weld, the time factor playing a very important part.

As it is impossible, in a general way, to anneal welds in boiler construction and so normalize the grain, it is essential that the time factor be reduced as much as possible. Fig. 6 is an illustration of this. The heat was at least 6000 deg. cent., the time something less than a second; hence the grains did not have the time to change and assume the size and formation of the superheated temperature. The metal in the vicinity of the weld is quite as strong as it was before welding, and the added metal has all the characteristics of fine-grained cast

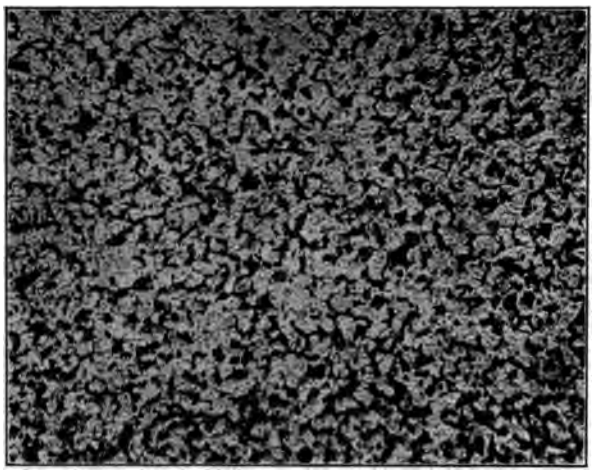

Fig. 7 Structure of Added Metal, Magnification 250

steel, in this instance 0.125 carbon. The structure of the added metal is given in Fig. 7, 250 magnifications. It is taken about half an inch above the line of diffusion. The grain is of small size and excellent structure, the pearlite and ferrite grains being very consistently arranged.

Table 1 is taken from a Report of the Department of Commerce and Labor, Steamboat Inspection Service, dated January 27, 1916, on the tensile strength of four samples of slag welding for marine boilers submitted.

Sample No. 1 is the original steel of about 56,000 lb.; it broke at 56,680 lb. Sample No. 2 is a plate cut in half, welded and machine flush; this broke at 56,530 lb., 2½ in. outside the weld, proving that the weld was stronger than the original metal and that there was no thermal disturbance in the vicinity of the weld. The elongation is 25 per cent, but this shows absolutely nothing, because the sample started to neck 2½ in. away from the weld, and the reduced area resulting from this necking caused the fracture. Sample No. 3 is a plate cut in two, welded and reinforced ½ in. down the entire length of one side. This sample broke in the weld and the tensile strength was 56,680 lb., identical with sample No. 1, which proves that the weld was of about 56,680 lb. tensile strength. The elongation was 7 per cent, and as the sample broke in the weld the necking occurred there, and it is quite reasonable to suppose that the elongation in the weld was somewhere near 7 per cent. Sample No. 4 was a plate reinforced on one side and then the welded metal entirely machined off, leaving

TABLE 1 TENSILE TESTS ON SLAG-WELDED PLATES

Numbers on plates	1	2	3	4
Thickness of samples, in.	0.494	0.494	0.743	0.521
Widths of samples, in.	1.00	0.991	1.00	1.023
Strain at which each sample parted, lb. per sq. in.	28000	27500	42120	31890
Strain per sq. in. of section, lb.	56689	56530	56680	59840
Reduced thickness of sample, in.	0.685	0.660	0.948	0.852
Reduced width of sample, in.	0.282	0.300	0.725	0.419
Reduction of area, per cent	60.9	59.5	7.6	33
Length of straight part in center of test piece, in.	8	8	8	8
Elongation, percentage of	29.6	25	7	14

the original plate at its original thickness. This plate was prepared so as to find out what effect the thermal disturbance had upon the metal adjacent to the weld.

As will be seen, the tensile strength of the metal has been increased and the ductility reduced. This is to be expected from the slight thermal disturbance taking place, as the added metal was as thick as the original steel. The increased tensile strength of the original steel is probably due to a slight hardening, or addition of the amorphous area between the crystals, or perhaps to surface tension of the crystals. The results according to the test sheet show at least the great tensile strength of the welds made by this slag electrode.

Corrosion and welding are closely associated. It has been proved that corrosion is electrolytic in character, the positive pole of the small galvanic couples being highly corroded and the negative pole being practically free. When testing a piece of boiler steel for polarity, it is shown that there are numerous places where corrosion can be set up, due to electrolysis; all mechanical work on iron and steel will immediately start corrosion due to electrolysis, the electrolyte being supplied by the atmospheric moisture. A simple experiment will prove this. If a small portion of a piece of steel, neutral across its entire area, is hammered and tested for polarity, a voltaic circuit is present, the stress from the hammered portion being electropositive and the original steel being electronegative; therefore it follows that every rivet head, calked edge or hammered portion of the boiler is electropositive to its shell generally. Thermal disturbance and small mechanical work will set up polarity, and such a weld will be distinctly electropositive to the surrounding metal. This is particularly noticeable in machine-welded flue tubes in locomotives; 90 per cent of the corrosion taking place will always be present when the added length of tube has been made to recover short tubes. I would like to advance a theory for the immediate electropositive effect of stressed metal. I believe that local growth of the amorphous area between the crystals resulting from strain is responsible, as annealing locally strained areas removes polarity and equalizes the amorphous contents. Autogenous welding is distinctly electropositive to the surrounding metal, because the added metal is less pure, containing magnetic oxide of iron and other impurities. An oxy-acetylene or gas weld made on a pressure tank or boiler may be badly pitted by corrosion. A weld by carbon arc corrodes to a greater extent, and welds by the bare-wire pencil method are certainly no better. Tests can be easily made by means of a millivoltmeter.

The deposit by the slag electrode is so pure that the added metal by this process is quite electronegative to the surrounding metal and to a very large degree obviates corrosion at the weld. If the subject of corrosion is borne in mind, I am tempted to say that forge welding will not be permitted; some fusion process will be adopted wherein the corrosive influence of an electrolytic circuit is to a very large degree restricted.

VICTOR MAUCK [1] (written). In all processes of welding it is necessary to raise the temperature of the metal to be welded to the point of fusion. Given a neutral flame free from non-combustible impurities, such as sulphur, nitrogen, etc., the arrest of the heating process at the exact point of fusion, and a uniform contact of the parts, a perfect weld of practically equivalent strength, section for section, to the adjacent metal would result. However, there are so many factors involved over which we have but indifferent control that this result is

[1] President, John Wood Mfg. Co., Conshohocken, Pa.

rarely attained. It is therefore necessary first, to rate the efficiency of the weld based on average manufacturing practice; second, to provide an ample factor of safety. A sufficiently high hydrostatic test pressure should be specified to insure ample minimum strength; and I lay particular stress on high test pressure.

In a riveted joint the fundamental operations are uniform and of a mechanical nature, affected but slightly by the human equation. The weld, on the other hand, has a much higher theoretical strength, but is *all* human equation, hence the importance of the high test pressure. A weld will vary widely in strength throughout its length, but this variation is less than in a riveted seam, which has no strength between the rivets. Welds cannot be made commercially (except by the Thomson method of induction electric welding) without more or less crystallization of the metal, which will lead to eventual failure under vibratory action if provision is not made for a sufficient factor of safety. It is possible to surround the processes with reasonable precautions, in the public interest, and at the same time allow them that latitude for development they deserve, and which is conceded them abroad.

The process of electric welding by induction (Thomson method) probably attains the nearest to the theoretical possibilities of any weld we have. The operation, being entirely mechanical, is uniform and under perfect control. There is no crystallization of the metals and the human equation is practically eliminated. The heat in this process is generated within the stock itself, radiating to the surface, as opposed to the application of a very intense external heat, as in the oxy-acetylene flame or electric arc. In the latter cases the surface is overheated before the body of the metal reaches welding temperature, with resultant burning and crystallization. In the Thomson process 100 per cent welds are the rule rather than the exception.

The Author. In reply to the several persons who entered into the discussion of my paper, I would say that I do not think the term *autogenous* properly applies to this particular form of welding, because it so closely follows the cycle of conditions of ordinary forge welding, whereas in autogenous welding, fluidity is a necessary condition.

In reply to Mr. McCabe, the temperatures in question are estimated from the color. In metal-pencil electric-arc welding just the same putty-like condition occurs as in ordinary forge welding. The estimate may be either above or below the actual temperature, and is used more especially to direct notice to the fact that fluidity is not a condition present in this last-developed form of spot welding except that known as the "quasi arc."

As to the elimination of the possible stresses set up in the weld, the figures as quoted by Mr. McCabe cannot apply to this form of welding. In this case the area is very much less, and being free from the highly expanded condition of fluidity (in many cases almost reaching vaporization), the differential for each degree of temperature is different and the internal stress less. However, no attempt is made to eliminate internal stresses, nor is this necessary, because these stresses are at a maximum only when new, and are constantly changing as time elapses. This is true of any of the later forms of welding wherein the bulk of the work is always cold and the part worked on is made hot. The point to recognize here, and this is fairly well explained in the paper, is that in this form of welding these internal stresses are the least of any and are consequently more readily and sooner neutralized. With this consideration, therefore, no differential per degree of temperature can be used.

The determining of the safety of either a riveted or welded vessel is accomplished by the hydrostatic test, and this is perfectly trustworthy when accompanied by hammering while the test is on, the test being applied to the customary amount, that is, one-half more than the pressure under which the vessel is to work.

It is to be expected that the welding material will vary widely in any form of welding except the approved form, in which it may vary some, but not widely, and except the form that the paper defends, and Mr. McCabe is justified in his conclusion regarding this; but I would suggest that the metal-pencil form of electric-arc welding should not be made to carry the burden of the failures of other forms known as autogenous welding, since the former places the art on a newer, entirely different and safer basis.

In reply to Mr. Bierbaum, the disturbance of the adjacent metal is dwelt upon in the paper and was shown in the lantern slides used at its presentation, and these ought to form a conclusive answer. It is true that in all other forms of welding wherein fluidity is resorted to the disturbance is considerable, and one slide used by Mr. Armstrong in his discussion showed the adjacent metal practically destroyed. I wish to dwell strongly on the fact that in this metal-pencil electric-arc method no such condition exists, and the adjacent metal is not disturbed to a fraction of the extent of that which takes place in gas and carbon-arc welding and in the form of welding recognized in the A.S.M.E. Boiler Code and known as forge welding, and this is due to the extremely localized character of the process and also its great rapidity of action — it does not have time enough to overheat so as to create the disturbance referred to.

In reply to Mr. Armstrong, his discussion practically embodies a paper in itself and describes a further improvement in this particular form of welding, viz., a metal-pencil electric arc, but one in which fluidity obtains as a condition; but this occurs to a certain degree, beyond which it does not go, and is controllable to a nicety, approaching an automatic action of its own similar to that described in the paper, which can be depended upon to prevent vaporization or any carelessness regarding overheating exactly in the way I have described, and altogether analogous thereto. In analyzing Mr. Armstrong's extended discussion, however, I find in his references to the chemistry of these welds that he mentions the oxidizing which occurs around the crater when the metal pencil is fused. Now to obtain fusion a high voltage must be used with electrodes of this character, and an abnormally long arc maintained. In his slides this was shown, and he states "about $\frac{1}{8}$ in." as the length of such arc, but it is safe to say that it must be more, since the slides themselves showed practically full-size dimensions — see Figs. 3 and 4.

While these metallic arcs require a higher voltage than the arcs obtained otherwise, the voltages used with the pencil arcs are normally low, from 60 to 70, and with such voltages an arc cannot be held with a wire in size equal to 0.161 in. diameter, or No. 8 B.W.G., more than about $\frac{1}{16}$ in. Therefore, fusion under such conditions cannot take place, and the piece of plastic, putty-like metal flies for the plate, being constantly pulled into it, and consequently does not have time to become fused and the oxide formed does not amount to enough to deserve any considerable attention. However, the little that is present in what Mr. Armstrong terms the bare wire is almost entirely absent in the slag-covered wire, and though a predetermined limited amount of fusion occurs, the deposited metal shown in the slides exhibits a remarkable similarity to the metal of the plate it joins. These slides also showed that the least disturbance of the adjacent metal and the added metal is, as he says, "just the same as ingot steel." Taking his discussion in a general way, it is a strong endorsement of the metal-pencil electric-arc method of welding, and shows it to be a remarkable and highly efficient improvement in the art.

Mr. Armstrong introduces for the first time the question of corrosion as being associated with welding and lays considerable stress upon it. In this regard I wish to say, however, that I have seen many pieces of forge welding wherein no corrosion has ever appeared; thousands are in evidence, such, for instance, as those in chains, wagon tires and innumerable things of everyday occurrence. Test pieces which I have furnished the Boiler Code Committee of the A.S.M.E. and which have now been in their possession about a year, welded with the pencil method, show absolutely no indication of corrosion. However, I claim that in the several other methods of welding referred to in this discussion, wherein fluidity takes place, disturbances are set up which produce conditions wherein the question of corrosion may be a consideration, but not in the form of welding described in the paper, nor in the improved form which Mr. Armstrong advocates, nor in the form approved in the A.S.M.E. Boiler Code.

In reply to Mr. Mauck, if he means that the metals "in all processes of welding" have to be raised to the point of fusion and no more, that is, to just a plastic condition, then his discussion is not a criticism of the metal-pencil electric-arc welding, since that is exactly what takes place therein.

If there are no leaks when a moderate hydrostatic test to one and one-half times the working pressure is applied, it shows that there are no porous or spongy places — that the joint is solid. Then why a higher test? I do not think it necessary or advisable to carry the testing pressure to extremes and so unduly strain the vessel.

In conclusion, let me say that with the knowledge we have in our possession, derived from much investigation and from many experiments and tests, all reduced to tables and curves, we know just the right voltage to use, just how much current to employ, just what are the right-size pencils, and what should be the direction of the current; we have also learned how to utilize to advantage other factors such as magnetism, slag protection, and purifying agents. The process now contains nothing haphazard or problematical; it has been reduced to an exact system of working, and may, therefore, be considered as belonging in the list of things that have been settled.

No. 1570

THE DEVELOPMENT OF OUR FLEET AND NAVAL STATIONS[1]

BY W. L. CATHCART, PHILADELPHIA, PA.
Member of the Society

In his opening remarks Mr. Cathcart stated that his subject was so broad that he would attempt only a rapid review of its salient points. He thought it scarcely necessary to say that the whole question of naval strength was one of compelling interest to engineers. The European conflict had shown in many striking ways that war by land and by sea was very largely but a matter of applied science, of physics and chemistry, and chiefly of engineering in all its branches. Years ago Theodore Roosevelt had said that the naval officer of our time was fundamentally a "fighting engineer," and this description was wholly accurate with regard to the structures and mechanisms of naval war, though of course he had added functions as a strategist and tactician.

The United States had need of a great navy, because, in the first place, it was the richest and, owing to its vast extent of coast line, the most vulnerable of all the great Powers. And, second, like a modern Atlas, it staggered — diplomatically and militarily — under the weight of certain national policies which, while just, were as world-irritating and war-breeding as any that history had known — the Monroe Doctrine and the Neutralization of the Panama Canal, for example.

The elements of naval strength, said Mr. Cathcart, were (1) the fleet — its ships and men, and (2) its shore stations — navy yards at home and naval bases in our island possessions — which docked, repaired and equipped the ships and from which the fleet might strike. Manifestly, the location of these stations with regard to our possible battlegrounds of the future was of primary importance, and a brief study of the strategic situation of our eastern and western

[1] Brief outline of an address delivered at the Annual Meeting, December 1916, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. The address was printed with but slight abridgment in THE JOURNAL, February 1917, p. 101.

coasts, in this respect, would show, among other things, the urgent need of repair facilities for dreadnaughts on the South Atlantic and Gulf Coasts, and of the fortification and equipment of our West Indian and North Pacific bases.

Mr. Cathcart then discussed certain features of interest alike to naval officers and to engineers, such as the probable 8-mile limiting battle range, the superior marksmanship of our dreadnaught fleet — brought about by the adoption of an effective method of fire control, and the possibilities of modern guns and powders.

In a consideration of our naval strength, after giving certain figures of the relative dreadnaught strength of England, Germany and the United States, Mr. Cathcart drew attention to the unbalanced condition of the fleet, its great need of additional cruisers, and its even more pressing need of destroyers and an adequate fleet of coast-defense submarines. The United States needed a great navy, not for aggression, but to keep the peace and to exert what Mahan called "the silent force of sea power." From every viewpoint of strategy and of common sense, the conclusion was inevitable that we should keep permanently in each ocean a battle fleet strong enough to defeat decisively any probable enemy there.

Mr. Cathcart's address was profusely illustrated with lantern slides and made a profound impression upon those who heard it. Among the illustrations of special interest were strategic charts of the North Atlantic, the Caribbean and the North Pacific, furnished by courtesy of the U. S. Naval War College.

The discussion which followed was participated in by Rear-Admirals John R. Edwards and Bradley A. Fiske, by President D. S. Jacobus and President-Elect Ira N. Hollis, and by Prof. M. E. Cooley and Messrs. Carl G. Barth, M. A. Stone and F. G. Coburn.

No. 1571

HEAT TREATMENT OF WROUGHT-IRON CHAIN CABLE

By W. W. Webster,[1] Bremerton, Wash.
and
E. L. Patch,[2] Portsmouth, N. H.
Non-Members

With Foreword by F. G. Coburn, Mem.Am.Soc.M.E.
Naval Constructor, U. S. Navy

The following paper on the thermal treatment of wrought iron is a restatement of the graduation thesis on that subject prepared by W. W. Webster and E. L. Patch, Assistant Naval Constructors, U. S. Navy, at The Massachusetts Institute of Technology.

The subject was suggested by the writer, as a result of experience at the U. S. Navy Yard, Boston, Mass., in the manufacture of chain cables for naval vessels. Prior to July 1, 1914, all chain cables for the navy were manufactured at that yard, by hand, after the fashion of the old English chainmakers, still followed in other chain shops in this country. Experiments had been under way for several years, looking toward a steam-hammer process, and such a process had, at that time, been developed; satisfactory to the extent that it effectively and cheaply welded the chain, but unsatisfactory in that the chain, apparently perfect, would not meet the breaking-strength requirements.

It became suddenly necessary, in July, 1914, to make the process work, in order to supplant hand manufacture. The writer, just at that time, came into direct charge of the work. The process seemed to be good, but it was mystifying, indeed, to see apparently perfect chain snap off under test like cast iron. It seemed as if an *addition* to the process was needed rather than a change, and heat treatment suggested itself. There was no literature available on the heat treat-

[1] Puget Sound Navy Yard.
[2] Portsmouth Navy Yard.

For presentation at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

ment of either iron or low-carbon steel. In fact, this thesis appears to be the first work of its kind. A number of engineers familiar with the art of heat treatment were consulted by the writer; but none of them knew of any work having been done which would be of assistance.

It was peculiar that *hand*-welded chain was successful under test; yet we knew that it was not so thoroughly welded as the *hammer*-welded chain. The latter was very thoroughly hammered, too, and kneaded, so that it should be stiff and strong. In this very stiffness the trouble finally was found.

The distribution of stresses in a studded link under tension is such that there are maxima of bending moment at the ends of the axes of the link and maxima of shear at the quarters. It is well known that iron is weaker in shear than in tension. It was observed that the hammer-welded links always failed in the quarters; that while the hand-welded links stretched freely, the power-welded links would not stretch much prior to breaking.

The following hypothesis was then formulated by the writer to explain the above phenomena: that the hammer-welded link was so stiff that the shearing stress could build up in the quarters to such a degree that the link would fail by shearing, whereas the hand-welded link was soft and ductile enough to deform under the shearing stress, failure occurring later due to a combination of shear and tension when a higher applied tensile load was reached.

There was no pyrometric equipment available; so in the first experiments, which were necessarily crude, the writer gaged temperatures by eye. Test doublets were heated to a temperature believed to be above the upper critical point, cooled in air, and pulled, giving satisfactory results as regards stretch, character of fracture and ultimate tensile strength. Heat treatment was plainly the answer to the problem.

The writer therefore proceeded to equip a laboratory and to equip the shop for temperature measurement. After some experimenting, a method of heat treatment was evolved which gave fairly good results. But these results were not always consistent, and while the plant was put on a manufacturing basis, still there were very puzzling questions arising. It was considered very desirable, indeed, to find out really why heat treatment was required for power-forged chain and not for hand-welded chain, what the very best heat treatment should be, how much heat treatment wrought iron would respond to, and, in general, to develop working standards for shop practice.

Messrs. Webster and Patch took up the task as their graduation thesis, being assisted in the work later by J. J. Crowe, Physical Metallurgist, who came to the work from the Bureau of Standards by courtesy of the Director, Dr. S. W. Stratton, and of Dr. G. K. Burgess.

F. G. Coburn.

MANUFACTURE OF POWER-FORGED CHAIN

The various operations involved in the process of power-forging chain are briefly as follows:

Shearing. The bar of round stock is rolled from the storage skids on to rollers which guide it through the shears and against the stop which gages the length of the "bolts." After cutting off, the bolts are packed in special baskets for transfer by crane to the scarfing furnace.

Scarfing. One end of the bolt is heated in a special oil furnace for a distance of about a foot and the end is then bent and upset by a single operation in the upsetting machine. During the same heat the bolt is scarfed under a 2500-lb. steam drop hammer, using special steel dies. The "flash" or web is then removed by a trimming press. The operation is repeated for the other end and the bars are packed in baskets for transport to the bending-machine furnace.

Bending. The scarfed link is heated throughout to about 1100 deg. cent. in a special oil furnace, from which it is swung by a special jib crane to the hydraulic bending press. Here it is bent by wiping it around a mandrel having the shape of the inside of the chain link. This operation leaves the links practically closed and it is necessary to pry apart the scarfs with a crowbar in order to thread the links in the chain.

Welding. The link is first preheated, threaded into the end of the growing chain, and the scarfs closed under a heavy hammer. It is then brought to a welding heat of about 1350 deg. cent. in a special oil-burning chain forge and welded on special "dolly dies" under a light hammer (250–350 lb.). To give the link the proper shape it is brought to welding heat a second time and finished in the dies of the heavy hammer (1800–3000 lb.).

Trimming. The last welding process leaves a "drop-forge flash" inside and outside on the welded end of the link which is trimmed off by hand.

Studding. The drop-forged chain stud, which is inserted to preserve the shape of the link, increase the strength and prevent kinking of the chain, is held in place with the link on its side under the steam hammer and pinched in place by a light blow of the hammer.

Heat Treatment. This is accomplished by loading the chain on to a steel flat car, which is run into a long annealing furnace fired by oil

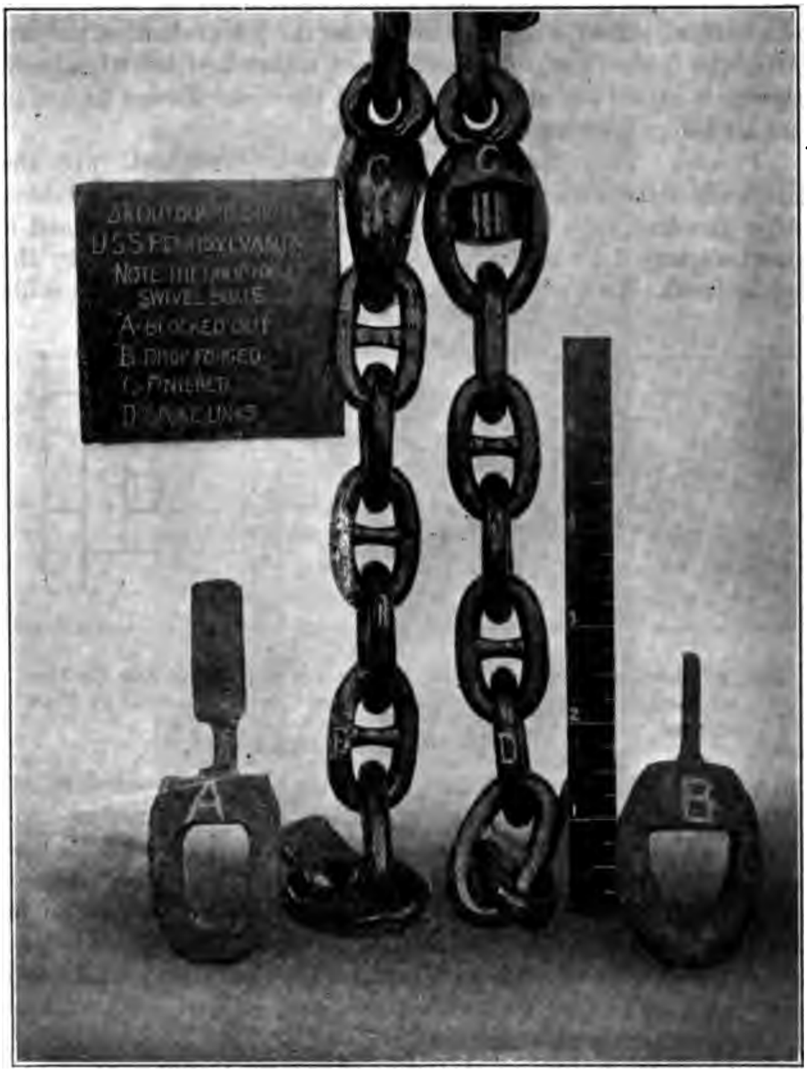

Fig. 1 Sections of 3¼-in. Shots showing Power-Forged Links and Drop-Forged Swivel Boxes

burners. The temperature is brought evenly to about 950 deg. cent., well above the upper critical point, as determined by the indications of nine base-metal thermocouples distributed about the furnace, one couple being placed in a bolt of iron under the pile of chain on the car. After the desired temperature has been maintained for 10 minutes the car is hauled out of the furnace and the chain allowed to cool in the air before proofing.

Proofing. Each shot of chain is given a "proof test" with the hydraulic testing machine up to values given in the chain tables. After proofing, each link of the shot is examined for defects and if unsatisfactory links are found they are cut out and replaced by the repair crews. In addition to the proof test, a breaking test is made

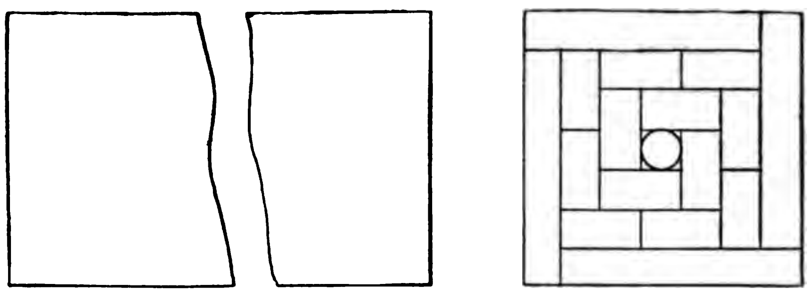

FIG. 2 METHOD OF PILING GRADE "B" BARS FOR HEATING AND ROLLING TO PRODUCE GRADE "A" IRON. PILES 2 TO 3 FT. LONG BY 7 IN. SQUARE

on a "doublet" taken at random during the making of each shot. If this does not equal the tabulated standard, others are tried on each side, and if they fail the whole shot is condemned. The proofing load is approximately 60 per cent of the breaking load. The tables of stresses are standard with the Navy Department, being based on both foreign and domestic practice.

Painting. Shots which have been proofed and found satisfactory are coated by hauling through a hot bath of asphalt paint, and are then stowed ready for shipment.

PROPERTIES OF STOCK MATERIAL

2 At present the iron used for the most important sizes of chain is made by the Burden Iron Company, of Troy, N. Y. In the manufacture of this iron, common gray-iron pig is puddled until the carbon,

phosphorus, and sulphur contents are correct. Puddle balls weighing two or three hundred pounds each are then extracted and rolled roughly into muck bars. These are cut into two- or three-foot lengths, piled five or six together, heated to welding temperature, and rolled again to give grade "B" iron. Grade "A" iron is further

FIG. 3 SECTIONS OF BURDEN IRON ETCHED 1 MINUTE

refined. It is made by cutting up grade "B" bars into short lengths of two or three feet, making up into piles 7 in. sq., as shown in Fig. 2, heating to welding heat, and passing successively through straight, hexagonal and round rolls down to the required diameters. The outside is formed by slabs 6 in. by 1 in., with a 1-in. round bar at the center.

SPECIFICATIONS

3 The Government specifications for wrought iron for chain making are in substance as follows:

4 Grade "A" must be of best quality American refined iron, puddled from all-ore pig iron and free from admixture of steel or scrap. The specified limits of chemical analysis are:

Phosphorus, not to exceed 0.10 per cent
Sulphur, not to exceed 0.015 per cent

FIG. 4[1] HEAVY PEARLITE STRUCTURE — FROM HEAVILY SHADED AREAS IN FIG. 3

The physical requirements are:

Tensile strength	48,000 lb. per sq. in.
Yield point, not less than ½ tensile strength	
Elongation	26 per cent in 8 in.
Contraction of area	40 per cent

CHEMICAL ANALYSIS

5 Chemical analysis of Burden iron, made by the chemical laboratory of the Boston Navy Yard from borings taken from several stock bars, gave the following average results:

[1] All microphotographs in this paper have a standard magnification of 100 diameters, with the exception of Figs. 7 and 24, which have a magnification of 50 diameters.

Carbon	0.10 per cent
Silicon	0.10 per cent
Phosphorus	0.085 per cent
Sulphur	0.008 per cent

6 The average phosphorus and sulphur contents are well within the specifications and did not exceed them in any particular case. The analysis indicates a very good grade of commercial wrought iron.

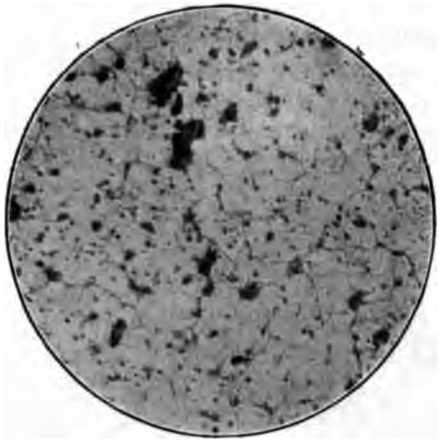

FIG. 5 FERRITE STRUCTURE WITH TRACES OF PEARLITE — FROM LIGHTLY SHADED AREAS IN FIG. 3

TENSILE TESTS

7 Twelve tensile specimens cut from a 3¼-in. Burden stock bar were tested, and in the following table the results are summarized and compared with the specification requirements. These results, like those of the chemical analysis, indicate a good commercial iron.

	Test results	Specification
Yield point, lb. per sq. in.	26,100	24,000
Tensile strength, lb. per sq. in	49,000	48,000
Elongation, per cent	35.5	26
Contraction of area, per cent	50	40

IMPACT TESTS

8 Impact tests of eight Charpy specimens cut from a 3¼-in. stock bar averaged as follows:

Longitudinal, 150 ft-lb. per sq. in.
Transverse, 39 ft-lb. per sq. in.

9 The difference of resistance to shock due to the direction of the slag fibers is here clearly shown. The definitions of "longitudinal" and "transverse" specimens and their relation to the direction of slag fibers are discussed in Par. 26.

METALLOGRAPHIC EXAMINATION

10 The appearance of transverse and longitudinal sections of a stock bar, when polished and etched, is shown in Fig. 3. The etching of the transverse section brings out clearly the outlines of the different grade "B" bars and slabs which were piled together to make up the grade "A" bar in the manner already illustrated in Fig. 2. Also each

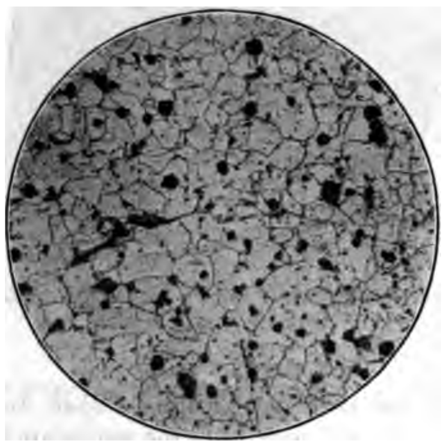

FIG. 6 PURE IRON STRUCTURE — FROM LIGHTEST AREAS IN FIG. 3

grade "B" bar is seen to be made up of well-defined areas of different shades or depths of color, representing the muckbars from which it was rolled.

11 Under the microscope the heavily shaded areas have a pearlite and ferrite structure, with numerous slag spots (Fig. 4), showing them to be wrought iron with about 0.20 per cent carbon. The more lightly shaded areas indicate less pearlite (Fig. 5), typical of about 0.05 per cent carbon, and the unshaded areas a pure wrought-iron structure (Fig. 6). Metallographic examination was made of specimens cut from three stock bars all of which displayed the same characteristics of segregated carbon; the line between adjacent areas of high carbon content and of low carbon is frequently very sharp, as in Fig. 7.

HEAT TREATMENT OF BURDEN CHAIN IRON

12 Preliminary to tests upon specimens cut from the links of chain cables, a laboratory investigation was made of the effects of certain variables in the process of heat treatment, upon the physical properties of Burden iron stock.

HEATING AND COOLING CURVES

13 The "thermal analysis" of the iron, or the determination of its "critical" points, is quite necessary for an intelligent study of its heat treatment. The first heating-cooling curves made with chain

FIG. 7 ADJACENT AREAS OF HIGH AND LOW CARBON (50 DIAMETERS)

iron were the simple temperature vs. time curves, or more strictly millivolts vs. time. These curves gave good indications of the A_1 (recalescence point) and A_2 (lower temperature point of thermal retardation in cooling) on heating and cooling, but no conclusive indications of the A_3 point (upper temperature point of thermal retardation in cooling).[1]

[1] For a detailed description of different methods and apparatus, reference should be made to: "Methods of Obtaining Cooling Curves," by George K. Burgess, Associate Physicist, Bureau of Standards, published in Reprint No. 99, from Bulletin of the Bureau of Standards, Vol. 5, No. 2, Aug. 3, 1908; Scientific Paper No. 213, "Critical Ranges A_2 and A_3 of Pure Iron," by G. K. Burgess, and J. J. Crowe (from Bulletin of Bureau of Standards, Vol. 10, Sept. 22, 1913); a pamphlet on "Determination of Critical Points" from the Metallographic Laboratory of the Scientific Materials Company, Pittsburgh, Pa.; the standard textbook of Burgess and Le Chatelier, and Bureau of Standards Bulletin No. 7 on Pyrometry.

14 The next step was to arrange for differential curve data, using a neutral body to give the furnace temperature. The test piece was ¾ in. in diameter and 1 in. long, with a through axial hole. The neutral body was a piece of nickel steel of the same size with a small hole in one end. These two bodies were placed on the floor of the muffle, side by side, tied together with asbestos cord. One of the two hot junctions of the differential couple was placed in the neutral body, the other was placed in the specimen, and the junction of the temperature couple was inserted in the test piece from the other end. The millivolt readings on the differential couple were taken on the precision potentiometer, while the temperature couple readings were taken on a portable potentiometer by another observer, readings being taken at equal time intervals of 30 sec. The $\theta - \theta'$ vs. time

TABLE 1 CRITICAL POINTS FROM HEATING AND COOLING CURVES FOR BURDEN IRON

Critical Point	Burden Stock		Average of Heating and Cooling, from Equilibrium Diagram		
	Heating, deg. cent.	Cooling, deg. cent.	Pure Iron, deg. cent.	0.1% C, deg. cent.	0.2% C, deg. cent.
A_3	915	875	900	850	800
A_2	768	768	768	768	768
A_1	738	700	690	690	690

curves plotted from these data showed good indications of an A_3 point, also several other variations not directly accounted for. Heating-cooling curves made later by J. J. Crowe were obtained with the use of a chronograph to keep the time in place of the ordinary stop watch used in the first determinations.

15 Typical heating and cooling curves of the Burden iron are shown in Fig. 8. Table 1 gives the critical points as determined by these curves; and also, for purposes of comparison, the average critical points for pure iron, 0.10 per cent carbon and 0.20 per cent carbon, as taken from Sauveur's Equilibrium Diagram of Iron-Carbon Alloys.

CRITICAL RANGE OF TEMPERATURE

16 In the heating curve, which is the important curve in connection with heat-treating, the indications of an upper critical point are indistinct and, at first glance, apparently indeterminate.

17 But the small point shown at 915 deg. cent. was duplicated in size, shape and position on all of the curves taken; as were also the indications of changes shown by irregularities of the curve from about 825 deg. cent. up to 915 deg. cent. It may therefore be concluded that the upper critical point is not sharp but extends over the range indicated, with a well-defined end at 915 deg. cent. This

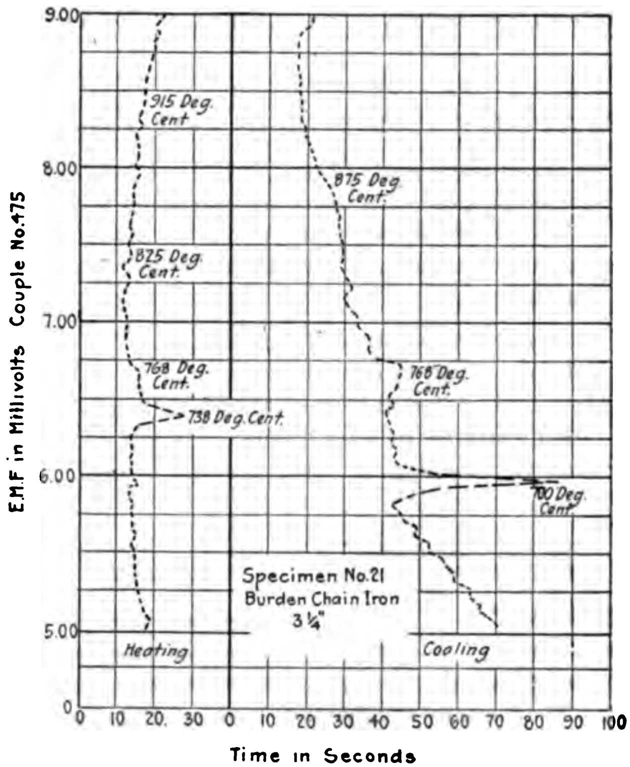

Fig. 8 Typical Heating and Cooling Curves for Burden Iron

phenomenon is explained by the unequal distribution of the carbon in the iron, different parts of which vary in composition from pure wrought iron to 0.2 per cent carbon, as was determined by metallographic examination. The second and lower critical points are well defined at the temperature indicated.

18 The fact that the actual upper and lower points are higher than those indicated in the iron-carbon equilibrium, is attributed to the presence of impurities, especially manganese.

19 In the cooling curve, the upper critical range has the same character as in the heating curve, but is about 40 deg. lower.

20 The practical conclusion from the location of the upper end of the upper critical range on the heating curve at 915 deg. cent. is that to anneal or air-quench the chain from above the critical range it should be heated to about 950 deg. cent., or about 56 mv. on the furnace thermocouples, instead of 52 mv., or 890 deg. cent. as has previously been the practice, in the manufacture of chain at the Navy Yard.

LABORATORY EXPERIMENTS IN HEAT TREATMENT

21 The preliminary study of heat treatment made in the laboratory was for the purpose of securing an indication of the best treatment to be used in the actual manufacture of the chain cable. In this investigation the attempt was made to find the effect of the following variables:

1 Maximum temperature of annealing
2 Rate of cooling, or quenching
3 Drawing to different temperatures after heating to maximum temperatures, or after different rates of cooling
4 Time of annealing, or time material is held at maximum temperature.

HEAT TREATMENTS TO WHICH IRON WAS SUBJECTED

22 The list of heat treatments developed is given in Table 2.[1] Each charge consisted of three tensile specimens and two blocks from which four longitudinal and four transverse Charpy specimens were machined. The furnace was heated to about 750 deg. cent. before specimens were introduced. The temperature was then raised, rapidly at first and slower as the upper limit was approached, to the point where the increase was about 5 deg. in 15 min.

PHYSICAL TESTS OF HEAT-TREATED SPECIMENS

23 After the specimens had received their heat treatments as outlined in Table 2, the following tensile and impact tests were made upon the heat-treated specimens:

[1] With the exception of heat treatments Nos. 1 to 6 inclusive, all heat treatments were made by J. J. Crowe, Physical Metallurgist.

1 Tensile tests giving
  a Yield point
  b Breaking stress
  c Per cent elongation in 2 in.
  d Per cent reduction in area from 0.20 sq. in.

TABLE 2 TABULATION OF HEAT TREATMENTS OF BURDEN IRON STOCK

Series (a): Rate of Cooling
1 Heat to 900 deg. cent. and cool in furnace
2 Heat to 900 deg. cent. and cool in air
3 Heat to 900 deg. cent. and quench in oil
4 Heat to 900 deg. cent. and quench in water

Series (b): Rate of Cooling
5 Heat to 1060 deg. cent. and cool in furnace
6 Heat to 1060 deg. cent. and cool in air
7 Heat to 1060 deg. cent. and quench in oil
8 Heat to 1060 deg. cent. and quench in water

Series (c): Rate of Cooling before Drawing
9 Heat to 1000 deg. cent., cool in furnace, reheat to 900 deg., quench in oil and draw to 650 deg.
10 Same as (9) but quenched in water

Series (d): Maximum Temperature before Cooling
11 Heat to 800 deg. cent. and quench in water
12 Heat to 850 deg. cent. and quench in water
13 Heat to 900 deg. cent. and quench in water
14 Heat to 950 deg. cent. and quench in water
15 Heat to 1000 deg. cent. and quench in water
16 Heat to 1050 deg. cent. and quench in water

Series (e): Temperature of Drawing
17 Heat to 1000 deg. cent., quench in water, draw to 550 deg.
18 Heat to 1000 deg. cent., quench in water, draw to 650 deg.
19 Heat to 1000 deg. cent., quench in water, draw to 750 deg.

Series (f): Rate of Cooling
20 Heat to 1000 deg. cent., quench in oil
21 Heat to 1000 deg. cent., cool in air
22, 23, 24, 25 } Series (*a*) Repeated
26, 27 } 5 and 6 of Series (*b*) Repeated

Series (g): Time of Annealing
28 Heat to 970 deg., hold 1 min., cool in air
29 Heat to 970 deg., hold 15 min., cool in air
30 Heat to 970 deg., hold 30 min., cool in air
31 Heat to 970 deg., hold 120 min., cool in air

2 Impact tests giving
  Resistance to shock in ft-lb. per sq. in. of an area of about 0.0785 sq. in. in a bar 10 mm. square by 55 mm. long, with 40 mm. between supports.

24 The tensile tests were made at the Boston Navy Yard and the impact tests were made on the Charpy impact machine at the U. S. Arsenal, Watertown, Mass. The results of these tests are summarized in Table 3.

25 It was considered important to make impact tests for the reason that chain cable which fails is usually broken by the severe shocks to which it may be subjected.

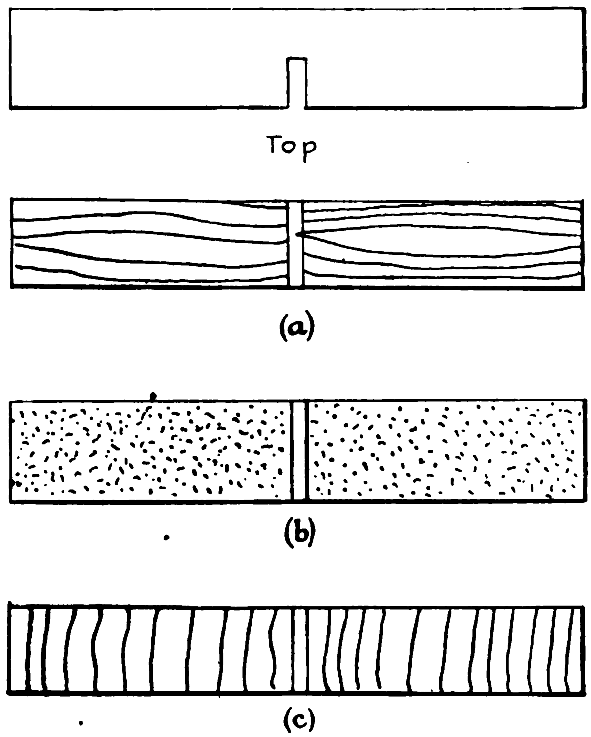

Fig. 9 Variations in Impact Specimens

26 In making Charpy tests of wrought iron, it is important to note the great difference between the resistance of the iron to shock, depending upon the relative direction of the nick in the specimen and the "grain" or slag streaks, in the iron. In Fig. 9, *a* represents the nick across the grain, this being called a "longitudinal" specimen, since the length of the specimen is parallel to the slag. Sketches *b* and *c* represent two variations of the nick parallel with the grain, depending upon which face the specimen is nicked. Both may be called "transverse specimens," since the length is perpendicular to the slag.

TABLE 3 RESULTS OF TESTS ON HEAT-TREATED SPECIMENS OF TABLE 2

No. of Heat Treatment	Maximum Temperature	Cooling Media		Tensile Results: Yield Point	Tensile Results: Breaking Stress	Tensile Results: % Elong. in 2 in.	Tensile Results: % Reduction of Area from 0.2 sq. in.	Impact Results: Charpy Impact Results, Long.	Impact Results: Charpy Impact Results, Transverse
Specifications:				24,000 (½ Breaking Strength)	48,000	26 (in 8 in.)	40		
Untreated Material:				26,600	49,000	36	50	150	39
Series (a): Rate of Cooling									
22	900	Furnace		23,465	46,965	38.2	56.4	93	46.9
23	900	Air		32,365	51,330	34.6	45.2	223	40.1
24	900	Oil		41,400	57,050	31.4	48.4	338	52.5
25	900	Water		49,275	68,720	24.7	41.7	406	43.8
Series (b): Rate of Cooling									
26	1060	Furnace		26,700	47,280	36.2	50.0	88	41.6
27	1060	Air		31,330	49,700	36.0	49.0	254	45.3
7	1060	Oil		39,435	56,900	30.0	53.0	341	34.5
8	1060	Water		44,250	65,400	26.0	51.0	401	37.1
Series (c): Hardening and Drawing			Draw. Temp. °C.						
24	900	Oil	..	41,400	57,050	31.4	48.4	338	52.5
9	900	Oil	650	32,900	50,325	39.0	58.0	268	59.9
25	900	Water	..	49,275	68,720	24.7	41.7	406	43.8
10	900	Water	650	31,525	49,450	38.0	59.0	394	46.8
Series (d): Maximum Temperature Before Cooling									
Water Quenched									
11	800	Water		49,165	68,350	19.1	41.4	240	39.3
12	850	Water		42,700	64,130	24.8	47.6	271	45.9
13	900	Water		43,215	62,880	28.3	49.4	371	33.8
25	900	Water		49,275	68,720	24.7	41.7	406	43.8
14	950	Water		50,615	69,880	22.9	51.7	335	33.7
15	1000	Water		48,835	67,135	25.9	54.3	362	37.5
16	1050	Water		45,780	63,130	25.9	52.3	389	37.5
8	1060	Water		44,250	65,400	26.0	51.0	401	37.1
Oil Quenched									
24	900	Oil		41,400	57,050	31.4	48.4	328	52.5
20	1000	Oil		41,750	55,785	34.6	54.5	383	49.3
7	1060	Oil		39,435	56,900	30.0	53.0	341	34.5
Air Cooled									
23	900	Air		32,365	51,330	34.6	45.2	223	40.1
21	1000	Air		33,365	49,515	36.5	55.5	262	43.1
27	1060	Air		31,330	49,700	36 0	49.0	254	45.3
Furnace Cooled									
22	900	Furnace		23,465	46,965	38.2	56.4	93	46.9
5	1060	Furnace		27,780	49,085	34.0	50.0	102	42.6
26	1060	Furnace		26,700	47,280	36.2	50.0	88	41.6
Series (e): Temperature of Drawing			Draw Temp.						
15	1000	Water	..	48,835	67,135	25.9	54.3	362	37.5
17	1000	Water	550	42,285	58,225	34.2	58.1	368	51.0
18	1000	Water	650	35,585	53,985	35.6	58.4	378	63.3
19	1000	Water	750	34,365	49,985	38.4	57.4	306	49.5
Series (g): Time of Annealing			Time at Max. Temp. Minutes						
28	970	Air	1	31,365	49,550	36.1	52.1	261	63.4
29	970	Air	15	32,300	49,200	37.4	55.2	282	63.2
30	970	Air	30	30,830	48,300	37.4	53.7	274	56.2
31	770	Air	120	29,800	48,480	37.8	53.1	320	57.7

27 Testing longitudinal Charpy specimens has the most significance since the relative direction of the shock and the slag is the same as in the case in the ends of links which are snapped taut in a chain. In transverse type *b*, the relative direction of shock and slag is the same as in the side of the link. Transverse type *c* has no significance in this investigation, and no specimens of that description were tested.

28 The results of the tensile tests upon specimens (1) to (6), inclusive, were not satisfactory, owing possibly to faulty machining of the specimens and possibly to an error in the technique of heating and quenching. These heat treatments were therefore repeated as indicated under Nos. 22 to 27, inclusive, in Table 3, and the specimens afterwards tested.

DISCUSSION OF RESULTS OF PHYSICAL TESTS

29 All tensile results were within 10 per cent of the average results from the three tensile specimens previously mentioned as having been subjected to heat treatments in each series. The averages are used in all plots.

30 The Charpy impact results varied greatly; in certain instances as much as 80 per cent from the average. For example, one specimen showed a fine, silky fibrous fracture with 321 ft-lb. per sq. in. resistance, and the next specimen, supposedly from the same bar, a coarse crystalline and dirty fracture with only 39 ft-lb. per sq. in. resistance (heat treatment No. 11).

31 These variations are probably due to variations in the character of the metal in different sections of the bar, as shown by the metallographic examination. It would be possible even for adjacent specimens to have widely different characteristics, and the probability of this is enhanced by the fact that the breaking sectional area of a Charpy specimen is only about 0.0785 sq. in.

32 Plots showing the effects of different rates of cooling, series (*a*), heated to 900 deg. cent. and series (*b*), heated to 1060 deg. cent. are given in Fig. 10.

The results of tests on the stock material before any heat treatment was given are shown by points on the left. It may be readily seen: (1) that furnace cooling reduces the Charpy and tensile strength and increases the elongation and reduction of area; (2) that air cooling gives greater strength than the original material with less elongation and reduction of area; (3) that oil and water quenching increase the strength considerably. In the case of heat treatment No. 25 in which the specimens were heated to 900 deg. cent. and quenched in water, the tensile breaking stress was increased from the 49,000 average to 68,700, which is an increase of 40 per cent, and the longitudinal impact resistance was increased from 150 average to 406, which is an increase of 165 per cent.

33 The effect of the rate of cooling on series (*b*), in which the specimens were heated to 1060 deg. cent., was generally less than in series (*c*), in which the heating was carried only to 900 deg. cent.

34 The effect of tempering, or hardening and drawing, is shown graphically in Fig. 11. On the left is shown series (*c*), the effect of drawing to 650 deg. cent. of specimens heated to 900 deg. cent. and quenched in oil and water, respectively. It is seen that the drawing brought the tensile breaking stress and elongation nearly down to the average untreated results, as shown in the left center of the sheet, while the per cent reduction of area was greatly increased over the average stock results, and the reduction in the Charpy longitudinal strength was very small.

35 On the right is shown series (*e*), all heated to 1000 deg. and water-quenched, the abscissæ being the drawing temperatures, with the results of the undrawn given at the extreme right for comparison. The general effect of increasing the drawing temperature is to reduce the strength and increase the elongation. There is little change in the impact strength and reduction of area, which remain considerably greater than in the untreated-stock results.

36 The effect of the maximum temperature of heating before water quenching, series (*d*), is plotted in Fig. 12. This comprises heat treatments Nos. 11 to 16, inclusive, in which the material was raised to different temperatures and quenched in water; and in addition heat treatments Nos. 8 and 25. Heat treatments Nos. 13 and 25 are the same and the difference in the results may be due either to the nature of the material, or to a slight variation in the heat treatment, as apparently there is considerable change near 900 deg. cent. It appears from these results that a temperature of about 950 deg. cent. would give the best results with water quenching, and any increase beyond 1000 deg. cent. gives less strength, although the Charpy longitudinal specimens gave best results at 900 deg. cent. and 1060 deg. cent. The data of the oil-quenched and air- and furnace-cooled specimens were not sufficiently complete to warrant plotting.

37 It was suspected that the length of time during which the specimens remained at maximum temperature in the furnace might have some effect on the results, so a special series (*g*) was added of

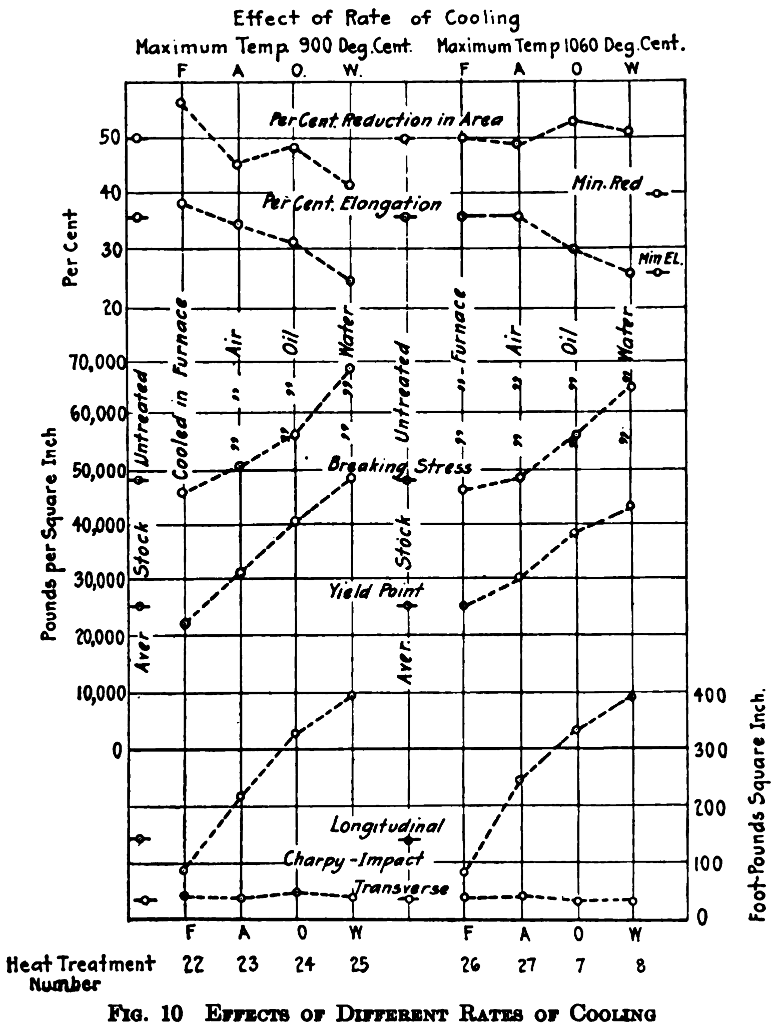

Fig. 10 Effects of Different Rates of Cooling

Specimens heated to 900 deg. cent. and 1060 deg. cent. cooled in furnace, cooled in air, quenched in oil, and quenched in water. Results on untreated material shown vertically at the left and in the center. The diagrams in Figs. 10, 11 and 12 show tensile and impact results.

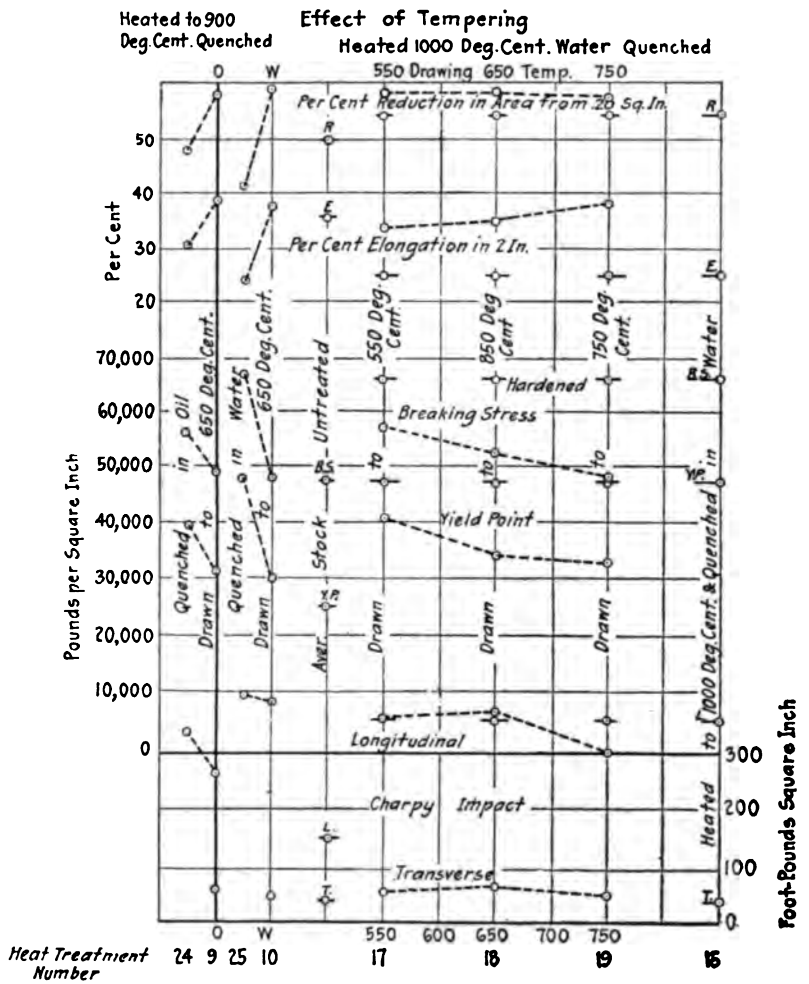

Fig. 11 Effects of Hardening and Drawing

Specimens heated to 900 deg. cent. and quenched in oil and quenched in water; and specimens heated to 1000 deg. cent. and quenched in water. Results on untreated stock material shown vertically at left center; results of quenching without drawing at extreme right.

heat treatments Nos. 28 to 31, which were all heated to 970 deg. cent. and maintained at that temperature for intervals of 1 min., 15 min.,

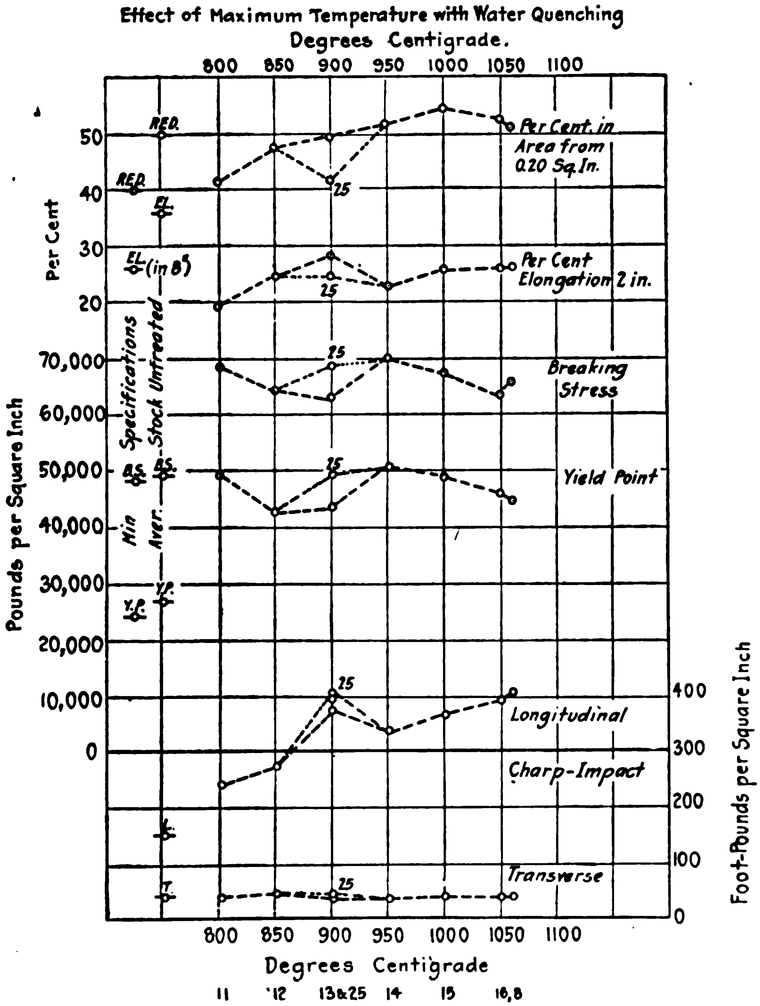

FIG. 12 EFFECT OF MAXIMUM TEMPERATURES OF HEATING WITH WATER QUENCHING

Specimens heated to from 800 to 1060 deg. cent. Results of untreated material at left.

30 min. and 2 hours. The results indicated that on specimens of that size the duration of the time of annealing has no appreciable effect.

38 As a general conclusion from the results of the physical tests of heat-treated Burden chain iron, it may be stated that it is affected by heat treatment in a manner very similar to low-carbon steel, giving an increase of 40 per cent in the tensile strength and 125 per cent in the longitudinal impact strength when heated to 950 deg. cent. and quenched in water. This was confirmed by microphotographs showing sorbitic and martensitic structure typical of the hardening structure of mild steel. It is therefore recommended that a further investigation be made of the effect of this heat treatment of chain cable, and the practicability of its application to the manufacture of chain cable.

PRELIMINARY EXPERIMENTS WITH CHAIN LINKS

39 Two preliminary experiments were performed with full-size chain, (1) to confirm the results of the laboratory experiments as to the best temperature to which the chain should be heated for annealing by air cooling, and (2) to determine the best sequence of annealing and proofing.

TABLE 4 EFFECT OF ANNEALING FROM DIFFERENT TEMPERATURES

Furnace Thermocouple, Mv.	Breaking Load, Lb.	Average Breaking Load, Lb.
48	410,000 462,000	436,000
52	474,000 448,000	461,000
56	445,000 326,000[1] 460,000 447,000	451,000
60	460,000 446,000	453,000

[1] Failure in weld.

TEMPERATURE OF ANNEALING

40 It was concluded from both the investigation of the stock material and the laboratory experiments on this subject that the chain should be annealed from about 950 deg. cent. (56 mv.) instead of 890 deg. cent. (52 mv.) as hitherto had been the practice. To confirm this conclusion, $2\frac{5}{8}$-in. Burden iron doublets were heated to temperatures corresponding to 48, 52, 56 and 60 mv. These were then annealed and pulled according to the results in Table 4.

41 There is no appreciable difference in the average breaking loads of the specimens heated to 52, 56 and 60 mv., and the variation of the few determinations is sufficient to render the numerical results of no particular significance. The fractures of the 56- and 60-mv. links, however, were, as a whole, very clean, fine and silky, and much better looking than those of the other links, and the practice has therefore been followed in the manufacture of the chain, of annealing from approximately 950 deg. cent. (56 mv.), with the result that stronger chain has been produced, as determined by the routine test "doublets" referred to in the first paragraph under *Proofing*.

TABLE 5 EFFECT OF SEQUENCE OF ANNEALING AND PROOFING

Sequence	Breaking Load, Lb.	Character of Fracture	Average Breaking Load, Lb.
1 Annealed, Proofed, Pulled.	445,000	Bent quarter all fibrous	451,000
	336,000	Failure in weld	
	460,000	Welded quarter all fibrous	
	447,000	Welded quarter 40% crystalline	
2 Proofed, Annealed, Pulled.	397,000	[1]	480,000
	431,000	[1]	
	401,000	[1]	
	441,000	[1]	
	476,000	Bent quarter all fibrous	
	484,000		
3 Annealed, Proofed, Annealed, Pulled.	460,000	Bent quarter all fibrous	451,000
	441,000		
	441,000		
	445,000	Welded quarter all fibrous	
	470,000		
	391,000	[1]	

[1] Failure in weld.

SEQUENCE OF ANNEALING AND PROOFING

42 The second series of preliminary tests was to determine the best order to follow in annealing and proofing. Chain may be (1) annealed before proofing, which is the regular practice, (2) annealed after proofing, or (3) annealed before and after.

43 When chain is proofed, it is strained well over the elastic limit of the link as a whole. For example, $3\frac{1}{4}$-in. test links are proofed to 367,000 lb. which leaves a permanent set of about $\frac{3}{8}$ in. The ratio of proof to pulling load is 367,000 : 620,000 = 0.59, while the ratio of

elastic limit to ultimate stress of the material is 27,000 : 49,000 = 0.55.

44 Proofing therefore strains certain parts of the link above the elastic limit and leaves it in an internally strained condition. If method (1) is used the chain leaves the shop for service with internal strains, but if method (2) is used the internal strains of proofing are relieved by annealing and from this consideration alone each link should be stronger.

45 For the purpose of testing, $2\frac{5}{8}$-in. Burden iron doublets were annealed from 56 mv. as summarized in Table 5.

46 As shown by the appreciably greater breaking load of the two test links which did not break by a failure of the weld, method (2) of annealing after proofing appears to produce the stronger link. On the other hand, it must be noted that four of the six test doublets of method (2) broke by failure of the weld before they were strained to their breaking stress. One of the best men in the shop forged all of the links for this experiment, knowing that they were for experimental purposes. This, combined with the fact that there was only one failure of the weld in the tests made for each of the other methods, indicates that the failure of these four welds was due to the sequence of operations.

47 The logical explanation is that when a link is proofed before it is annealed there is a tendency to start an opening in the weld due to local strains which may have been produced by forging. Then when the link is subjected to an applied load it will break through the weld before the breaking stress of the material is reached. If the link is annealed before proofing, this condition will be relieved.

48 After four of the six doublets for method (2) broke through the weld, it was decided to run another series of tests on this method. Eight $2\frac{3}{4}$-in. Burden iron doublets were made by the man referred to above, who took particular pains to make the weld good, after his former apparent failure. Of the eight, two broke low with a dull, lusterless fracture, indicating overheating. The average of the other six was 504,000 lb., failing generally across the welded quarter with a fibrous fracture. In spite of the careful welding, the point of the scarf in one link was found not welded when pulled to destruction.

49 These results were compared with results from tests on $2\frac{3}{4}$-in. doublets annealed and pulled in the course of regular shop manufacture. These doublets were treated by method (1) by first annealing and then proofing.

50 The average pulling strength of six of these links was 482,000 lb., which, compared with the 504,000 of the links treated by method (2), shows that the latter process produces the stronger links, provided there is no injury to the weld due to the action of proofing on local strains.

51 Because this tendency to injure the weld does exist where method (2) is used, the conclusion was reached that method (1) of annealing before proofing should be followed and not method (2). Theoretically method (3) would seem to be much preferable to both (1) and (2), since it combined the advantages of each, but the average breaking load was the same as for method (1). The fractures, however, were remarkably clean, pure, and silky. There would probably be a small advantage in using method (3), but it is not

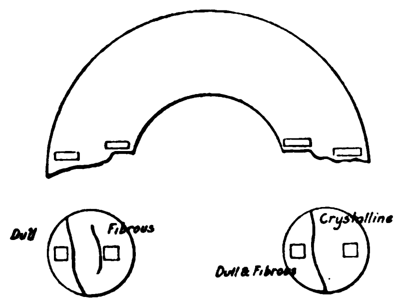

Fig. 13 Character of Fractures in Burden Iron Link

practical to go to the great additional trouble and expense of two annealing processes for the slight advantage to be gained.

PRELIMINARY METALLURGICAL EXAMINATIONS

52 It had been observed that pulled chain links showed three different varieties of fractures, dull, fibrous, and crystalline or granular; and that often two or three varieties occurred in the same fracture. It was therefore desired to determine whether any definite relation could be established between the structures of the fractures as they appear under the microscope and to the eye, whereby faults in heat treatment of the links could be detected merely by observing the character of the fracture of test links when pulled.

53 For this investigation a 3¼-in. Burden iron link was selected which had been pulled and gave fractures varying in character,

exactly suitable for the purpose. It gave a very low pulling test, and did not break through the weld, but across both welded quarters, both fractures being irregular and jagged. The link evidently had been burned.

54 As shown in Fig. 13, each fracture consisted of two well-defined areas differing widely in character as follows: On one side, a dull area and a fibrous area; and on the other side crystalline or granular, and dull and fibrous. From each of these a specimen was cut as near to the fracture as possible, for the purpose of microscopic examination.

55 The dull specimen, when polished, showed to the eye the weld across one corner, but otherwise a smooth surface. Under the microscope the grain size of the structure varied considerably, but there were no signs of burning. Fig. 14 shows a typical section.

56 The specimen from the fibrous fracture showed to the eye a smooth surface without cracks or slag streaks. Under the microscope the structure varied in grain size and carbon content, and in parts showed signs of burning. Fig. 15 is from this specimen.

57 The specimen from the crystalline fracture showed pronounced slag streaks and air cavities dividing the specimen into large grain-like areas. As in the fibrous specimen the structure varied greatly in different parts, but was generally uniform over the surface of any one "grain." A typical structure is shown in Fig. 16.

58 From a study of the microphotographs of which Figs. 14 to 16 inclusive are representative, the conclusion was reached that there is no evident relation between the microscopic character of fracture and the microscope structure. A more detailed investigation, however, might bring out some such relations.

TESTS OF FULL-SIZE LINKS BEFORE AND AFTER ANNEALING

DISCUSSION ON WELDING AND ANNEALING

59 It was mentioned by Mr. Coburn in the introduction to this paper that the "stiffness" of the first power-forged links was so great as to cause fractures and that this stiffness was removed by annealing. "Stiffness" is the customary expression for lack of ductility as regards the stretching or elongation of chain links.

60 In order to explain the failure of these power-forged, unannealed links, and the way in which annealing restored their strength, it is important to understand the essential differences between hand and power forging.

61 The use of the steam hammer in forging enables a much greater reduction in area to be made at the weld. and hence produces a much greater amount of internal work. It also permits the use of

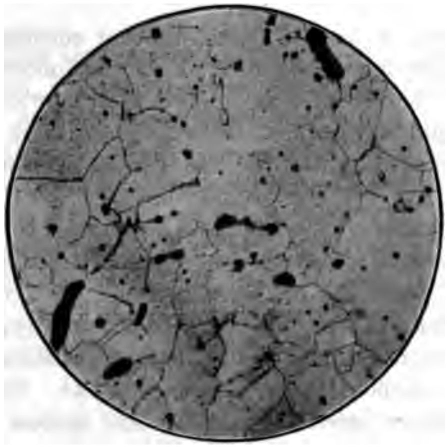

FIG. 14 FERRITE STRUCTURE, FROM DULL SECTION, FIG. 13

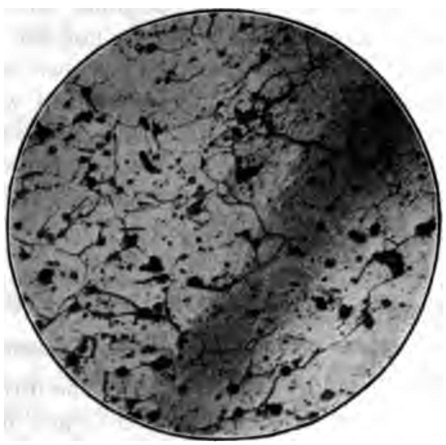

FIG. 15 ALL FERRITE, LARGE GRAIN, BURNING, FROM FIBROUS SECTION, FIG. 13

a longer scarf, which necessitates a welding heat further around on the quarter of the link. In power forging, therefore, the link is brought up to a white heat well around on the quarter, and then shades off through light orange, dark orange, and bright red, to a dull red at the bent end. In hand forging, the white heat does not cover so large an area and shades down to a dull red on the side of the link, and to black on the bent quarters and end.

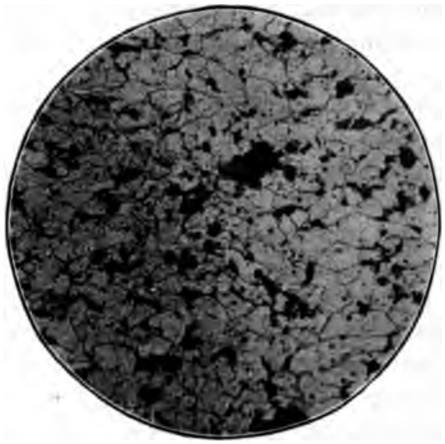

FIG. 16 SOME PEARLITE, SMALL GRAIN, NO BURNING, FROM CRYSTALLINE SECTION, FIG. 13

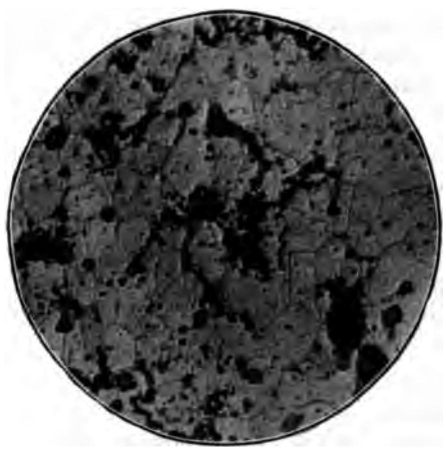

FIG. 17 CHARACTERISTIC SECTION HAVING MARKED BURNING

62 It would be expected that these conditions would permit a greater crystal growth of the power-forged chain, except under the hammer at the welded end. Since the welded end of the power-forged link receives a much greater amount of work from hammering, it

should have a much smaller grain size than the hand-welded link at the finishing temperature. On the other hand, the power-forged link has a greater cooling range during which the crystals may grow, since its finishing temperature is higher; but most probably this effect is small in comparison with that of hammering. The net results should be an appreciably smaller grain size at the weld of the power-forged link.

63 In comparison with the other parts of the same link, the grain size at the weld should increase from the middle to the end of the weld as the hammering effect decreases, and then should decrease from the weld around to the bend and in proportion to the maximum temperature to which the part was heated.

64 Hammering also deforms the crystals at the weld, but this effect should be obliterated by the crystal growth during cooling from 1200 deg. cent. down through the critical range, after hammering has ceased, and there should be no resultant "strain hardening."[1]

65 The effect of the carbon content should further be considered. Above the critical range, carbon is present in the solid solution of iron, or austenite. As the metal after forging cools slowly in air through the critical range, all the carbon should separate freely from the austenite crystals in the form of pearlite. Therefore, carbon should have little hardening effect in either hand- or power-forged chains; and, since there is no practical difference between the cooling conditions of the two, it should cause no difference between them in hardening.

66 The only remaining difference between the structures of the hand-forged and power-forged links which could cause the greater stiffness of the latter, is in the grain size. In the side of the link where most of the stretching takes place, the grain size is probably somewhat greater in the power-forged link, while in the welded quarter, where the first power-forged links broke, the difference is probably not appreciable. According to Rosenhain, coarse structure

[1] With respect to these points Rosenhain says, in his Introduction to the Study of Physical Metallurgy, p. 299: "If the working operation is stopped at any instant and the metal is allowed to cool down from such a high temperature, there will be no direct signs of the application of work, *i.e.*, the metal will be completely annealed and will consist of an aggregate of equi-axed crystals. These crystals will be very small, compared with those of the original ingot, for example, because the crystals have been deeply disturbed, and those finally present have only been allowed a very short time for their formation and growth, but there will be no signs of distortion and no mechanical hardening effect which could be removed by subsequent annealing."

generally does not affect tensile strength or ductility to a marked degree, although it does greatly decrease resistance to shock.

67 Therefore, from a theoretical consideration of the material as an alloy of iron and carbon only, we can find no apparent cause for stiffness in the power-forging process. In the tests which are hereafter described, however, it developed that in the power-forged link there was an overheated, distorted structure, which was relieved by the recrystallization in annealing. The stiffness of the link was undoubtedly removed primarily by relieving this condition, but also to some extent by the lessened hardening effect of carbon and the refinement of grain size caused by annealing.

68 The welding of the link is of prime importance, and scientific methods of determining and regulating the welding heat, such as by the use of optical pyrometer, should undoubtedly be developed. At the time this investigation was made, however, dependence was still placed upon the experience of the mechanics who do the welding, and since only about one in twenty-five of the links tested in the course of manufacture break through the weld, it was considered that the question of the strength of the weld could be eliminated in most of the investigations described in this paper.

69 In forging the links it is evident that underheating will produce a very weak pseudo weld, but an experienced chain maker seldom underheats. Overheating above 1400 deg. cent. will, of course, produce a very coarse structure which it is difficult to remove by annealing, and, therefore, should be avoided. If the link is heated to the neighborhood of 1450 deg. cent., it will be "burnt"; that is, it begins to run between the grains. This condition, which is not infrequent, cannot be removed by annealing and results in a weak and brittle structure, particularly when subjected to shock. Signs of slag running and collecting between grains is the characteristic indication of burning and can easily be detected under the microscope, as shown in Fig. 17 taken from the crystalline section in Fig. 13.

TEST LINKS

70 For the purpose of testing, a shot of five 3½-in. Burden iron links was forged in the regular manner and four of these were annealed. Specimens were cut from the unannealed link (designated as No. 1) and from one of the annealed links (designated as No. 2). The remaining three links were pulled. Ten sections were located around links Nos. 1 and 2, as shown in Fig. 18. Piece 7–8 was used for

tensile specimens and Charpy specimens were cut from the straight section 2–3 and from the curved section 0–1.

71 A slice $\frac{3}{4}$ in. thick was cut from each of sections 0 to 6, and from 9, for metallographic examination. Eight specimens $\frac{3}{4}$ by $\frac{3}{4}$ by $\frac{1}{2}$ in., were cut from each of these slices, numbered from 1 to 8, and their faces polished as required.

72 At the time the tests were started it was the practice to heat the chain to about 890 deg. cent. (52 mv.) and this value was used throughout. The present practice, as previously stated, is to heat to

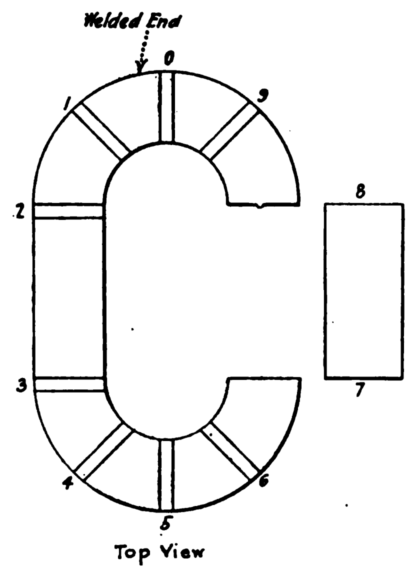

Fig. 18 Location of Test Specimens in Link

950 deg. cent. (56 mv.). The effect which this change might be expected to have on the results will be discussed later.

73 It should be noted that the tensile specimens, cut as they were from the side of the link, give a test of the material in that place only, and not of the material in other parts, nor of the strength of the link as a whole. It was impracticable to cut tensile specimens from the curved ends of the link, where the characteristics of the metal are modified by forging and where, also, the links usually fail.

74 Of the three links which were pulled, one broke across the welded quarter at a stress of 683,000 lb. The requirements for these links are: proof test, 376,000 lb.; and breaking load, 620,000 lb.

RESULTS OF PHYSICAL TESTS

75 The results of the physical tests of the specimens are given in Table 6. In the third column are entered for comparison the results of tests on untreated stock material previously given. However, only a rough comparison can be made between these results and those from link 1, for the reason that in the link specimens there is a curvature of the slag grain with respect to the axes of the specimens, due to the curvature of the link. This curvature probably renders of no significance any difference under 10 per cent. The only marked difference is in the longitudinal Charpy tests which gave results 45 per cent lower in the case of those cut from the side of the link, due

TABLE 6 TEST OF UNANNEALED AND ANNEALED LINKS

	Average Results		
	Link No. 1	Link No. 2	Stock Material
Yield point, lb. per sq. in.	27,900	26,600	26,100
Tensile strength, lb. per sq. in.	47,600	45,500	49,000
Elongation, per cent	37	38	34
Reduction of area, per cent	51	55	50
Charpy tests, ft-lb. per sq. in.			
Longitudinal, quarter	85	122	150
Longitudinal, side	90	176	
Transverse, quarter	48	41	39
Transverse, side	41	53	

to a combination of the effects of carbon and overheating in addition to curvature. The longitudinal specimens from the quarter gave somewhat lower results than those from the side, due to the effects of greater overheating, greater grain curvature, and larger grain size. The results of transverse specimens, as shown by the laboratory heat treatments, are generally contradictory and unreliable, and are therefore neglected.

76 In comparing the results of links No. 1 and No. 2, the question of curvature is of course eliminated, as the specimens from both links were in the same condition. Link No. 2 gives a small decrease in yield point and strength, and a small increase in ductility, which shows that annealing in air has lessened the hardening effects of carbon. Although annealing has also relieved the overheated condition and refined the grain size, these conditions do not have an appreciable effect in increasing the strength as shown by static tests.

77 In link No. 2 the longitudinal Charpy tests, side and quarter, were increased respectively from 90 to 176 ft-lb. per sq. in. (practically 100 per cent), and from 85 to 122 (almost 45 per cent), showing that annealing has greatly increased the resistance of the metal to shock, due mostly to relieving the overheated condition and refining the grain size, which have a decided effect on ability to resist shock.

78 Comparing the results of link No. 2 to those of the Laboratory Experiment on Burden iron, which received the same heat treatment (No. 23), it is seen that the yield point, tensile strength, and Charpy results are smaller, and the ductility greater in the case of the link specimens. This indicates that a heat treatment applied to actual chain will not have as great an effect in changing the physical properties of the metal as it does when applied to laboratory specimens, on account of the much greater cross-section of the link, and of the mechanical treatment which it has received.

79 The foregoing tests lead to the conclusion that the present heat treatment of cooling in air slightly decreases the yield point and tensile strength, increases the ductility, and greatly increases the resistance to shock of the metal of the forged link.

GRAIN SIZE

80 It was desired further to study the effect of the heating and hammering processes, and of annealing upon the grain size of different sections of the links.

81 The average grain size for four of the metallographic sections of the link was estimated by the method of counting grain size which was described and recommended by Jeffries, Kline and Zimmer.[1]

82 Tracing-paper sheets 4 in. by 5 in. each were cut and on each was inscribed in ink a circle 79.8 mm. in diameter, corresponding to the standard magnification of 100 diameters. For each count one of these sheets was held down on a clear glass plate in the focusing slide of the camera. The place to be counted was brought into the field of the microscope and focused at 100 diameters on to the tracing paper as a screen. Then the grains were counted, whole grains being marked off by checks; and partial or border grains by crosses, as shown in Fig. 19.

83 To obtain the grain size, the checks and crosses were each counted; the sum of the crosses multiplied by 0.6 and added to the

[1] The Determination of the Grain Size in Metals — Bulletin A.I.M.E., Dec., 1915.

sum of the checks. The resultant total was multiplied by 2, which is the factor corresponding to 100 diameters, as given by the article referred to. This gives the grain size in grains per square millimeter. The average error of the method is estimated at only 2.1 per cent.

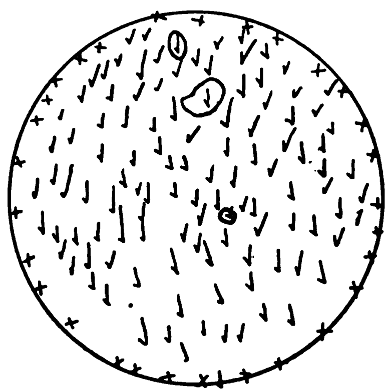

FIG. 19 METHOD OF RECORDING GRAINS OF SPECIMEN PROJECTED ON PAPER BY CAMERA FOR PURPOSE OF COUNTING

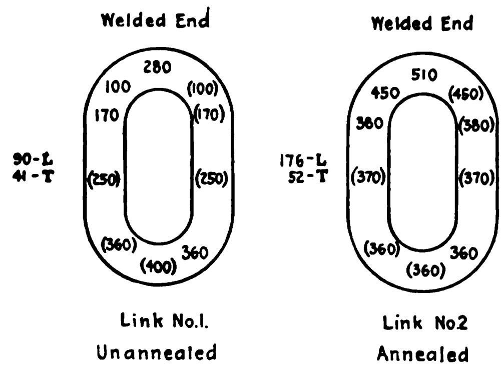

FIG. 20 GRAINS PER SQ. MM. AT VARIOUS SECTIONS OF LINKS 1 AND 2

84 To determine the average grain size for each section, the faces of the eight specimens were examined and a spot of average grain size on each was selected for counting. The average of the eight counts was taken as the grain size of the section. After link No. 1 had been completed in this manner, the great variation in the count of the different specimens of one section emphasized the fact that the method was not really scientific, although the averages so obtained were probably not far from right.

85 A more rational method was therefore adopted for link No. 2. It had been noticed that areas which contained an appreciable amount of pearlite generally had a smaller grain size, roughly in proportion to the amount of pearlite. The percentage of the face of each specimen containing enough pearlite to affect the grain size appreciably was estimated by eye from the different shades of etching of parts of the face (see Fig. 3). A spot of the average grain size of these "pearlite" areas was counted for each specimen as well as one for the average grain size of the "ferrite" areas. Then to find the average of the specimen, the pearlite and ferrite counts were each multiplied by its percentage of area, and the total of the two divided by 100 per cent. For example:

Type of Area	Link No. 2, Grain, Size	Section No. 0, Percentage of Total	Specimen No. 4, Multiple
Ferrite...........	354	70	24780
Pearlite..........	830	30	24900
			49680

49680/100 = 497

86 This method gave fairly uniform results for the different specimens of a section. The average of the eight specimens gave the grain size of the section as before. The results from links Nos. 1 and 2 were as follows:

Section	Link No. 1	Link No. 2
0	280 gr. per sq. mm.	510 gr. per sq. mm.
1	100 gr. per sq. mm.	453 gr. per sq. mm.
2	169 gr. per sq. mm.	388 gr. per sq. mm.
6	359 gr. per sq. mm.	365 gr. per sq. mm.

87 It is not contended that these results are exact. In fact, they are probably not good to within 10 per cent, but it is stated with confidence that they give a very good rough comparison of the grain size of the various sections.

88 In Fig. 20 are sketches of links Nos. 1 and 2, showing the grain size at the various sections. The determinations in the preceding table are indicated to the closest ten, without parentheses. The other figures, in parentheses, were obtained by induction as follows:

89 Since both sides of the link are in the same condition by similarity, section No. 9 should be the same as section No. 1, No. 8 the same as No. 2, and No. 4 the same as No. 6, and they are so indicated. The middle of the side of the link should have a grain size about half way between that of the welded and bent quarters, or about 250 gr. per sq. mm. for link No. 1, and 370 gr. per sq. mm. for link No. 2. From the decrease in grain size towards the bent end of link No. 1, its grain size at the bent end is probably about 400 gr. per sq. mm. Since there is practically no variation in grain size away from the welded end of link No. 2, the grain size at its bent end is probably about the same as at the quarters, or 360 gr. per sq. mm.

90 In link No. 1 the grain size is fairly small right at the middle of the weld, but increases rapidly towards the quarter, since the effect of hammering on crystal growth then decreases faster than the effect of heating. But from the welded quarter to the bent end the grain size decreases again; since there has been no hammering and the temperature resulting from heating the link decreases as the distance from the weld.

91 Comparing link No. 1 with link No. 2, it is seen that annealing has the effect of refining the grain size of the link, especially at the welded end. This gradation to a smaller grain size at the welded end shows that some effect from the smaller grains in that part before annealing still remains to produce a somewhat smaller grain after annealing in spite of recrystallization.

METALLOGRAPHIC EXAMINATIONS

92 A microscopic study was made of the structure of the metal and its variation in different sections, and for this purpose both transverse and longitudinal faces of specimens from all parts of each link were examined.

CHARACTER OF STRUCTURE OF UNANNEALED LINK (NO. 1)

93 In no section of link No. 1 is the carbon distributed homogeneously; it appears in segregated areas, except that near the boundaries of such areas the carbon has become somewhat diffused into the neighboring ferrite areas due to the various heats which the link had received during manufacture. The proportion of pearlite areas containing greater than 0.05 per cent carbon in different specimens varies from about 15 per cent to about 50 per cent, the average being about 30 per cent.

FIG. 21 LINK 1, WELDED END. FERRITE WITH TRACES OF PEARLITE, LARGE GRAIN SIZE, SLIGHT DISTORTION

94 Typical areas from the welded end containing varying percentages of carbon are shown by Figs. 21–24. The notable feature is the distortion of the ferrite grains into elongated bands with angular bands of pearlite between them. This distortion is proportionate to the amount of pearlite in the area. It does not, however, extend through all parts of the welded end. A large number of the

FIG. 22 LINK 1, WELDED END. FERRITE AND PEARLITE (0.1 PER CENT C). MEDIUM GRAIN SIZE, SLIGHT DISTORTION

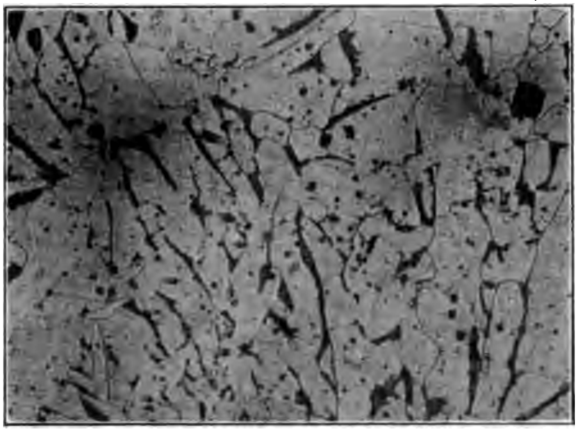

FIG. 23 LINK 1, WELDED QUARTER. FERRITE AND PEARLITE (0.15 PER CENT C). MEDIUM GRAIN SIZE, MARKED DISTORTION

specimens at this end show a normal or only slightly distorted structure, and in certain of them signs of slight burning were observed in the vicinity of the weld.

95 In the other sections of the link, away from the weld, there were no signs of distortion of the grains, and the pearlite had separated from the ferrite crystals in a perfectly normal manner, as shown by

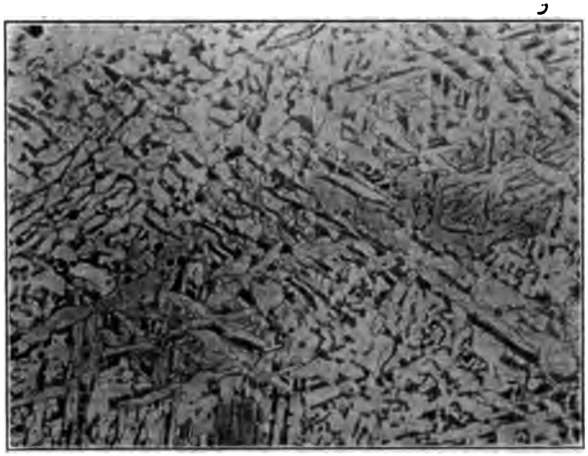

FIG. 24 LINK 1, WELDED QUARTER. FERRITE AND PEARLITE (0.15 PER CENT C). MEDIUM GRAIN SIZE, MARKED DISTORTION WITH ELONGATED FERRITE AND PEARLITE BANDS (50 DIAMETERS)

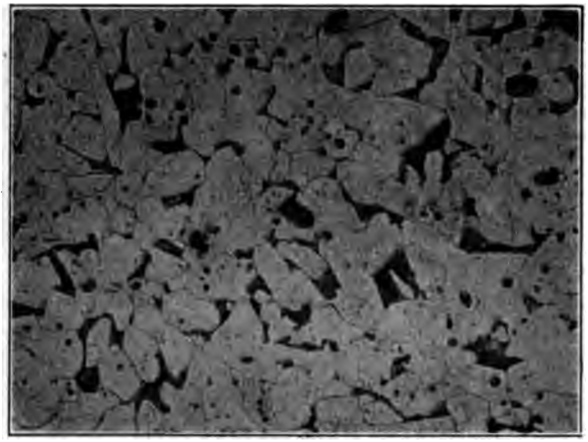

FIG. 25 LINK 1, BENT QUARTER. FERRITE AND PEARLITE (0.15 PER CENT C). MEDIUM GRAIN SIZE, NO DISTORTION

Fig. 25. The smaller average grain size in the bent end of the link was clearly brought out by this examination.

CHARACTER OF STRUCTURE OF ANNEALED LINK (NO. 2)

96 In the annealed link the same unhomogeneous distribution of carbon was found, as would be expected. Around the welded end, however, pearlite areas were relatively fewer than in link No. 1;

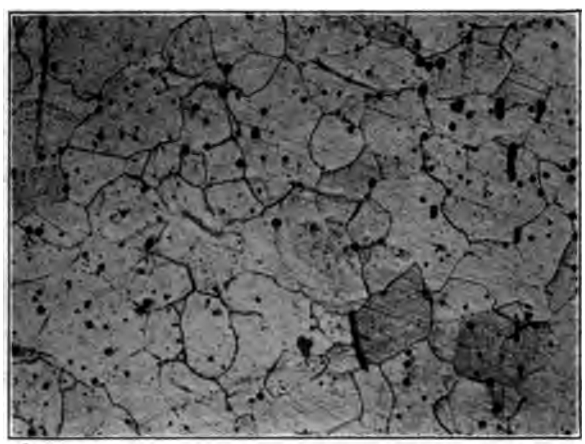

FIG. 26 LINK 2, WELDED END. FERRITE, MEDIUM GRAIN SIZE

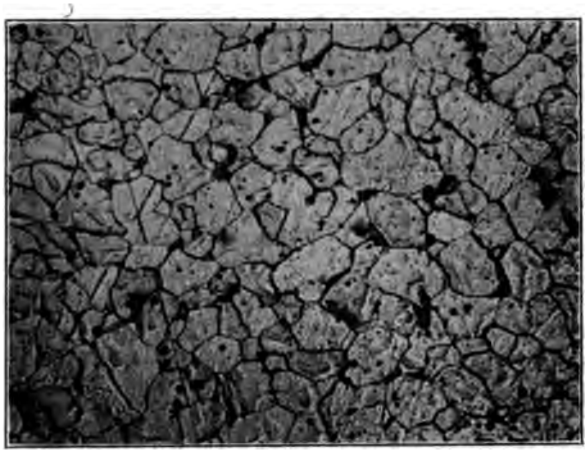

FIG. 27 LINK 2, WELDED END. FERRITE AND TRACES OF PEARLITE. SMALL GRAIN SIZE

other parts of the link were about the same as link No. 1. The possibility of this non-uniformity between the two links should have been avoided by insuring that both links were made out of the same bar, but the difference was not appreciable enough to affect any of the general conclusions which might be drawn from the experiment.

97 Typical areas from this link, with varying percentages of carbon, are shown by Figs. 26 to 30. The important point brought

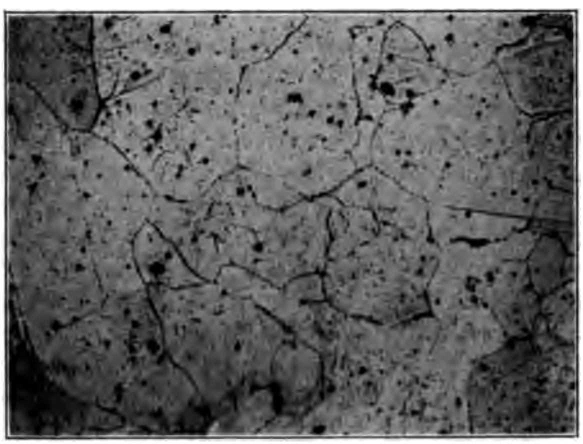

FIG. 28 LINK 2, WELDED QUARTER. FERRITE. VERY LARGE GRAIN SIZE

out is that there are no signs of distorted structure in any part of the link. The pearlite has separated and segregated around equi-axed grains in the normal manner. The average grain size is fairly uniform but is smaller in pearlite than in pure ferrite areas. Near the weld itself the structure was often very fine, as in Fig. 27, but away from the weld it would grow rapidly coarser, often to a grain size as large as that shown in Fig. 28.

EFFECT OF HEAT TREATMENT

98 From the foregoing examination, the conclusion is reached that the stiffness of the unannealed link is largely due to the overheated distorted structure in the welded end.

99 The fact that the metal of the forged link does not return to its normal condition during slow cooling after forging, even if it has been "overheated" to 1350 deg. cent., is probably to a large extent due to the effects of slag and other impurities. Annealing relieves this condition by the process of recrystallization, which practically wipes out all former structure and gives a finer and more normal grain size.

100 In regard to this overheating, Rosenhain says:[1] ". . . We find that by 'overheating' steel, *i.e.*, by exposing it to unduly high temperatures, or for too long a time at any temperature above Ac_3, the growth of a very coarse iron structure results, and this, on cooling down, gives rise to a corresponding coarse ferrite-pearlite structure. Not only this, but the arrangement and forms assumed by the pearlite which is formed from such steel is characteristic; there is a strong tendency for the ferrite to take the form of straight bands with elongated and angular patches of pearlite between them, the ferrite bands frequently crossing one another, at angles of 60 deg. Such a coarse, sharply angular structure is, of course, extremely undesirable; there is a minimum of interlocking between ferrite and pearlite, and the straightness of the arrangement facilitates the propagation of slip or cleavage through the crystals. Such structures are, in fact, frequently met with in steel objects which have failed in service. Under test they generally exhibit some degree of weakness as regards shock and alternating stresses, but their tensile strength and elongation are frequently quite satisfactory. The most typical feature, however, is a decided drop in the yield point as compared with that of the same material in a more normal condition."

[1] Introduction to the Study of Physical Metallurgy, pp. 281–282.

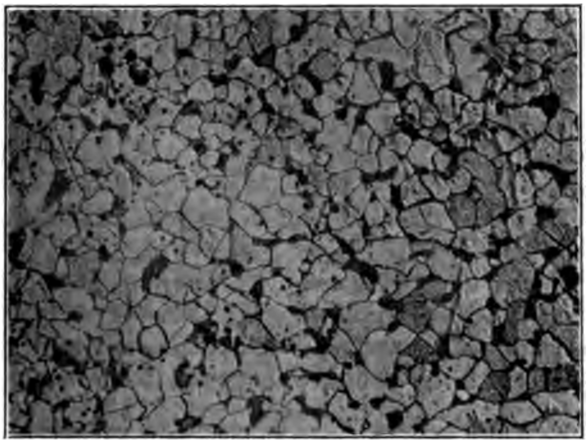

FIG. 29 LINK 2, WELDED QUARTER. FERRITE AND PEARLITE (0.1 PER CENT C). VERY SMALL GRAIN SIZE

101 The above description fits exactly the structure which has been found in the carbon areas of the welded end of the unannealed link. In the results of the physical tests, it was seen that the shock-resisting quality of this link was actually low, but it appears that the tensile strength and elongation are "quite satisfactory" as indicated in the foregoing quotation.

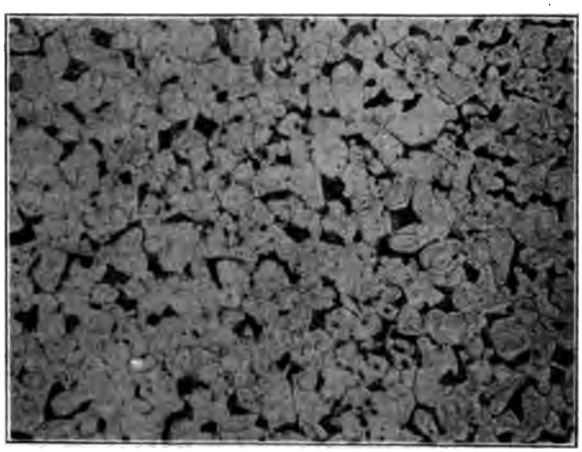

FIG. 30 LINK 2, BENT QUARTER. FERRITE AND PEARLITE (0.15 PER CENT C). VERY SMALL GRAIN SIZE

CONCLUSIONS

SUMMARY OF RESULTS

102 For heat-treating, Burden iron should be heated to about 950 deg. cent. (56 mv.) instead of 890 deg. cent. (52 mv.) which was the former practice.

103 There is no apparent relation between character of fracture and character of structure.

104 Heating to 950 deg. cent. (56 mv.) for cooling in air gives stronger and better chain than heating to lower temperatures, and heating to higher temperatures gives no improvement.

105 The present procedure of annealing before proofing should be continued.

106 Stiffness of the unannealed link compared with the annealed link is mainly caused by overheated distorted structure at the weld.

107 Annealing removes stiffness of the forged link by relieving overheated distorted structure. It decreases the tensile strength and yield point, and increases the ductility and resistance to shock, of the metal; but it increases the strength, as well as the ductility and resistance to shock, of the link as a whole.

108 The laboratory experiments indicate the following conclusions:

(1) *Rate of cooling.*

Cooling in furnace:

Reduces tensile strength from 49,000 to 47,000
Reduces impact resistance from 150 to 90
Increases elongation and reduction of area.

Cooling in air:

Increases tensile strength from 49,000 to 50,500
Increases impact resistance from 150 to 240
Reduces elongation and reduction of area.

Quenching in oil:

Increases tensile strength from 49,000 to 57,000
Increases impact resistance from 150 to 340
Reduces elongation from 36 to 31 per cent.

Quenching in water:

Increases strength from 49,000 to 67,000
Increases impact resistance from 150 to 400
Reduces elongation from 36 to 25 per cent.

(2) *Temperature from which cooled or quenched.* Increase of temperature above 950 deg. cent. slightly decreases tensile strength, increases elongation and reduction of area, and does not appreciably affect impact resistance. Temperature of 950 deg. cent. gives best results.

(3) *Drawing effect.* The higher the drawing temperature, the greater the decrease in tensile strength and increase in elongation. Drawing has little effect on impact resistance.

(4) *Increased time of annealing* slightly decreases tensile strength, and slightly increases elongation, reduction of area, and impact resistance.

DISCUSSION

Carle R. Hayward[1] (written). This paper is an important addition to the meager knowledge available of the effect of heat treatment on wrought iron. The careful and thorough manner in which the experiments were performed makes the conclusions all the more important.

It seems to the writer that one feature is deserving of more comment, viz., the distribution of the carbon, for upon this will depend to a large extent the properties of the furnished product. This is referred to in Par. 85 and emphasized by Figs. 21–28. It seems probable that if the total carbon found in the material had been uniformly distributed a different temperature might have been found desirable. Such uniform distribution cannot be expected in wrought iron, however. It is also evident that a high-carbon area would be greatly detrimental in the vicinity of the weld, and the possibility of one occurring there lends an element of uncertainty to the strength of a link.

Another feature deserving of comment is the effect of slag where the strain may be along the grain. This is referred to in Par. 27.

The possible detrimental effect of irregularly distributed carbon and the presence of slag suggest at once the use of a mild steel, where these two difficulties will be eliminated. The writer understands that tests with steel are contemplated by Mr. Coburn. The results will be awaited with interest.

[1] Asst. Professor of Mining Engineering and Metallurgy, Mass. Inst. of Technology, Boston, Mass.

Henry Goldmark[1] said that he wished to endorse the statement of Mr. Coburn with regard to the great advance which has been made in the manufacture of these chains by power rather than by hand. In connection with his work as designing engineer at Panama, he had found it necessary to procure some 10,500 ft. of 3-in. chain, about one-third being stud-link and the other part open-link, for stretching across the canal locks just above the lock gates. One of the greatest difficulties they had was to obtain chain of the proper strength, and toward the end of the time, fortunately under Mr. Coburn's direction at the Boston Navy Yard, they began to make power chains; and it was only fair to say that the power-made chains had uniformity of strength and presented a good appearance, and more than that they had the advantage of speed and cheapness of manufacture. The hand-made chains, particularly the open-link chains, ran quite a little lower in strength than the specifications called for, but there was a great uniformity in the power link chains; and while there were only six of the latter made out of the 24 that were required, they were accepted, everyone without question, and showed a strength very nearly up to the stud-link anchor cables.

F. G. Coburn, in closing, said that in the Panama Zone work, so far as strength was concerned, the stud-chain requirement was 525,000 lb., and that formerly they considered themselves fortunate if they got 450,000 lb. But when they began to use the new method of power forging, they found that they could easily attain the required 525,000 lb.; and in his museum he had one chain that had carried 576,000 lb., and which they had not strained further for fear that it would hurt the machine.

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No. 1572

CODE OF SAFETY STANDARDS FOR CRANES

Prepared by the Sub-Committee on the Protection of Industrial Workers of the American Society of Mechanical Engineers

The word "SHALL" where used is to be understood as mandatory and "SHOULD" as advisory.

The following Standards apply to cranes which are regularly used in and form part of a permanent industrial plant. In addition to Electric Traveling Cranes, these regulations are to cover Jib Cranes, Monorail Cranes, Hand Power Cranes, and other hoisting apparatus of a similar nature, in so far as the various sections apply.

The provisions of all Safety Standards issued by the Society shall apply in cases not specifically covered herein.

Caution: Employees shall not remove or make ineffective any safeguards except for the purpose of making repairs, and safeguards so removed shall be replaced when repairs are completed.

ELECTRIC TRAVELING CRANES

GENERAL CONSTRUCTION

1 Proper provisions for strength shall be made for all parts subject to impact and rough usage. Journals and shafts shall be of sufficient size to bring the pressure within safe limits.

2 All apparatus shall hereafter be designed throughout with not less than the following structural factors of safety, under static full rated load stresses, based on the ultimate strength of the material used:

a All gears, and complete hoisting mechanism, factor of not less than eight (8).

b All other parts, factor of not less than five (5).

3 Calculations for wind pressure on outside cranes shall be based on not less than thirty pounds per square foot of exposed surface.

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4 Cranes should be of what is known as "All Steel Construction;" no cast iron should be used except for such parts as drums, bearings, brackets, etc. Cast iron shall not be used for trolley and truck housings subject to tensile or compressive stress. No combustible material should be used.

5 All bolts should be of the through type, and be equipped with approved lock nuts or lock washers.

6 Where access to the crane is necessary, steps or stairs with hand rails should be used.

7 Platforms should be provided for changing and repairing truck wheels on end trucks and have stairways leading to them.

8 A platform or footwalk to give access to the crane shall be provided, which is accessible from one or more fixed ladders or stairways, and shall be not less than twenty (20) inches in width.

9 A footwalk shall be placed along the entire length of the bridge on the motor side except when the construction of the crane prevents or when such a platform would not ordinarily be used for the repair or maintenance of the crane. This walk should be at least six feet six inches (6′ 6″) below the bottom of the overhead trusses.

10 Footwalks should be placed across the ends of the trolleys at right angles to the bridge walks. When so placed they shall be not less than twelve (12) inches in width.

11 Footwalks shall be of substantial construction and rigidly braced.

12 No openings shall be permitted between the bridge footwalks and the crane girder. When wire mesh is used to cover such openings the mesh opening must not be greater than one-half (½) inch.

13 Each footwalk shall have a standard metal railing and toeguard at all exposed edges wherever practicable.

14 Not less than twelve (12) inches actual clearance should be allowed between highest point of a crane and the overhead trusses, and not less than two (2) inches between any part of the crane and building, column, or other stationary structure. Where there are more than two crane runways in parallel there should be a clearance of not less than twenty-four (24) inches between the extremities of the cranes.

15 Means of escape shall be provided for operators of hot-metal cranes.

16 The operator's cage shall be located at a place from which signals can be clearly distinguished and be securely fastened in place and be well braced, to minimize vibration. It shall be large enough to allow ample room for the control equipment and the operator.

The operator shall not be required to step over an open space of more than eighteen (18) inches when entering or leaving the crane. A pail filled with sand or an approved fire extinguisher shall be carried in the crane cage for use in case of fire.

17 A foot- or hand-operated gong, or other effective warning signal, shall be placed in a location convenient to the operator and be securely fastened.

18 Ladle and other cranes subjected to heat from below should have a steel-plate shield not less than one-eighth ($\frac{1}{8}$) inch thick and placed not less than six (6) inches below the bottom of the floor of the cage.

19 The cages of cranes hereafter erected shall be of fire-resisting construction.

20 All gears on cranes hereafter erected shall be provided with standard guards. This provision should apply to all existing cranes where practical.

21 No overhung gears shall be used unless provided with an effective means of keeping them in place, and keys shall be secured in an approved manner to prevent the gears from working loose.

22 Unprotected keys shall not be left projecting from ends of shafts.

23 The construction of the crane shall be such that all parts may be safely lubricated when the crane is not in operation.

24 The installation of the switchboard, wiring, and all electrical equipment must fully comply with the safety regulations of the United States Bureau of Standards and the fire-prevention regulations of the National Board of Fire Underwriters.

25 There shall be a main-line switch or its equivalent so arranged as to cut off all power from the crane, and so constructed that it may be locked in its open position. Convenient and lockable means should be provided on the floor for cutting the power from any part of the crane structure.

26 Open-type controllers shall have an asbestos-lined steel guard over the movable contact parts, both to protect the operator's eyes and to prevent articles from falling on contact parts.

27 A hoist-limiting device should be provided for each hoist.

28 Suitable brakes shall be provided for the hoist and bridge travel. Each hoist shall be equipped with effective brakes which shall be capable of sustaining at least two (2) times the full rated load.

29 The drums on cranes hereafter erected shall have a flange at each end to prevent the ropes from getting off the drum, and be

so designed that there will be not less than two full wraps of hoisting cable in the grooves when the hook is at its lowest position.

30 The hook block shall be of a type so arranged that it will lift vertically without twisting. The hook should be provided with a handle and should be painted white.

31 Bottom sheaves shall be protected by close-fitting guards, to prevent the rope from becoming misplaced.

32 Crane bumpers shall be provided, and shall be at least one-half of the diameter of the truck wheel in height. Both truck-wheel and trolley bumpers should be fastened to the girder and not to the rails. Bumpers shall be built up of plates and angles, or be made of cast steel.

33 Truck fenders shall be installed which extend below the top of the rail and project in front of all bridge and trolley track wheels, and shall be attached to the trolley or the bridge and frame. They shall be of a shape and form that will tend to push and raise a man's hand, arm, or leg off the rail and away from wheel.

34 Heavy safety lugs or brackets shall be placed on trolley frames and bridge end carriages, to limit drop to one inch or less if a wheel or axle should break.

35 A capacity plate showing the maximum capacity of each hoist in pounds shall be placed on each crane girder in such a manner as to be clearly legible from the floor.

36 A metal tool box or receptacle shall be permanently secured in the cage or on the runway for the storing of oil cans, tools, etc.

37 The trolley should be completely floored.

38 Cranes in outside service shall have the following additional provisions:

a Floors of footwalks shall be so constructed as to provide proper drainage.

b The cage shall be enclosed and of fire-resisting construction; there shall be windows on three sides of the cage, and windows in the front and the side opposite the door shall be the full width of the cage.

c The floor of the cage on outdoor cranes should be extended to an entrance landing which shall be equipped with a handrail and toeguard of standard construction.

d Where there are no members over the crane suitable for attaching blocks for repair work, a structural-steel outrigger should be arranged on the crane of sufficient strength to lift the heaviest part of the trolley.

39 All gantry cranes should be equipped with automatic warning signals.

40 The truck wheels of gantry cranes shall be provided with guards or fenders.

OPERATION OF CRANES

RULES FOR OPERATORS

101 Cranes shall be operated only by regular crane operators, authorized substitutes who have had at least two weeks' experience and training under the supervision of a competent operator, crane repairmen, or inspector; no one else should enter a crane cage.

102 Hands shall be kept free when going up and down ladders. Articles which are too large to go into pockets or belts should be lifted to or lowered from crane by hand line (except where stairways are provided).

103 Cages shall be kept free of clothing and other personal belongings. Tools, extra fuses, oil cans, waste, and other articles necessary in the crane cage shall be stored in a tool box, and not left loose on or about crane.

104 The operator shall familiarize himself fully with all crane rules and with the crane mechanism and its proper care. If adjustments and repairs are necessary, he shall report the facts at once to the proper authority.

105 The operator should not eat, smoke, or read while on duty nor operate the crane when he is physically unfit.

106 The operator or some one specially designated shall lubricate all working parts of the crane.

107 Cranes shall be examined daily for loose parts or defects.

108 Cranes shall be kept clean.

109 Operators shall avoid, as far as possible, carrying loads over workmen; this must be absolutely avoided when carrying molten metal or when using a magnet.

110 Whenever the operator finds the main or emergency switch open, he shall not close it, even when starting on regular duty, until he has made sure that no one is on or about the crane, and he shall not oil or repair the crane unless the main switch is locked open.

111 Before closing the main switch, the operator shall make sure that all controllers are in "OFF" position.

112 If the power goes off, the operator shall immediately throw all controllers to "OFF" position until the power is again vaailable.

113 When leaving the cage, the operator shall throw all controllers to "OFF" position and open the main switch.

114 The operator should not reverse a motor until it has come to a full stop, except to avoid accidents.

115 The operator shall pay special attention to the block, when long hitches are made, to avoid tripping the limit switch or running the block upon the drum.

116 The operator shall recognize signals only from the one man who is supervising the lift. Operating signals should follow an approved standard; they should be manual, never verbal. Whistle signals may be used where one crane only is in operation.

117 Before starting to hoist, the operator shall place the trolley directly over the load to avoid swinging it when being hoisted. This precaution is especially important when handling molten metal.

118 The operator shall not make side pulls with the crane except when especially instructed by the proper authority.

119 When handling maximum loads, particularly ladles of molten metal, the operator shall test the hoist brakes after the load has been lifted a few inches; if the brakes do not hold, the load should be lowered at once and the brakes adjusted or repaired.

120 Bumping into runway stops or other cranes shall be avoided. When the operator is ordered to engage with or push other cranes, he shall do so with special care for the safety of persons and cranes.

121 When lowering a load, the operator shall proceed carefully and make sure that he has the load under safe control.

122 If the crane is located out of doors, the operator shall also lock the crane in a secure position to prevent it from being blown off or along the track by a severe wind.

123 No person shall be permitted to operate a crane who cannot speak and read the English language, or who is under eighteen (18) years of age.

124 No person shall be permitted to operate a crane whose hearing or eyesight is defective, or who is suffering from heart disease or other ailments that might suddenly incapacitate him. A physical examination is required at least once each year.

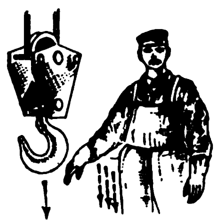
RULES FOR FLOORMEN

201 Floormen shall give all signals to the operator. Signals preferably manual should conform to the illustrated code given in Fig. 1.

HOIST — Make small horizontal circles with the hand, holding the forearm in a vertical position and forefinger extended

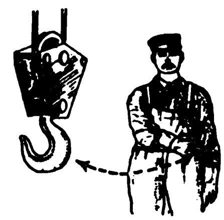
RACK — Jerk hand in direction of racking, with arm extended, hand just above hip, fingers closed, thumb extended horizontally

STOP — Hold position rigid, with arm extended and hand level with the hip



LOWER — Wave forearm downward with arm extended, hand below the hip and palm downward

TRAVEL — With forearm vertical and hand open with palm in direction of travel, wave forearm in direction of travel



EMERGENCY STOP — Move hand quickly to right and left with arm extended, hand level with the hip

FIG. 1 ILLUSTRATED CODE OF MANUAL SIGNALS FOR CRANE OPERATION

202 Floormen shall be responsible for the condition and selection of all hoisting accessories and for all hitches and slings.

203 Before the operator moves a crane upon which an empty chain sling is hanging, the floorman should hook both ends of the sling to the block.

204 Floormen where necessary should walk ahead of a moving load and warn people to keep clear of it. They shall see that the load is carried high enough to clear all obstructions. Permanent high obstructions should be distinctively painted or otherwise marked.

205 Floormen shall notify the foreman in advance when an unusually heavy load is to be handled.

206 Floormen shall not ride or allow others to ride on the hook or load.

RULES FOR REPAIRMEN

301 Repairmen should have a crane that is to be repaired run to a location where the repair work will least interfere with other cranes and with operations on the floor.

302 Before starting repairs, repairmen shall see that all controllers are thrown to "OFF" position; that main or emergency switches are opened; one of these shall be locked.

303 Repairmen shall immediately place warning signs or "OUT OF ORDER" signs on a crane to be repaired and also on the floor beneath. If other cranes are operated on the same runway, they should also place rail stops at a safe distance or make other safe provision.

304 When repairing runways, repairmen shall place rail stops and warning signs or signals so as to protect both ends of the section to be repaired.

305 Repairmen shall take care to prevent loose parts from falling or being thrown upon the floor beneath.

306 Repairs shall not be considered complete until all guards and safety devices have been put in place and the block and tackle and other loose material have been removed.

John Price Jackson, *Chairman* John H. Barr Melville W. Mix M. W. Alexander G. R. Olshausen John W. Upp William A. Viall Carl M. Hansen Wm. P. Eales	Sub-Committee on the Protection of Industrial Workers

No. 1573 *a*

GRAPHICAL CONTROL ON THE EXCEPTION PRINCIPLE FOR EXECUTIVES

By Frank B. Gilbreth, Providence, R. I.
Member of the Society

We have stated many times that the greatest waste in the world today is from unnecessary, inefficient and ill-directed motions. Many people think that this statement refers only to such activities as those of the bricklayer, the shopworker and other kinds of mechanics and manual workers. It does refer to them, but by no means to them only. It refers to the activity of every one and, by no means least, to that of managers and all other executives.

To one trained in the sciences of management and motion study, nothing is more ridiculous and pitiful than the average executive when he tries to enforce new motion methods on those farthest below him in the industrial scale, while he at the same time commits nearly all the motion wastes in his own personal work. The personal work of the executive should consist as much as possible of making decisions and as little as possible of making motions. General recognition of this fact has resulted in the common practice of assigning to the executive one or more secretaries, or clerks, to relieve him of certain parts of his work which involve mere motions and less important decisions than that part of the work retained by the executive. This procedure varies in degree according to the kind of work done by the executive and how well he realizes the possibilities of eliminating waste through the use of the "exception principle" in management.

Many executives have used this principle unconsciously, and have carried the practice to the extent of making decisions result from a study of sheets of figures on which "amounts which are less than those of the last fiscal period" are carefully noted in black ink and the amounts which are greater are carefully noted in red

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ink. Written causes of the fluctuations are conspicuous by their absence.

INFORMATION A SATISFACTORY CHART SYSTEM SHOULD AFFORD

Some executives, but fewer by far in number, are furnished with charts which show by means of comparable curves the increase or diminution in outputs, costs, overhead expenses and, in comparatively rare instances, even in results as compared with budgets. As compared with an organization which has no cost system, such a recapitulation even in the form of an "expenditure system" and such cost statements and graphical charts are a great step forward. No cost system nor chart system, however, can be considered really satisfactory unless it fulfills the following requirements; *i.e.*, it must determine and show —

1 What the quantities of individual outputs should be (prophecies of outputs)
2 Prompt records of individual outputs
3 What the costs should be (prophecies of costs)
4 Prompt records of costs
5 Causes of fluctuations and deviations of outputs and costs from prophesied outputs and costs.

The executive may have much to do with originally determining items 1 and 3; but after the computations of 1 and 3 have been completed, he can best attack the problem of enforcing items 2 and 4 and, also, of determining 5 by the use of graphical charts. He should be provided with charts which will tell him how promptly such records of output and cost have been made; or, in other words, how much time has elapsed between the completion of the output and the recording of it and its attending costs.

A long experience has shown us that the by-products of a properly operated chart system are even more valuable than its direct product. We find that the psychological effect of the variable "promptness" itself makes the curves representing outputs and costs fall more nearly in the proximity of the established norms and locations prophesied on the charts. Such charts give the executive and his colleagues accurate measured information of deviations from class in all departments. The motions that an executive would expend in getting information by such old methods as, for example, walking through the works to see with his unreliable eyes conditions which are not typical, partly owing to his presence, bring results of little value compared with the results that can be obtained by the same amount of time and motions concentrated on those facts and conditions which cause the great fluctuations from the desired output.

While this fact is generally recognized, the number of installations of chart departments throughout the country is increasing with surprising slowness. Even in those organizations where there is a satisfactory cost system supplemented by charts with curves — showing results as compared with expected conditions and ideals, the executive too often finds himself flooded with charts. Then being human, he postpones studying them. As a result, many benefits which come from making records of outputs and costs *promptly* are lost by his delayed action. It is here that the "control on the exception principle" plays the important part.

It is obvious that the foreman, or other functionary, should see *promptly* all the records of output in his particular department after they are achieved. In most cases he will be able to handle his duties still more satisfactorily if he, also, sees the cost of the outputs of his department. The time of the over-foreman, however, who may have several foremen and departments under him, is too valuable to have him, also, examine with care *all* the records of all the men under him. Consequently, he should be furnished with information in concise form, in order that as little as possible of his time may be taken. This has often been furnished him in the form of "averages."

USE AND VALUE OF PROGRESSIVE AVERAGES

Ordinary averages have their use. Progressive averages are, however, more valuable, because they show the trend of progress and of efficiency. It sometimes pays to make ordinary averages, but the value of examining such ordinary averages is slight compared with the benefits which result from concentrating the same amount of motions and attention on those individual cases that brought the average away from the ideal. A case of "bad average" may be the excuse for "putting the foreman on the carpet," but the results of this do not compare with the good results that are derived from having the over-foreman investigate promptly the case or cases that spoiled the average. For here the over-foreman, the foreman and the workman have every opportunity to secure super-coöperation, and the over-foreman can give that constructive criticism

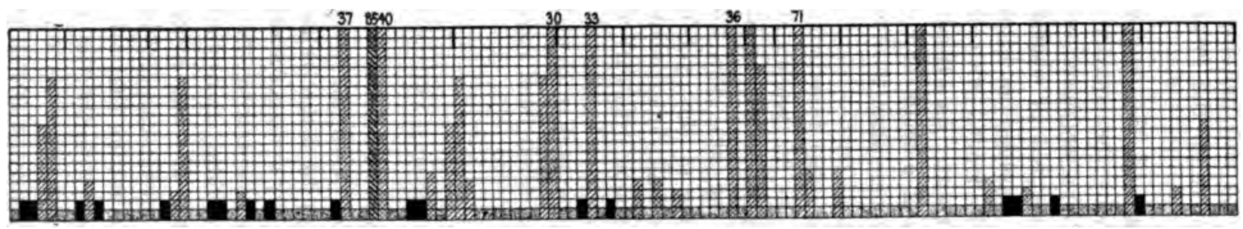

Fig. 1 Simple Illustration of a Chart Used on the Exception Principle

Such a chart may, for example, show the promptness of performance of an information bureau of a factory in charge, say, of trade publications, estimates of work under consideration, and records of design sketches. Each horizontal space may represent a specific request for information and each vertical space the time required to supply it. Colors may be used to indicate notable time differences. The point of the exception principle is that the chart shows at a glance when it took too long for a definite inquiry. Then it may be the duty of one individual to investigate all instances when the time is over 10 min., and of another when the time is, say, over 20 min. In the so-called three-position plan of promotion (in which the occupant of any particular job gets supervision from the one promoted from it and serves as a tutor to one in line to fill it), the result is an effort to keep the time intervals within the safety zone, so to speak, so that No. 1 of the three-position plan is not likely to be demoted to improve the service.

afforded by reason of the experience and knowledge he should, and probably does, possess.

Moreover, the decisions of the over-foreman can be made more quickly, for he has the information which comes from locating the trouble accurately. Instead of "tearing out" the foreman or the workmen, he will find, from the causes marked on the chart, that the worker's low output is due to lack of the proper tools; to his not having been furnished with tools in standard conditions; to the routing system having failed to give him proper materials in the right quantities, in the right sequence, at the right time; to something which has gone wrong with the equipment or surrounding conditions; to the man's not having been properly instructed; to there having been an unwise selection of the man or the machine, or both, for the particular job. Whatever the cause, the tendency toward killing coöperation by having a "brainstorm" prior to an investigation will be gradually eliminated.

The worker, also, is more careful not to do anything which is not expected of him, because he knows that the exception will surely be noticed by the executives higher up and will interfere with his chances for promotion or transfer to work of a more desirable kind. Knowing that they will be investigated properly will create a tendency on the part of the foreman and the workers to coöperate with others whose work affects theirs, or who in turn may be investigated. This coöperation becomes general, and sooner or later becomes a habit.

OPERATION OF THE EXCEPTION PRINCIPLE

Now the time of the executive next above the over-foreman is still more valuable than that of the over-foreman, and so on up to and including the managing director or president. No executive should make a routine motion of handling, turning over or examining charts containing data, either normal or with considerable deviation from class, where the causes of the deviation can be handled properly by those in lower executive positions. The exclusion of such cases can be obtained by having the executive determine *zones* on the charts, it being understood that as long as the points fall within the zone he is not to see the charts unless he specially requests to see them. He is, however, to have sent to him, for initialing, any chart having a point that falls outside his excluded zone.

In other words, he sees the charts on the "exception principle." So long as everything goes right — so long as points do not fall outside his zone of exclusion, he is not disturbed. When they do fall outside, he needs to and does see them. He can easily make other standing orders, in cases, for example, where the line is a certain distance away from the normal for a certain number of days in succession.

An executive of any class will find it beneficial to see exceptionally large cases of deviation on the desired side of the line so that he can recognize and appreciate and take a personal interest in cases of unusual efficiency. It is through such cases that he gets in touch

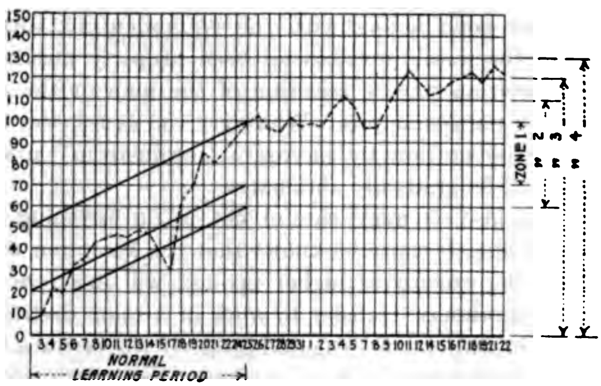

FIG. 2 EXAMPLE OF EXCEPTION-PRINCIPLE CHART COVERING OUTPUT OF AN INDIVIDUAL, INCLUDING LEARNING PERIOD

In this chart zones are established so that points of performance falling below or above certain limits come automatically by that fact to the attention of certain individuals. The lower the record or the higher the record the higher in the industrial-management scale will be the individual who must inquire into the conditions. The results may indicate poor instruction or incompetency in foremanship or they may bring the executive into personal contact with the worker in the matter of commendation for unusually high performance.

with unusually good methods. This is a check on the exception principle of the time-study-man's work. It also gives the executive valuable opportunities on the exception principle for proper managerial decisions in cases of the selection of candidates for promotion under the "three-position plan" of promotion and organization building. The curves showing progressive averages of departments may be examined at times farther and farther apart, these intervals to be determined in each particular case by the favorable or unfavorable comparison of records of such averages showing outputs and costs, with the prophesied outputs and costs. The executive is thus relieved later of work which is necessary at first, but which is not necessary when the particular case is running satisfactorily.

It is impossible to prophesy with accuracy without motion study and time study what the amounts of outputs and costs should be. But once these have been made and the actual outputs and actual costs approximate those prophesied, the high executives should devote very little time indeed to inspecting this class of charts. Instead, they should spend their time on other work, other departments, and more important things where their supervision will bring more valuable results.

It will be seen that these "Output, Cost and Causes Charts," with the "exclusion zones," enable the executive to eliminate the motions required for general oversight and inspection until a place on a chart is brought automatically to his attention where he can actually help those below him and furnish them with better instructions for handling their work more efficiently; or for making such changes as will naturally result in promotion, or the selection or shifting of individuals better fitted to do work elsewhere. The possibilities of relieving the executive of unnecessary motions and of enabling him to be more efficient in his own work are not exceeded in the case of any manual worker.

IMPROVING THE QUALITY OF EXECUTIVES

This system of Graphical Control for Executives on the Exception Principle has another by-product of great value to us as a nation. It will provide us with better executives — a subject of great importance, not to be treated lightly; for when the executive tells those below him "how to do it," the effect of his decisions, the value of his judgment, his efficiency as an executive, his standing as a leader of a department or undertaking, or of the destinies of those workers who have thrown their lot with him, can be measured and compared just as efficiently and accurately as can the output of any other workers.

In the great commercial war which will soon be upon us, as a nation and as individuals, we need better leaders. Our national and individual prosperity depends upon how efficiently we are led. This method of graphical control on the exception principle provides for the human side in developing personal contact, the square deal and the recognition of the efficient man, and at the same time saves the motions and time of the executives.

1573 b

TOPICAL DISCUSSION ON MANAGEMENT

It is the custom of the Society occasionally to hold meetings for topical discussion and interchange of data at which the papers or other contributions are informal in character. At the New York meeting, December 1916, such a session was arranged by the Committee on Meetings for a discussion of Recent Developments in Industrial Management.

The preceding discussion on Graphical Control on the Exception Principle by Frank B. Gilbreth was presented at this session, an account of which will be found in THE JOURNAL for January, 1917. In what follows are excerpts from the remarks of several of the speakers which it is desired to place on permanent record. — EDITOR.

COÖPERATION OF WORKMEN

E. E. BARNEY spoke on How to Secure the Coöperation of the Workmen and made a unique comparison of the laws of physical science with those of human nature. In summarizing the subject he said that one should classify the individual employee according to his ability; warm him with the interest he deserves; enlighten and train him to the mutual advantage of employer and employee; and attract him by fairness and consideration. One's organization at all times should carry the proper message of the spirit of coöperation to and beyond the workmen, who after all are the most valuable asset of any industry and should be the most valued. Buildings decay and machinery wears out; but workmen grow more valuable and more desirable with time, if the employer does his part by showing a willingness to coöperate with them.

R. B. WOLF pointed out the danger in attempting to push the realm of facts, or exact science, beyond the natural or generic field (as typified by the laws of chemistry, physics, mechanics, etc.). He showed that management, because it was directing men, was functioning in another realm, and that "the will of man" was essentially free and creative. The really efficient organization, therefore, was the one which scientifically studied the problem of human relationships and how to produce a desire on the part of the workman to do good work. The management must coöperate with this will and not antagonize it. In this way it would give the employee a chance to do creative work, in which he would have an opportunity for self-expression, which was the most fundamental instinct in human nature. The real purpose of scientific analysis of the forces and materials used in manufacturing was to give the men in the organization a chance to create intelligently the necessary conditions for increased economy of operation. The true science of management, therefore, was how to develop men, and simultaneously with a development of their own individuality, to make them conscious of their place in the organization; a realization, in other words, of their obligation to their fellow workmen, which, of course, generates *esprit de corps.*

W. S. Rogers emphasized the importance of the *man* as an asset in organization, and said that 30 years ago his friend and preceptor, the late John E. Sweet, had built a special machine which he went to see, and after inspection remarked, "It is a success." "No," said Uncle John, "I have made a big mistake. I should have built a man first."

In 1896 Mr. Rogers went to Cincinnati on a two-years' contract to take charge of an "old scrap pile of a plant" which was losing money. At the end of the contract they were making twenty per cent dividends and the owners complimented him highly for the things he had developed as a mechanical engineer. As a matter of fact, he "had not used any mechanical brains whatever." He had simply studied and experimented with the men. He had not added a single new machine nor made a single new jig. Neither had a man been discharged nor a man added to the payroll.

Twelve years ago he had the desire of his life in a plant where he did not have to ask a man on the board of directors what he should do. The cash capital "was $14.95 and they could not borrow a cent." Today the plant is a great success and its stock is selling for "six times its original par value."

This was accomplished through coöperation with the men to make them feel that they were a part of the organization. Houses were put up, which they were assisted in buying. Profit sharing was introduced, and payments were made *monthly*, a plan Mr. Rogers had learned in England. A shop legislature was introduced, with a House which met in the superintendent's office and a Senate in the manager's office, and in case of necessity they meet jointly. No man could be discharged without bringing the case to the House, and if possible another place would be found for him in the works.

A special training school for employees was started. Charts were introduced in the House for showing the output of production. Once a month there was a meeting of the entire force at which the different men or departments were shown where money was lost; and how such loss took money out of the other partners' pockets and how it might be rectified.

COÖPERATIVE AND SERVICE WORK FOR EMPLOYEES

A. J. Baker gave an interesting account of the employees' service department installed at the works of the Cincinnati Milling Machine Company. In discussing the value of such departments with one well suited to appreciate their worth, he said that the statement was made that their most important function was so to arrange the relationship between manager and employee that the earning capacity of the employee could be increased, and in thinking over this statement he had arrived at the conclusion that it came as near as any to summarizing the whole result that it was desired to attain. If the amount in the pay envelope was increased, everybody was happy; if not, no matter what the system, nothing had really been accomplished.

It might be interesting to note that the average individual loss from sickness amounted to nine days per man per year and that this loss had in some cases been reduced to as much as two days per man per year through the maintenance of an industrial hygiene department, this effecting a saving of one week's wages, or 2 per cent of the annual income of the workman.

At the plant in question, a combination of premium and bonus systems was in use, although the speaker believed that the founders of these, Mr. Halsey and Mr. Taylor, would hardly recognize their own children. In effect this system planned for a standard time, the achievement of which was rewarded by a high bonus. If the workman failed to reach this standard time, he still had the opportunity of coming under the premium time, which was 40 per cent in addition to the standard time and under which he secured one-half the saving. Furthermore, should the operator be of exceptional skill, he could still further increase his earnings by beating the standard time, in which case premium was paid for all additional savings.

With an arrangement of this kind, there was an inducement for high-grade men who did not content themselves with just reaching the standard time set; there was an inducement for the standard man who just measured up to the time-study limit; and for the man who was really not worth the money he was being paid, there was an inducement to work up to the point where his ability would correspond with his hourly rate. As a result, it had been possible to hold the organization together in a very satisfactory way during a period of labor trouble that had been expensive to both Cincinnati manufacturers and workmen. This time-setting system, Mr. Baker said, was the starting point of the special relations that had been established.

In the study of men and conditions it was decided that the physical condition of the employee became a most important element, and a physician was hired and established with headquarters in the factory, who at once proceeded to establish new points of contact between the officers of the firm and the men in the shop. It was aimed as far as possible to return somewhat to the old feelings and close relations that had previously existed between employer and employee, but which it had often seemed impossible to maintain under present industrial conditions.

It used to be that the apprentice was a member of his employer's family and the relationships were, under such conditions, extremely satisfactory. In contrast with this was the paternalism now practiced by some of our large organizations, which Mr. Baker felt was not entirely in keeping with the institutions of this country, particularly when extended to the time outside the factory hours. His firm limited itself to investigation of the man during shop time, and since it was difficult to have the officers of the company attend to their business duties and also act in an intermediary capacity to bring about these important personal relations, this work was entrusted to the physician, who by virtue of his office and the character of his work was a suitable and satisfactory man for the purpose.

It was shown that as many as 80 to 100 cases were sometimes treated per day in a plant employing only 1750 men. They did not come necessarily because of serious illness or injury, but often because of minor things in respect to themselves or members of their families. The physician thus got quite close to the men and was often consulted on matters not related to personal health, such as loans, etc.

Among the matters investigated was the question of food. Oakley, the suburb where the works are located, had inadequate lunch facilities, and the men who did not live in the vicinity either brought a lunch with them, which they ate cold, or else they went out to a saloon. A cafeteria with self-service was installed by the company and was turned over to the employees, who were told to use the equipment for operating their own lunch room. They appointed a committee to purchase their own food and set their own prices; in short, they had a lunch room with which the company was in no way connected and it had proved extraordinarily successful. Here again the physician came in, for he was able to suggest to the men the dietetic value of certain foods and to assist them not only in securing good nourishment but in reducing the cost. By this means the health of the employees had undoubtedly been helped.

A mutual aid association was also started which was run by the employees, and to this the firm contributed the life-insurance features, the amount of insurance being based on the years of service. Here again the physician, having the confidence of the men, was in many cases able to forestall serious illness. It was surprising how many of the men were found to have incipient diseases unknown to them, but which were discovered through the examinations and visits and treated successfully because of their early discovery. Bad teeth were found responsible for much absence and inefficiency. In certain cases careful treatment of this condition resulted in almost doubling the earning capacity. Further, the physician could very frequently state to the superintendent that a certain man was not physically fitted for the kind of work he was doing, which would lead to his being transferred to a department adapted to his condition, thus saving the money lost in breaking in a new man and winning the gratitude of the one who was transferred.

No man might be discharged without having the medical department look into the situation; and, as a result of their investigation, it was often possible to find the cause of dissatisfaction and to put the man in some other department where his peculiar make-up would not bring him into conflict with either his fellow-workman or the foreman of the department.

Referring to a remark by Mr. Rogers to the effect that one should "study to make them believe that they are a part of us," Mr. Baker said it was an ideal expression of what his firm was trying to do. While his own statement at the outset as to the financial end being the most important might seem a little cold-blooded, yet in the end they were trying to make the men feel they were a part of them, and not only feel but believe that they really were a part of them, and the building up of such a feeling was the whole thing. He also outlined a plan for paying a monthly or quarterly bonus, according to the average output of production per man in addition to the extra earnings which a man would receive from his own individual increased production.

STANDARDIZATION IN INDUSTRIAL MANAGEMENT

SANFORD E. THOMPSON, in introducing the subject of standardization, said that whereas apparatus and objects used in manufacturing had been extensively standardized, machinery had been improved, plant layout perfected, and in many cases the personnel of the managing force carefully selected, the standardization of the processes of manufacture was as yet almost ridiculously crude. In construction we find laborers selecting their own shovels regardless of the class of earth or the adaptability of a long or short handle, and we see them using shovels in whatever way they see fit. In the shop workmen are not only allowed to select their tools, but allowed to utilize their material and to do the job in any way they choose. It cannot even be said that they use their "initiative;" they simply follow in a rut, often a very deep and muddy rut, worn out by their predecessors on the particular job in the particular shop.

Those who have given thorough consideration to modern industrial methods appreciate that there usually is one best way to do each job and that every operative ought to handle every piece of work by this standard method. Some may not agree with this, perhaps, because they emphasize too strongly the difference in make-up of different operatives; but the tendency is toward standardization. Not until we arrive at standards in methods of performing individual operations, as well as standards for the layout and general handling of the work, can we hope to have a properly managed plant.

Referring to stop-watches as an element in the study of methods, the speaker said that while they are a necessity for standardization, if they are used simply to find the time in which an operation is being performed instead of the time and the manner in which it ought to be performed, they are comparatively useless. It is the getting at the proper methods and determining the proper tools and machinery by scientific study that means real saving in time and material.

In piece work as ordinarily practiced, either the output is very variable or a level rate of production is arbitrarily maintained by the workman. Where this level rate is maintained, employees are all following the slowest and poorest man, and the management is helpless because it does not know the best or the quickest way. By giving definite instructions in all classes of work and training the worker just as is being done in many cases of machine-shop practice, it is possible for the majority of the men to learn instead to work at the speed and by the methods adopted by the most highly skilled, with large resulting benefit to themselves.

In certain kinds of work, standardization may be practically the whole thing. It is amazing how standardization simplifies certain processes which have been considered so variable as to be impossible to make uniform or produce uniform results. A good example of this is in the cooking of sulphite pulp. The wood comes to the digesters, which are large steel tanks, perhaps 12 ft. in diameter by 30 ft. high, where it is cooked in bisulphite of lime liquor under high pressure. Formerly it was the universal belief of all pulp manufacturers that on account of variations in moisture, differences in quality of wood, variations in the character of the acid, and many other variables, the largest of which was designated as "pure cussedness," the uniform length of time for cooking and a uniform product were impossible to obtain. Cooking, in other words, was an occult process known only by a few experienced men, and even with those experienced men operating the digesters, variables were impossible to allow for. Now, as a result of standardization which involves the manufacture of bisulphite liquor as well as the cooking process itself, it has been found possible to produce pulp which from the paper-mill standpoint is practically uniform in quality, from ordinary wood coming from various sources, and to bring off the cooks at lengths which vary not over five or ten minutes.

The standardizing in the digester house consisted essentially in determining the curves of pressure and of temperature which should be followed during the process of cooking, giving these definite standards to the men handling the digesters, and requiring them to follow the standard set. In order to obtain the most uniform results, however, it was found advisable, and very effective, to make definite records of just how much each cook varied from the given curve and give a bonus for maintaining the curve. Similar plans have been found very effective in other classes of work, illustrating very clearly how useful is the form of bonus payment for maintaining quality at a high standard and in other cases of maintaining standards of output.

H. L. Gantt said that a lot of people had been talking efficiency and getting up fine charts, and when asked what they did with them they would say that they did nothing. We were all making charts to show how efficient the workman was, but very few were making charts to show how efficient the management was. There might be workmen doing something that need not be done at all, and doing it very efficiently. Until our efficiency system reaches the executive, and we measure his efficiency with the same degree of accuracy that we now measure the efficiency of the employees, it will not amount to very much.

Mr. Gantt referred to the Valuation Session (reported later in this volume) and said that most of the valuation by accountants and financiers gave the valuation of bricks and mortar and machinery; but he had noticed in Mr. Polakov's paper to be presented on Valuation of Industrial Properties vs. Valuation of Industrial Methods, that the value of machinery was not in the machinery itself but in how it was used—in other words, in the human element behind it. Mr. Rogers had told about the little old plant that had gone to pieces and was not paying expenses when he took it, but by putting men and brains into it he got something out of it. How would public accountants have valued that plant before Mr. Rogers went there, and afterwards?

No. 1574

ACCURATE APPRAISALS BY SHORT METHODS

By John G. Morse,[1] Boston, Mass.
Non-Member

Inventories and appraisals of industrial properties have been in existence, to some extent, for many years. The earlier ones were crude and few in number. Modern methods of bookkeeping rendered inventories more necessary. For the purpose of fixing the amount of insurance to be placed and for carrying the plant as an asset, nearly all the larger manufacturing concerns finally had a home-made appraisal in one form or another. Personal inspection of a great number of these has led to the conclusion that the methods used were almost as numerous as the appraisals and that the accuracy varied as much as the methods.

2 With the advent of more elaborate cost systems and with the campaigns for efficiency, the need of more reliable work in this line became manifest. It was, therefore, but natural that appraisal companies should enter the field, offering to inventory property to the most minute detail and affix a value to each item.

3 But long before the first-mentioned practice of making home appraisals became at all common, the Factory Mutual Fire Insurance Companies inaugurated a system of making approximate valuations for the purpose of fixing the amount of insurance. From a beginning of a simple examination of the books of the assured, they soon took the shape of rough inventories, the appraiser using his judgment in affixing values to both buildings and contents.

4 As the Factory Mutual Companies developed their methods of making plans and inspections and their tests of fire-preventing appliances to the complete forms in use today, so also did they develop the making of insurance appraisals.

[1] Appraiser, Insp. Dept., Assoc. Factory Mutual Fire Ins. Cos.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

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5 At the present time these consist of inventories in as much detail as necessary to determine accurate values, while a system has been adopted for obtaining a reasonable estimate of all the minor items without going into elaborate detail.

6 These appraisals are made without cost to the assured and have been appreciated to such an extent that they are constantly used as a basis for bookkeeping and cost systems, for mergers and other business uses.

7 In explaining this method by which a large manufacturing plant can be accurately appraised in a very short time and with what seems like a comparatively small amount of work, I shall endeavor to show how useless it is to waste time on more laborious detail. It is an admitted fact that no appraisal can be made without incorporating many figures that are based simply on estimate. It is, therefore, needless to carry the detail in any direction where it will not add to the accuracy of the final result.

8 Disregarding new factories, where records of actual cost are easily available, let us consider an older plant and particularly one where no records of cost have been preserved. A building may be measured so carefully that the exact amount of each material is known. The market price of each material may be obtained. Yet the amount for waste must be estimated and the amount for labor and all contingent expenses must be estimated.

9 The inventory of machinery may be made to the last trifling detail, yet the exact value can be obtained only for standard machines. Each special machine, as well as the cost for erection of every machine, must be estimated. Constant changes in market values always affect the accuracy of the result, but, aside from that, the amount of depreciation on both buildings and machinery must be estimated (or largely guessed at) before the actual present value is found.

10 The more experience one has in making appraisals, the more one realizes the truth of the above remarks and also that in making estimates the great law of averages counterbalances all minor errors.

A METHOD OF APPROXIMATION

11 The method developed by the appraisal department of the Factory Mutual Companies I have been asked to explain. It is a method based entirely on the theory that if the larger factors are carefully appraised, the less important items may be estimated in groups.

12 The property to be appraised is divided into two parts —

buildings and machinery. The buildings are considered as empty structures. All elevators, piping, wiring and indeed anything that can be removed without altering the building, are classed under machinery. Attention might be called to the fact that in insurance appraisals, foundations and other underground work are ignored, but the method would be practically the same were they included.

13 Machinery is divided into machines proper, shafting, belting, piping, electric wiring and furniture and miscellaneous apparatus. Special small tools, dies, patterns, drawings, molds, lasts and any objects of similar nature that exist in large quantities in the plant under consideration, are treated independently. The subject "miscellaneous apparatus," as its name implies, covers all objects of minor value not easily classified under any of the other heads.

14 The price values used for buildings and all the subdivisions of machinery are based on *replacing new at today's market* (regardless of original cost), and these price values are then depreciated as judgment dictates. I will explain our method of obtaining and applying new values before discussing the subject of depreciation.

BUILDINGS

15 As stated above, by the term "building" we mean simply the empty structure. In appraising we use the square foot of floor area as a basis. Many architects and engineers use a factor based on the cubic foot of contents, but it is floor area that gives manufacturing facility, and the figures thus obtained are also useful in other parts of the appraisal, as will be shown later. As a groundwork we use the tables prepared by Charles T. Main, Mem. Am. Soc. M. E. These must, of course, be changed for different heights of stud, degrees of finish, thickness of walls, etc., and also for constant fluctuations in the market value of labor and materials. As new factors are obtained for reinforced concrete and other modern forms of construction, the tables form a guide for consistency as lengths and widths vary.

16 It is in appraising buildings that the uselessness of extended detail is strikingly apparent. Those who have had occasion to call for bids are aware of wide variance in figures submitted. Suppose two identical buildings are constructed, one by the highest bidder and one by the lowest. They may be more than ten thousand dollars apart. Is the actual cost of either a fair appraisal figure? An estimate based on the square foot of floor area that showed a figure half way between the extremes would give a much more reasonable value.

Indeed, Mr. Main states that several contractors have told him that they use his tables as a check before submitting bids.

17 The actual cost of foundations is far more unsafe to use for appraisal purposes. A building resting on extremely expensive foundations in quicksand is worth no more than a duplicate building resting on a ledge. A fair estimate for the cost of sufficient foundations in average ground would be a better figure to use for either when appraising an industrial property. With appraisals of public-service corporations conditions are different, but that comparison is not to be discussed here.

MACHINES

18 It is our custom to make a complete inventory of all fixed machines, not only producing machines but those used for power and for maintenance. This inventory also includes all elevators and cranes. There is no hard and fast rule, but the practice is to list everything (exclusive of furniture, small tools, etc.) that has enough value to be worth considering separately. In making this inventory, the shortest description that will identify a machine is used. When ordering a new machine, detailed specifications are usually necessary, but to incorporate a long description in a list of machines simply serves to confuse when one is searching for an item and gives no real aid in determining the value.

19 As an example, for a cotton spinning frame "one ring frame 256 sp. 3″ met. bds. sep." means a ring frame with 256 spindles 3-in. gage, with metallic thread boards and separators. "One engine lathe 14 × 6 comp. taper" means an engine lathe with 14-in. swing, 6-ft. bed over all, screw-cutting, without special gearing, with compound rest and taper attachment. Other special features that materially affect the cost are, of course, noted. In each case the above brief notes are all that are necessary to determine the value of the machine in question. By adhering to this method the fixed machines in a large factory can be inventoried in a much shorter time than would be deemed sufficient by one not acquainted with the work.

20 Having completed the inventory, it is a simple matter to appraise standard machines either directly from price lists or from data compiled from individual quotations and actual costs obtained when making other appraisals. Of course, all such prices are given in confidence and are used without the source of the information being divulged. In the same way machines built on order or by the

owner can be appraised as data on similar machines accumulate. Allowance is made in all cases for the cost of freight, cartage and labor of erection. It should be understood that no prices are fixed arbitrarily by the appraiser. On the contrary, the figures are discussed with the owner or manager and as much assistance as possible obtained from their records.

APPROXIMATION ON MINOR ITEMS

21 The appraisal of buildings and machines forms the most important part of the work, both in time consumed and in results obtained. It would, therefore, be inconsistent to devote a greater amount of effort to the smaller part of the value. It is here that accurate "short-cut" methods have been developed in the appraisal department of the Factory Mutuals. The subjects will be described in the order mentioned earlier in this paper.

SHAFTING

22 By figuring in detail representative lines of shafting, with couplings, hangers and pulleys included, various factors per lineal foot erected have been obtained. Experience soon teaches which factor to apply when examining lines of shafting in actual use. Opportunities are repeatedly presenting themselves to compare the results obtained with those of a detailed appraisal, and the accuracy of the approximate method is confirmed.

BELTING

23 The most satisfactory method by which belting may be rapidly inventoried is to measure the main belts by eye. The machine belts can then be classified in groups, as similar machines will require about the same amount of belting. Individual motor drives are lessening the amount of belting in a modern factory, and where the machines are scattered and the belts vary greatly, it is usually easier to list them by using eye measurement than to attempt to apply any shorter method.

PIPING

24 It is probably needless to call attention to the fact that a detailed inventory of piping and pipe fittings would require an amount of time out of all proportion to that devoted to the remainder of the work. To obtain a "short-cut" method has not been as easy as with

shafting and belting, due largely to the fact that there is a much greater variety in the material to be considered and that much of it is hidden.

25 The different uses to which the piping is put enable the appraiser to divide the subject into classes and these classes can be treated by different methods.

26 Automatic-sprinkler piping can be appraised at a price per sprinkler head or, by what amounts to the same thing, by a price per square foot of sprinklered floor area. This applies to the piping inside the buildings only.

27 Steam heating, where consistent throughout the plant, can also be appraised on a floor-area basis. Where the amount of steam heat varies, a price per lineal foot of coil or per radiator can be easily ascertained. The heating pipe in dry rooms, lumber dry kilns, paper dry lofts, etc., can be treated in a similar manner.

28 Gas lighting is not common today, but where found can be appraised at a price per light. Where a building is provided with gas lights at frequent intervals, a factor per square foot of floor area can easily be estimated.

29 The piping used for manufacturing purposes presents the greatest difficulty. The steam and water pipe in a steam-power plant will vary little from a standard figure per horsepower of boiler rating. For long runs of covered steam pipe through rooms where there are few or no outlets, a price per lineal foot can be used.

30 In plants where there is a great quantity of piping, particularly in bleacheries and paper mills, small factors cross-checked by large factors can be used. These factors are obtained from time to time when actual costs in a new plant are available. In all cases the large factor for steam is based on the horsepower rating of the boiler plant. In a bleachery the large factor for water is based on the number of kiers; in pulp and paper mills, for water and stock piping, on the number of pulp grinders, digesters, wet machines and paper machines, as the case may be. The small factors in all cases are varied with the pieces of apparatus. By using both methods in estimating a plant a sufficiently accurate total can be obtained. With factories making dyes, chemicals, soaps, etc., the same method can be employed.

31 Air piping, though quite extensive in modern machine shops, never amounts to more than a very small fraction of the value of the plant. The runs of pipe are easily followed and a factor per lineal foot or per machine supplied can be ascertained.

32 Fuel-oil and gas or gasoline piping is of still less importance from a value standpoint and can be easily estimated in a manner similar to that used with air piping.

ELECTRIC WIRING

33 Electric wiring should be divided into two main groups, one for lighting and the other for power. The number of light outlets in each room can usually be obtained from the electrician on the premises, and a factor per light applied to those of each size. The horsepower of motors can be obtained from the inventory of machines. As it will cost more to wire several small motors than a large one equal in power to the others combined, they can be divided into two or more groups, according to conditions, and a factor applied to each group. A percentage of this total amount of both light and power wiring should be added for cables at the power house or, in cases where current is supplied from outside, at the transformer house. For electroplating wiring a factor per square foot of surface of plating tanks has been found accurate. For the wiring to electric-welding machines, where these are supplied from a central dynamo, a factor per machine can be used. In plants manufacturing electrical apparatus a large amount of test wiring is found. Actual inventory of this can usually be avoided by consulting with the electricians in charge.

FURNITURE AND APPARATUS

34 In every plant there is a large amount of equipment (exclusive of small tools, dies, etc., which will be considered later) that cannot be classified under the head of machines and yet does not belong to any special class. This equipment is covered under the term "furniture and apparatus." It is a very elastic term, for with many of the items the appraiser must use his judgment as to whether or not to list them in the machine inventory.

35 All furniture, benches, racks, trucks and scales come under this heading. In a textile mill all bobbins, spools, etc., in a metal-working plant all boxes, trays and cans, indeed, every miscellaneous article, that cannot be included under any other definite title, can be covered under this.

36 To list all of these items in detail would be an arduous task. Many would be destroyed and new ones added before the list could be completed. The value, however, would be insignificant when compared with the total value of the plant. It is not difficult, however, to obtain a fairly accurate estimate.

37 In a cotton mill a factor per spindle can be used and in a woolen mill a factor per set of cards. In any textile mill, however, it is possible to obtain from the different foremen the approximate number of bobbins, spools, roving cans, filling boxes and trucks, while the appraiser can easily estimate, room by room, the value of what other miscellaneous apparatus there is. In a metal-working plant the benches and racks can be appraised by a factor per lineal foot; the number of tote boxes, cans and trucks can be obtained from the foremen, and the other miscellany can be estimated as above. A similar method can be applied to any plant. Office furniture can be appraised, without listing, by using judgment.

SMALL TOOLS AND DIES

38 All machine shops contain an equipment of small tools and all metal-working plants an equipment of dies. The proportion that the value of these bears to the value of the whole varies greatly with the class of work. In many concerns fairly accurate records are kept of the cost of such small tools and dies. Experience has shown that factors per producing machine can be used that will give a fairly accurate appraisal. Where a closer estimate is needed, the tool and die store rooms can be examined and fairly representative shelves or drawers can be appraised to obtain a factor or factors to be used for the remainder.

PATTERNS AND DRAWINGS

39 Depreciation plays an important part in finally determining the value of patterns and drawings, but, as mentioned above, this subject will be discussed later. To obtain the new value of patterns it may be possible to study the pattern-shop pay roll for a given period and add a proper amount of overhead expense. So many patterns are obtained from outside, however, that the most satisfactory method is to examine the shelves in the pattern storage, classify them into a few groups and estimate a price per square foot of shelf area for each group.

40 With drawings, the pay roll, plus overhead expense covering the entire time since the drawings began to accumulate, gives the most accurate basis. For replacing cost, the present rates of pay should be used, but a deduction should be made for time spent in experimental work in either case. A good cross-check is to obtain from the head draftsman the total number of drawings, divided into several groups for size and cost.

MOLDS, LASTS, ETC.

41 Molds, used both for hard and soft rubber and in several other classes of manufacture, are usually carried on accurate inventory by the owner, but if not, they can be appraised by methods similar to those used for small tools and dies. Lasts in shoe factories, copper shells in print works, print rolls in wall-paper factories and dandy rolls in paper mills are almost invariably carried on inventory. In the rare cases where not so listed they can be easily counted, divided into classes and an average price to be used for each class readily obtained.

ELECTROTYPES

42 Electrotypes, wood cuts and similar objects used in printing offices have a standard cost per square inch. As they are kept in drawers or on shelves it is comparatively easy to ascertain the total number of square inches of each kind.

MISCELLANEOUS

43 The above-mentioned subjects are the ones most commonly found. Occasionally the appraiser visits a manufacturing establishment which is out of the ordinary and special auxiliary apparatus is found in large quantity. By using some one of the methods described for appraising small tools, patterns, etc., the value of these extras can be obtained.

STOCK AND SUPPLIES

44 In making insurance valuations the Factory Mutual appraisal department almost invariably relies on the inventories of the assured to obtain the value of stock and supplies. In rare instances, however, where such are not obtainable, a rough average obtained from previous appraisals of similar plants can be used.

ROUGH APPROXIMATION

45 The methods described above enable the appraiser to make an accurate estimate of the value of buildings and machinery in a much shorter time than would be used were all items listed in detail. To obtain quickly an approximate valuation, a comparison can be made with other appraisals on a basis of the square foot of floor area or of the number of producing machines. In specific instances the comparison can be made per spindle, per set of cards, per kier or per paper machine.

DEPRECIATION

46 Depreciation, recognized almost universally today, has been considered in appraisal work only in comparatively recent years. There is a wide difference of opinion as to the amount of depreciation that should be allowed on buildings and various classes of machinery and as to the method of caring for such depreciation in the book accounts. Indeed, another argument in favor of the "short-cut" method of making valuations is the fact that two competent appraisers would show more difference in the value of a plant on account of varying opinions as to the amount of depreciation to be allowed than in errors made in estimating the new value.

47 There are also several kinds of depreciation. The greatest in amount is that applied to "second-hand" property and is governed by the desirability of the article or building under consideration rather than by its physical condition. A more conservative depreciation is applied in bookkeeping that the assets of a concern may be carried at a reasonable figure. The recent income-tax law has brought the discussion of this subject more prominently before the public than ever before.

DEPRECIATION FOR INSURANCE

48 A still more conservative depreciation is used for insurance purposes. A building that houses a going concern or a machine that is turning out a salable product deserves less depreciation from an insurance standpoint than from any other. It is the intention of the Factory Mutual Insurance Companies to deduct for depreciation what judgment shows is deserved for actual wear and tear and for obsolescence. The methods used in estimating this depreciation vary with buildings and with different kinds of machinery. In some cases a sliding scale can be applied and in others an average figure. These methods will be described in the same order as the methods of appraising for new value.

DEPRECIATION OF BUILDINGS

49 A building badly out of repair naturally deserves fairly heavy depreciation. A building in good repair, but so antiquated in size and shape that it is manifestly unsuited for the uses to which it is being put, also deserves a reasonably heavy deduction. When, however, a building is of such dimensions that it perfectly answers its purpose, has remained plumb and is constantly kept in repair, actual age has little influence on judgment. It is considered that about five

per cent of the new value is enough. In other words, buildings are not depreciated a certain per cent a year, but have a flat amount deducted on account of condition and not on account of age.

DEPRECIATION OF MACHINERY

50 Machines vary greatly both in the manner in which they wear out and in the rapidity with which they go out of date. In rare cases where a machine has been practically superseded in the market by one that will cost much less, it is better practice to use for a new value the cost of the less expensive machine rather than show an excessive depreciation. As a rule, the amount deducted applies chiefly to wear.

51 With machines that need repairs at all points from time to time, a day arrives after a period of years when it is better to throw them out altogether and replace with new rather than continue to repair them. Practically all textile machines come in this class, as do engines and other power-plant machines, and also some machine tools, wood-working and paper-working machines. To all of that nature a depreciation table is applied, allowing 2, 2½, 3, 4 or 5 per cent a year, deducted from the net and not from the gross. If a machine is entirely rebuilt, it is usually considered to be worth at that time within five per cent of new value and the table is applied for succeeding years. In either case the probable average life is ascertained and the table that best fits is used, but seldom is the depreciation carried to a point beyond 50 per cent.

52 There are many kinds of machines where the main portion, sometimes as much as 80 per cent of the total value, remains for years with practically no wear. The small moving parts, however, wear so rapidly that they are constantly being replaced. This is true of a great variety of machine tools, metal-, wood- and paper-working machines. With these it is considered that the wearing parts are always in a state of 50 per cent depreciation, and the amount deducted is half of the perçentage the value of the wearing parts bears to the total value of the machine. This method also applies to rolling mills, rubber mills and calenders where the frames and gearing remain intact for years and the rolls constantly wear down and are replaced..

53 There is another class where neither the depreciation table nor the definite average described above can be used. This includes most of the machinery in paper mills, bleacheries and dye works where wet processes are used. These machines wear rapidly and are frequently rebuilt. Paper machines in particular are composed of a train of parts, and from time to time different sections are either

rebuilt or removed entirely and replaced. The deɪ
cases depends upon the condition at the time of th
not influenced by the age of what remains of the o

DEPRECIATION OF SHAFTING, ETC

54 It is quite apparent that to obtain the age
ing, piping, etc., would in nearly all cases be im
general figures for depreciation are safest to use. Sh
slight wear that depreciation is seldom recognized.
the custom, however, to show either a slight deduc
posely record the new value at a conservative figure
of poor arrangement an amount in excess of what is

55 Main belts wear slowly, while machine b
average 50 per cent wear, so that, as a rule, the
belting is depreciated 33⅓ per cent.

56 Piping will last for years, except where expos
Pipe covering and valves show wear, but piping as a
depreciated more than 10 per cent.

57 Electric wiring wears little and is usually ke
to date on account of the rigid rules of both local au
insurance companies. It, therefore, seldom deserve
tion.

58 The miscellaneous equipment classed unc
"furniture and apparatus" is made up of objects m
constantly wearing out. The amount is, therefore,
ated from 20 to 50 per cent.

59 Small tools, dies, print rolls and electrotyp
they are affected to a great extent by obsolesc
drawings, molds and lasts are subject to depreciati
reason only. In determining the amount to be de
new value of any of these the appraiser must ascert
tion of the equipment is indispensable or practically

APPRAISAL COMPANIES

60 Having explained the method by which an a
can be made, expending the minimum amount of tim
well refer briefly to the method of making an appra
painstaking detail. This method is best illustrated
the appraisal companies. It is perhaps needless to
the Factory Mutual Fire Insurance Companies giv

their appraisers without extra charge to the assured, they are not to be regarded as in competition with anyone.

61 Opportunity has been given again and again to examine the records and the results of such appraisals. The finely prepared volumes are found to contain a list of substantially everything on the premises. They are overburdened with extended descriptions of buildings and machinery, far more than is necessary to determine value and decidedly confusing to the searcher for individual items. The usual proportion of the total value recorded is based on estimate only and invariably there are errors large enough to counterbalance whole pages of minor items so carefully noted. The law of averages usually balances these errors, so that the final result is approximately correct, but no more so than that of an appraisal made by the shorter method already described.

COURT CASES

62 It has often been stated that the courts will not accept a valuation unless made in detail. One has but to demonstrate how largely estimate must enter into every appraisal, regardless of the method used, to prove that the shorter can be as accurate as the longer. Although the Factory Mutual Fire Insurance Companies have always endeavored to avoid giving testimony in court, circumstances have compelled such service on rare occasions and the results have been in every case satisfactory.

IN CONCLUSION

63 This so-called approximate or "short-cut" method of making appraisals has been employed by the Factory Mutuals for a long period of years and, as has been explained, foundations and other uninsurable properties are omitted, while the depreciation charged off is for insurance purposes only. It should be emphasized that the figures thus obtained are not intended for other uses, though safe enough for proportioning a cost system. But the method described can be employed in making an appraisal for any purpose whatever by extending it to cover all property, whether or not insurable, and increasing the depreciation to a point more in accord with bookkeeping methods.

DISCUSSION

HARRY BARKER[1] wrote contrasting the high
costs of public-utility appraisals with the moderat
from the use of the author's method. It would l
how the author would apply his methods in ca
field where the effect of seeking out original cond
tion might change the valuation result by 100
over figures based on present conditions as superfici
where faithful adherence to a reproduction val
conditions might be most inequitable to earlier inv

MORRIS KNOWLES said that it was obvious th
were costing enormous sums of money. There w
sary refinement, for which the people paid in the
vania all matters of fact were referred to an engin
sisting of one engineer appointed by each side and
of the Commission sitting as chairman. Engineer
each other across a table found it easy to reach ag
going into an unnecessary amount of detail, an
from trying to persuade an attorney or the Court.

W. L. WHITTLESEY[2] thought that if depre
thing it was an item of operating cost and nothing
expensive statewide valuation of the New York Tel
costing about $3 per thousand of value, the idea
thorough engineering job, and to put an end to a
been dragging along for ten years. An important fa
now in valuation work was the price of copper;
knew, had tripled within the last year or two.

EDWARD W. BEMIS,[3] wrote that in his opinion
agreed, the side not giving a detailed appraisal w
advantage; and that only great reputation on th
praisal company would enable a short-cut appr
against a detailed one. Unfortunately, most of t
panies of large reputation had applied such high
and so strongly endorsed going value as to be lo

[1] Associate Editor, Engineering News, 10th Ave. and 3
[2] 15 Dey St., New York.
[3] City Hall Square Building, Chicago, Ill.

trust by cities. Personally, he believed that where both sides exhibited a coöperative spirit, short methods could be employed which would yield justice to all concerned.

Chas. T. Main, in a written discussion, said that for the purpose of purchase or sale, for condemnation, or for establishing a fair cost value for accounting purposes, it was unnecessary to make elaborate schedules of all the items in a plant; oftentimes a rough approximate value per unit, such as a spinning spindle, was near enough, if the plant was what could be termed standard.

The first step in determining the value of a plant was to ascertain its replacement cost, and then to compare this with the cost of a new plant constructed on modern principles, the output of which would be equal in amount to and of as good quality as that of the existing mill. A comparison should then be made of the cost of running the two mills, and if the organization of the existing mill could be so changed as to make it as efficient as a modern plant, the cost of such change would be deducted from the replacement cost. If it could not be changed, the greater annual operating expense would have to be capitalized to some reasonable per cent and the capitalized sum deduced from the value new.

After thus determining the value of the plant, if new, account had to be taken of depreciation. To be safe, it was necessary to mark off each year an amount that would fully cover all possible decrease in values from age, wear and tear, and obsolescence; but if this were carried too far the value on the books would become less than the true value, therefore revaluations should be made at stated periods, say every five years, at which time the write-off for obsolescence for that period, at least, might be readjusted and corrections made for decided changes up or down in replacement values and for any extraordinary repairs or renewals.

In a general way he had found that the useful life of the common types of machinery that had been run about ten hours a day, had received the usual good care, and had not been superseded by machines that would render them unprofitable to run, was about 30 years for cotton machinery and 50 years for woolen and worsted machinery. Well-designed and substantially built mills, when properly kept up, should be good for 100 years.

In appraising a property in operation there should be added to the physical value the "going concern value," or the amount required to get it started so that goods could be marketed. This amount was made up of (1) interest during const gineering expense; (3) interest, depreciation, tax for the period required to start up the mill and get market; (4) cost of starting up; and (5) allowanc damage to product. He had found that for the three items it was safe to take 10 per cent of the and for the remaining items, 5 per cent, or a total o

Chas. W. McKay[1] submitted a written discus outlined a method of manufacturing-plant apprais after current practice in public-utility valuati method is described in detail in The Journal, pp. 134–135.) He said that while the author's by short methods might satisfy his clients, or his had sufficient confidence in his judgment and kn he would be able to substantiate his valuation in t for claim adjustments, nevertheless it was quite surance organizations would be somewhat loath to made by outside engineers and based upon the broad-gage methods. And it was probable that an mitting a short-method appraisal would experie difficulty in getting the Court to accept it in the ev over the value of a manufacturing property or in u for capitalization. He thought that the auxiliary ment, such as elevators, piping and wiring, shoul sible, be considered a part of the building rather t the machine equipment. While some of the ap by the so-called appraisal companies contained ne short-cut methods of the author, on the other ha a little too approximate in their nature to warr adoption for engineering valuations. Probably th medium between two plans, which could best be re ing, and to a certain extent copying, the method public-utility appraisement field.

The Author, in closing the discussion, said t there were so many different opinions in regard to what should be considered depreciation, made it necessary to go into detail. As to the cost of h thought the range was from 15 to 25 cents per $10

[1] With McMeen and Miller, Engineers, 1454 Monadnoc

asked to outline his procedure in the case of an industrial property. He understood that in Vermont, in appraising the telephone company, they measured the length of every pole and put a different price on each pole where it varied six inches in height. He would have taken an average price per pole, and ascertained the number of poles to the mile and the number of miles of line the company owned, and obtained the value by using that factor. He would have averaged the poles and wires in that way. In the case of a railway valuation, he would take the track at so much per mile; he would take the trestles according to their construction and classify them into different types, at so many dollars per lineal foot. In short, he would go over the property, naming a unit factor on every part of the property and apply it; going into detail in each individual case for one sample only.

No. 1575

HOW DOES INDUSTRIAL VALUATION DIFFER FROM PUBLIC-UTILITY VALUATION?

By John H. Gray,[1] Minneapolis, Minn.
Non-Member

VALUATION, SPECULATION, MONOPOLY AND REGULATION

It is my object in this paper to show that value logically and necessarily means exchange value, and rests on the basis of capitalized earnings, real and estimated. It depends on strategic advantages and involves speculative elements. When under the police power an industry is declared a public utility and limited to a fair return to be determined by disinterested parties, the object is to destroy this strategic element and the opportunity for speculative gains. The property thereby has a genuine incumbrance placed upon it. What is a fair rate has no relation whatever to value, but is a matter of public policy to be determined in the supposed interests of the public welfare. The fundamental purpose of regulation is to limit the profits to a just amount, and thus to take that part of the property for the public which would inure to the private owners if the same monopoly remained in private hands unregulated. For the value of such an unregulated private monopoly has no limit but the necessities of the people and what they are willing and able to pay, rather than to do without the services of the monopoly.

2 Valuation has no significance save in connection with the specific purposes for which it is made. For want of space, I shall on this occasion deal chiefly with valuation of public utilities for the purposes of fixing rates, and beg that you will always remember that we are concerned solely with certain phases of the great problem of monopoly.

[1] Professor of Economics, University of Minnesota.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

3 I regard it as extremely unfortunate that we lay such emphasis on valuation in rate cases; but when the Supreme Court declared in 1898 that the owners of a public utility are entitled to a fair return of income on a fair value of the property in use at the time of valuation, it was inevitable that emphasis should be placed on valuation until the law is changed. Unfortunately, the law has not yet been changed, although I am led to believe that the method of determining a fair valuation has undergone a wholesome change, and that both the courts and the public-utility commissions are now in a much more scientific state of mind. This mental attitude is much more conducive to human progress than that which depended on the older theories of valuation. Under present law, at least, valuation of some sort is necessary if we are to attempt regulation, and to determine what in the language of the law is a fair value. If time permitted, it would be easy to show that any attempt at valuation is subversive of effective regulation, and, that, in basing regulation upon valuation, we are traveling a fallacious road that will never lead to a sound public policy in regard to public utilities. If the law is sound on this point, effective regulation is impossible, for the reason for attempting regulation at all rests on the desire and necessity of destroying some, at least a portion, of the speculative elements existing in private property, which affect seriously the value of such property.

4 Before we are in a position to discuss these elements in detail, it is necessary to give at least brief consideration to the relation of the limitations of strictly private property to the public welfare, and to take some note of the history of the limitations on the absolute rights of such property, in order that we may get a solid basis for comparing these limitations with those that we undertake to place upon public utilities. There can be no doubt that the history of property rights in America, for nearly three hundred years after the colonization of America, offers the nearest approach to absolute private property the world has ever seen. Time does not permit me to explain fully why this is so. Anyone who is familiar with the history of property, liberty and contract, and the fundamental assumptions of the theory of individual property and *laissez-faire*, will recognize the truth of the assertion.

5 This condition has prevailed so long that it might almost be said that the average American today resents restrictions of any sort and if he has a legal title to property, he asks insolently "Have I not a right to do as I like with my own?" the implication always being that the right to his property is an absolute right, and, that the mere

fact of legal title prevents the exercising of any restrictions upon the part of the State, and gives him a right to use that property irrespective of the rights of others or the power of the State itself. In fact, the owner of property was allowed until about the beginning of the twentieth century, to act almost as if his rights were absolute, although in the fundamental theory of all systems of jurisprudence, and, more particularly under the theory of the English civilization of which we are a part, he was enabled to act so, simply because the circumstances did not lead anybody to enforce the limitations which the theory of the law justified. In other words, owing to the fact that we were peopling and conquering a continent, and that our material resources appeared, even to the wisest and most disinterested, practically inexhaustible, the owners of private property were left, in practice, to put their own interpretation on their rights, and it was nobody's business to call them to time, or to make an attempt to enforce the rights of the public or the State over against the individual owner.

6 It is only in the last generation that we are beginning to ask what are just and proper limitations of strictly non-utility or private property, and do our system of jurisprudence, our present constitution and laws, permit us to enforce those restraints. In other words, the most significant tendency of the last two decades, the world over, and more particularly in America, has been towards the socialization of wealth; or to express it in another form, towards the limitation of the rights of private property.[1]

7 For instance, it is only a single generation since the courts of this country uniformly declared that the making and distribution of gas was a private industry. The most conservative would not undertake to maintain that view today. The placing of electricity within the class of public utilities is much more recent. But, perhaps, no other industry illustrates this point as well as the common carrier and the extent of regulation, which, in theory, is universally recognized as proper today, compared with that which we considered permissible a generation ago. A mere study of the phrases "reasonable rates" and "unjust discrimination" will show the progress we have made, for good or for ill, along this line.

8 Note the changed attitude on this question in the piteous appeal on behalf of the railroads that the State fix the wages of their employees. Whether or not the State ought to fix these wages is apart from our present purpose, but think how completely and how

[1] Ely: Property and Contract, vol. I, p. 17.

suddenly the railroad owners and managers h
position on this point.

9 The same social progress has brought urba
countries all kinds of land, under special publ
civilized country except America. What is tru
true of the whole range of insurance. The char
public mind towards the rights of banking illustr
A century and a half ago banking and insurance w
common law occupations freely open to anyone,

10 I am not for a moment saying that it w
this property, and all these human activities und
all, much less do I mean to imply that the regula
ministered has in all cases been wise, honest, e
to the public. This would necessarily raise th
public and private ownership, the whole questio
and possibility of leaving important monopolistic
hands, unregulated. In fact, it would involve dis
of what is a fair and equitable distribution of w
tions, however important, — and their importa
gerated — are altogether too large to be enter
settled on this occasion. I shall deal with a s
question in another place, as that phase bears dir
of public utilities.

11 For better or for worse, both in our phi
law, we have taken what are known as public util
gory of unrestricted private property. It is very
technical sense, in law, they remain private pro
not the slightest economic significance, since the
taken from this property the chief attribute of
the chance of speculative gains. The doctrine of
lic utilities forever separates it economically fro
property.

REASONS FOR CLASSIFYING CERTAIN INDUST
UTILITIES

12 But why have we classified certain in
service industries, or public utilities? We hav
considerations make these industries of more in
whelming interest to the public welfare than a
businesses. We throw them together, then, in a
say, "These require a more careful public super

greater restriction be placed on the rights of such property, and for that reason we will call them public utilities." This grouping, however, does not aid us much in our quest until we stop to ask what social, economic and political reasons justify us in putting any one of these industries in this special class. Why should the owners of a gas company, for example, be subject to a different public regulation or have a different limitation put on their property from that put on the owners of a shoe factory? What is there in the nature of the one industry that makes it more dangerous to the public to leave it unrestrained? What harm would come from leaving the gas industry in the same field with the shoe factory? If we face this issue squarely and find the real difference in the character of the properties and what the property rights connote, we shall have found a real starting point for our theory of valuation.

13 The theory of all law, and more particularly of Anglo-Saxon law, has always been that one must use his rights, so as not to injure the rights of others. About a century and a half ago there grew up a theory in considerable measure corresponding to the needs of that day, that under a system of free competition one's own self-interest would prevent him from doing injury to the public. And so long as there was an abundance of free land and large opportunity to expand, accompanied by a high degree of intellectual, social and educational equality, the doctrine was measurably true. If one asked too high a price for goods or services, someone else would offer them at a less price, and the first would lose his customers and voluntarily cease his extortion. The creation of fixed capital on a large scale, as a result of power machinery, made this doctrine unsafe as a practical rule of action. For the theory assumed that when a business became unprofitable, the owner would know of another that promised more profits and that he would be able without a total or material loss to transfer his activities, including his capital and labor, to the more promising industry. The limitation upon the opportunity for intellectual and educational development, which our inequality of wealth has brought about, when taken in connection with the enormous increase in the means of communication of knowledge and the consequent power of united action, has made every assumption underlying the theory of competition subject to so high a degree of limitation as to justify us in saying that competition, as above defined, is no longer a dominant characteristic of our highly complex American life. Therefore, self-interest can no longer be depended upon to hold in check the recklessness, greed, and predilection for extortion of the

owner of even private property. It is perhaps nee
tion in this place the fact that the interpretation
sider a business corporation a "person" entitled t
an individual, has greatly complicated the problen
emphasis to the above statement. The change is
described by that felicitous phrase of Brooks Ad
motion." Government regulation then may be a
the breakdown in the application of competition,[1]
solve the difficulty; it merely brings us one stage fu
sophical analysis.

14 If competition, as indicated, is not a rea
is the danger of leaving private parties unrestrain
tries where competition no longer works smootl
I think if we go thoroughly into this point, w
never before, that with the limitations of huma
more complex the machine became, the more ext
of labor, and the larger the mass of fixed capital, p
dangerous to the public was any mistake of the own
to turn this into a modern phrase, we should say t
the speculative element in private property, the l
and holding for an increase in price, that furni
greatest danger. One was allowed this freedom in
the supposition that the natural resources and th
property rights could be carved were virtually u
when one undertook to develop these resources and
his own, he was doing a public service. Since w
resources are not unlimited, and that we must lea
we have discovered that, in this particular, what o
is taken from another, or at least prevents the otl
like manner and to the same extent. In other wo
face with monopoly.

15 The world has now become alarmed on tl
regulated, privately owned monopolies, and with a
of civilization, we are likely to see the things pre
as halfway between private property and public u
declared public utilities. We have apparently com
that civilization cannot be maintained, on its pre
nothing of advancing it, unless this ever-enlargin
utilities can be made to render an adequate servi

[1] "The need of regulation in some other way than thro
be faced once for all." Taussig, Principles of Economics, v

charge and without unjust discrimination. Increasing emphasis must be put on the fact that stability, regularity and universality of continuous, non-discriminatory service are infinitely more important to the public than the general level of charges or the amount of profits made by a company.

SPECULATIVE ELEMENT IN PUBLIC UTILITIES THE ULTIMATE REASON FOR REGULATION

16 This brings us to the heart of the whole matter, namely, to the strictly speculative element in public utilities. One who has watched the growth of private property is inclined to say that one of its fundamental elements is speculation, that is the purchase of property not merely for the sake of income, but for the sake of possible gains in selling. In no country of the world has this doctrine had a more baleful influence than in this. Just as the world today does not permit unregulated cabmen to wrangle with each customer about cab fares, so it will not permit privately owned utilities to fluctuate in value so as to attract droves of speculators and thus divert the interest of the owners from the possible earnings of the property to the speculative gains[1] to be made in the stock market on the securities of the same property. I take it that this is the ultimate reason for regulating what we call public utilities, and for declaring that they are entitled to a fair income only. To express it in another way, speaking broadly, we still allow in this country the traditional speculative element in ordinary private property. Where that property tends toward monopoly or is accumulating into units of such vast size as to be threatening, we either declare it a public utility and limit it to fair gains, or we throw about it many of the restraints placed upon public utilities proper, such restraints as we have thrown about banking, insurance and certain other industries. When all these expedients fail we resort to the final remedy of public ownership.

17 The industries under consideration have in the past been unregulated and highly speculative. Whatever we may finally consider is justice in regard to their reckless, unsavory and speculative past, we may safely say that the world has now come to the view that for all future investments the amount contributed by the owners

[1] "The law does not intend that this business shall be a speculation in which the water company and the consumers shall respectively win or lose upon the casting of a die, or upon the equally unpredictable fluctuations of the markets." — Van Fleet, J., in San Diego Water Co. *v.* City of San Diego, 118 Cal., 556 (1897).

shall be the basis of calculation for rates, and that
be met by varying rates of charge and income rath
tions in the value. For so long as we have fluctua
owing to false methods of valuation, we shall have
speculation to bring about increased gains. This
the view of the world a generation ago, but sul
interested students accept it now. The difficultie
all turn on an attempt to "unscramble the eggs," o
or to settle the relative amount of blame of differen
that have gone wrong in the past. We are trying
of value to meet conditions of affairs that grew up
cance of public utilities was realized and befo
thought of. The problem is intensified by the
laissez-faire, and competition under which Americ
all of its town boosting, stock watering, building
demand, and so on. The natural and psycholog
that day no longer prevail, and the subject of val
the question how we are going to take conditions g
petitive circumstances and adapt them to an age
as little friction and injury as possible. In attacki
must never forget that speculation was the life of
and that the discovery of the evils growing out of t
the particular occasion for establishing regulatio
this clearly in mind we shall realize that regulatio
are inconsistent: if one exists, it destroys the other:
it prevents the other.

LIMITATIONS IMPOSED ON PUBLIC-UTILITY

18 We are here concerned with the meaning
the term "property": what are, in fact, and wha
limitations, if any, on the absolute rights of p
limitations on public utilities different from those
property, and are they more severe? There can b
present tendency on this subject. It is all toward
tion of wealth and limitation of private rights. O
is the growing tendency towards public ownership a
the public enjoyment of property shows this. The
of the list of industries that are subject to more an
in the public interest clearly points in the same di

19 It is true that many such industries are c

but they existed for a relatively long time before they were classified as public utilities. For instance, the common carrier is old; and it took us a long time to realize that the parlor-car business, the pipe lines (gas and oil), gas companies, the telegraph, the telephone, fall into the same class because of performing similar functions, and, that, if left unregulated, they are capable of inflicting like monopoly, injury and abuse. So we have progressed step by step from the common carrier through the utilities named to the street railway, the use of electricity in all its manifestations, to conduits, cold-storage warehouses and the like. Furthermore, we have recently entered upon a broad and wide field, with ill-defined limits, that comes neither within the realm of public utilities nor private property, as these terms have heretofore been used, but occupy what Mr. Bryan calls a "twilight zone."

20 Justice Brewer discusses such cases in the Kansas City Stock Yards case,[1] where be considers various industries requiring some special regulation, but not such as we impose upon public utilities. One cannot view the progress regulation has made, for good or for ill, without realizing that the emphasis on property rights has changed fundamentally in recent years. In the opinion of this generation, property either public, semi-public, or private is looked upon more and more as a public trust. The attitude of the public mind, and of the law, has not only changed towards the industries mentioned, but has changed towards the industrial trusts as well; hence our anti-trust laws. Recently we raised the question whether mere size is cause for added regulation. For an answer to this question we must wait for the Harvester case to be decided. The subject naturally shades off into pure-food laws, building ordinances, height of buildings, and so on.

21 We are even coming very rapidly to place restrictions, or limitations, on property in the interest of what is supposed to be ethical advance; and town planning even is coming to be considered respectable. The old idea of the common law that one could be restrained in the use of his private property only where the health, morals, or physical safety of the community were concerned, is hardly consistent with present-day philosophy. The attitude of our country on the duties of the common carrier — the oldest of our public utilities — demonstrates this. It has had theoretically to serve all at reasonable rates and without unjust discrimination. Today the limitations upon all the public utilities are much more liberally in-

[1] Cotting v. Kansas City Stock Yards, 22 Sup. Ct. Rep. 30 (1901).

terpreted in the public interest and, as Preside the common carrier is held to "strict accountab

22 Similar principles apply to the ever i trades and professions that require a license or p enter upon them.

THE FAIR VALUE OF PRIVATE INDUSTRI

23 The question of what is a fair value o dustrial property depends principally on what it This is determined by the actual past earning earnings in the future. But the fundamental e future earnings is the established charge. In a therefore, the main assumption and starting po rates of charge, and the possibility of the conti In case a private property is not subject to effec only way of arriving at its value is still by estin all the circumstances, what it would bring if so conditions, and this again involves the assumptio derived from a certain rate of charge. In rate c find what is a fair rate, but if we proceed in th determined before we find the value. But we a supposition that we are seeking value as a basis is a fair rate.

24 This argument rests upon the *laissez-f* petition. The idea of competition rests on the the traffic will bear, and on the assumption tha a limit to charges and thus prevent extortion government regulation is necessary or desirabl very antithesis of competition. The necessity from the fact that competition is totally absen protect the public and keep charges within the in harmony with the welfare of society.

25 I have gone into this long preliminary a proper background, or foundation for our dis also, that things are not as hard and fast as w suppose, and that changes in our legal system changes in the constitution, are not so difficult seemed to be. But let us now come back to the

HOW SHOULD A PUBLIC UTILITY BE VALUED?

26 What relation then does the fact that we are dealing with monopolies of vast significance to the general welfare have to valuation under a system of public regulation? Since the monopolies under question are not sold frequently enough under comparable circumstances to establish a market price, or value, in case of expropriation, we do, in fact, estimate the value on the basis of capitalized earnings and on the legal assumption that existing rates are fair and just. This is not only a fact, but agrees also with our theory of rights in unregulated private property. There is a legal assumption that the existing rates are just, because they have not been judicially called in question or proved to be unjust. Take, for example, the vexed and admittedly unsettled claim of the public to a share, at least, in the clear surplus of a company acquired out of earnings which have also yielded dividends at a fair rate. It is conceivable that the question might properly be decided one way in the case of a company going out of business, and in exactly the opposite way for a company compelled to continue to serve the public, or, one way in regard to a surplus acquired before the era of regulation, and quite another way in the case of a surplus accumulated under regulation. In the one case the rate from which the surplus came might be considered both legal and just, in the other as both illegal and unjust.

27 But all careful and disinterested students now recognize, notwithstanding great vacillation[1] of the United States Supreme Court on the subject, that the necessity for this super-regulation arises solely from the fact that the industries now under review are monopolistic. It cannot be said too often that regulation is simply a substitute for competition.

28 But, however the speculations, stock waterings, frauds, extortions, excessive charges, surpluses and unearned increments acquired up to the present time ought to be dealt with, or may, in fact, be dealt with, social peace and human progress require that in the future the gains from these sources ought not to be, and cannot be (so far as they cannot be entirely prevented or prohibited), allowed to the owners of public utilities. They should go largely or wholly to the public and not to the private owners. This is necessary to prevent the very evils from which we now suffer, and which give

[1] Brass *vs.* North Dakota (153 U.S. 491), for instance, does not seem to be consistent with any rational doctrine of regulation or to rest on any sound basis of economics or of public policy.

the only reason for regulation. For it cannot l emphasis that, if we include the common carrie this discussion), the whole trouble with our pu turns on this question of surplus and unearn what party ought to bear the brunt of the Past claims ought to be compromised or adjust new start made on a sound basis.

29 If these three matters, surplus, une mistakes on both sides under competition, cou pletely, and finally settled, all parties would ag out much trouble, that all that the owners entitled to in the future is a safeguarding of the to an enterprise and an annual rate of income c risk involved. In proportion as the monopoli industries becomes recognized, the risk diminis

30 Nor should it be forgotten in this conne as to the action of the State is the first and one o of risk. The risks of adverse state action have by the attitude of the companies, and, not leas of false theories of valuation on courts and com of the companies. This causes the public to los in the courts and commissions but in all tried n and hence the public ever seeks new means towards public ownership. Until about the be century, and until the public realized the mon public utilities, the chief risk was from raiding and often instigated by the public authorities. ignored that the improved spirit and tone of our done much to make these raids less frequent an where the legislature and administration may them.

31 In private property we permit compe value to normal value, or what the early economi of production, and furnishes a ready market to value is at any time. Such value, on the ma mined by present earnings together with estimat that is, the earnings, real or estimated, are c price or value.

32 But it must never be forgotten that determined by rates of charge. It is therefore argument to show, that this method of valuation,

in the expropriation of private property (and, even of public utilities), can find no proper application in the attempt to determine what is a fair rate. Although this method is admissible in the valuation of public-utility property in the case of sale or expropriation, yet the public-utility property, while still dedicated to a public use and while its owners still profess a public service in connection therewith, not only owes the duty of a continuous service to the public, but is bound to perform that duty on reasonable terms in every detail, and more particularly at a reasonable charge, while the public utility is expropriated strictly on the moral and legal assumption of present rates, which for obvious reasons are presumed to be just and fair.

33 Under our constitutional system no private property can be taken for public use without just compensation and due process of law. But by the rulings of our highest Court many kinds of property may not be valued in a rate case. Of these the more important items are contracts and franchises. The reason for this rests upon the fact that the prime distinction between private property and public-utility property is that the public-utility property is created in view of the right to regulate, and this right is as much of a limitation on the absolute dominion over property as a farm mortgage or a recognized easement.

34 The average man dreads to face this fact as a coward fears to face death, lest, governmental power, if acknowledged, should become so wicked, vicious and irresponsible as to destroy mankind. But in this he fails entirely to realize that we hold not only all property, but our very lives on terms and conditions granted by the government. These terms ultimately depend on the judgment of the public as to what is fair and just. That is, life and property both now and always, are subject to the needs of the sovereign.[1]

PUBLIC-UTILITY PROPERTY ENTITLED TO A FAIR RATE OF RETURN

35 Public-utility property is, in fact, property with an incumbrance upon it. The doctrine that it is devoted to public use involves the idea that it must be content with a fair return. And what is a fair return is to be determined by an outside party, namely, the public, and not by the owners. Otherwise there could be no regulation; and the doctrine of regulation rests on the simple economic maxim quite as significant as the legal maxim, that, in important matters affecting the life, happiness and prosperity of the race, one

[1] See the long list of authorities, economic, legal, ethical, and theological, cited by Ely in Property and Contract, vol. I, 191-199.

cannot be left to be judge in his own case. What is a fore, must be determined, under our system of ju public authority,[1] with an ultimate appeal to the ; means that the moment the property is dedicated to a by necessary implication and by the law of the land regulation with all that it involves. The only limit: striction is the public judgment, constitutionally ex] only safeguard is the constitutional prohibition of property for public use without due process of law a: sation.

36 The novel theory has recently been put fo: utility property is not in fact dedicated to public use service is so dedicated.[2] But this theory in view of al scarcely tenable. Such a limitation amounts to the the property as compared with private property is p with a genuine servitude or incumbrance. If one his private property with a mortgage, he does not ex large a net income therefrom as if the property we cumbered. Just so, on investing in public-utility does so under the present law and practice of regu legal notice of this incumbrance of regulation. To in this sense, he is not free to do as he pleases with hi fesses the public service.

RESTRICTIONS AND LIMITATIONS ARISING FROM

37 If we consider briefly the essentials of reg established by administrative commission, under our practice, we begin to get an idea of the restrictions arising from the fact of regulation. To particularize liable for continuous service; the owner cannot mal vestment without specific consent of the public; he accounts and make all reports asked for on forms p: public, and bear the expense thus necessitated; he mu whether he will or not, and he must not discriminate, able it may appear to him to do so. Above all, he : with a fair return, the amount of which is to be de

[1] "No doubt the authority thus given to a commission reaching one, not consistent with the traditions of competitiv better or worse, competition has ceased in great branches of ind Principles of Economics, vol. II, p. 396.

[2] Jared How, Discussions of the Economic Club of San Fra

himself, as in the case of private property, but by the public, an outside authority. He may not alienate his property or discontinue his service without permission. In short, if absolute dominion over property is financially advantageous to the owner, we see how the owner of this kind of property is hemmed in on every side, and, it goes without saying, that his chances for making large speculative gains, such as are usually permitted in the case of private property, are greatly diminished. This is the object and the aim of all regulation.

38 It was such limitations as these that I had in mind when I said that the object of regulation was to check, prevent and prohibit speculative gains. The significance of these limitations may be seen in the case of most of the real estate, rights of way and terminals of the railroads, which are by far the largest financially and the most important of all the public utilities. In the technical and usual sense, and in the average case, the railroad does not even own most of its real estate in the sense in which a private owner owns real estate. Railroads have often come into the control of their lands by the right of eminent domain, distinctly a sovereign right; and they may use the land but for a single purpose, that of transportation. Their so-called ownership is not in a strict sense ownership at all. It is a partial, a restricted, or, what the Court calls a technical, ownership only, and amounts to a mere grant by the public, which is not given primarily that the so-called owners may make money, but rather that the public may be properly served, and at a fair price. Under our system of law land cannot be taken by a railroad for any but a public purpose, hedged about with all the limitations involved in the doctrine of property dedicated to a public use. The right to regulate, with all that this means, is the significant part of these limitations. If property is limited by deed or contract to a single specific use, it is worth less than it would otherwise be worth. The only way it can be restored to its former full value, unincumbered, is by removing the incumbrance. So, in the case of a public utility, the only way it can escape the limitations of regulation, which place decided bounds to the amount of the value in any case, is to disassociate the property from the public service. That is, the company must, with the consent of the state, discontinue entirely its public service and devote its property to a strictly private use. In fact, we shall never come to a clear view of the object or reason for regulation until we make the clear distinction between one's rights over property which is devoted to a public use, and one's rights over this same property when it ceases to be so dedicated.

39 The methods of arriving at a fair value are in the two cases: in the one case the property is sul brance: in the other it is unincumbered; in the one seek the most profitable employment: in the otl to a single use.

40 This is the foundation stone of regulation, sidered as a part of every contract, and every char of a public utility. We get the full view of the sig of such limitations if we go a little apart and study tions mean in condemnation cases. When the Fe condemns private riparian lands on navigable ri allow any value for the commercial power and ever and structures put in the river by authority of t license of the Secretary of War. The Court admits cost the riparian owner much money and that they but it holds that such title as the riparian owner has acquired with the rights of Congress over navigati property as a limitation, or a sort of incumbrance or Ethically and economically the right to regulate p similar incumbrance, which destroys such portion o unjust, and to that degree lessens the value of the

41 We must not lose sight of the fact that fr is at the very foundation of all property rights, and or in law be separated from property as protected fourteenth amendments and other constitutional would I be understood to say, or even to imply, th tional safeguards are of no significance in the cas property. The whole point of my argument, how declaring certain property affected with a public thereby necessarily given such an interpretation tional provisions as seriously to incumber the pro fore, have restricted the uses to which it can be p dedicated to a public use.

42 This classification of property, and this int constitution, together with the limitations on publi arising therefrom, take this property entirely out private property as regards its right to a rate of in

[1] "It is a qualified title, a technical title, not at his absol upland, but to be held at all time subordinate to such use of and of the waters flowing over them, as may be consistent the public right of navigation." U.S. vs. Chandler-Dunbar pany, 33 Sup. Ct. Rep., 672 (1912).

could earn if it were not dedicated to a public use and had not previously been classed by the law as a monopoly. Because it is entitled to less chance of speculative gains, and a fair return only, it has less value, if we are permitted to use the word value at all in this connection.

43 Free property is to be valued at its value for any legal purpose. Public-utility property, on the other hand, is not to be valued for fixing rates, for any other purpose than that to which it is dedicated, and much property is not to be valued at all, so long as the property is to remain in the public service. If the owner wants full value (with all its speculative chances) for a public utility, he must first discontinue his public service and devote the property to private use. The position of local or secondary franchises in valuation cases brings this out clearly.

VALUATION OF FRANCHISES

44 There can be no doubt, whatever, that under our system of law a secondary or local franchise is property, but it is not to be valued, in ordinary rate cases, unless there is some specific agreement or contract on this point that makes valuation necessary. Unless the franchise gives an exclusive right and carries an inviolable contract for a fixed charge (which removes the case from that of regulation), franchises are not valued in rate cases under the established principles of regulation. The cases where they are valued under contracts bring us back to a case of capitalized earnings, on the assumption that existing rates are fair, and, that, of course, for reasons already given cannot be assumed in a rate case.[1] An exceptional case of valuing franchises is that of the people *ex rel*, Westchester Street Railway Company. The Westchester case was decided by the commission on the basis of estimated earnings, and the case in court[2] really turned on the fact that the Court substituted its own judgment[3]

[1] Judge Thayer expressed the law clearly on this point when he said: "It is obvious that the income derived therefrom by the owner, before it was subjected to legislative control, cannot always be accepted as a proper test of value, because the compensation which the owner charged for its use may have been excessive and unreasonable." — Cited in Whitten, Valuation, p. 23.

[2] The Westchester St. Ry. Co. *vs.* the Public Service Commission, 158 App. Div. 251; 143 N. W. Sup. 148, July 8, 1903.

[3] Justice Harlan states the law in these words: "But the courts cannot after the board has fully and fairly investigated and acted, by fixing what it believes to be reasonable rates, step in and say its action shall be set aside and nullified, because the courts upon a similar investigation have come to a different conclusion as to the reasonableness of the rate fixed. There must be actual fraud in fixing the rates, or they must be so palpably and grossly unreasonable and unjust as to amount . . . . to the same thing." San Diego Land & Town Co., 19 Sup. C. Rep., 808 (1899).

for the judgment of the commission, as to what a f
is contrary to the best American practice. This C
remarked, in reversing the decision of the commissi
mission had a right to raise rates as well as to low
that the rates fixed by the commission were unreas
the commission considered them reasonable. This
right to regulate, or, at least, place the judgment of t
of that of the commission on the question of wh
rates; for the commission had estimated the incom
what it considered reasonable rates and found that
not support a higher valuation, than it found. The
is that in rate cases the franchises are not to be va
cost, irrespective of their market value at the time, b
capitalized earnings. This is plain from the mere f
case the fairness of the present income is the point a
that in a recent Montreal case the franchises were
but strictly on the ground that the rates were fixed
chise. The contract gave the company a monopol
public ownership, and presumably prevented reduc

45 A striking illustration of the non-value of f
their actual cost, is found in the way that Wiscons
subject. In a recent case[1] the Court said: "An ind
. . . . is simply an authority granted by the State
utility to do business in a certain community, *su*
legislation,[2] . . . . but when the guillotine has fa
that is, when the municipality has legally decided
right, and it becomes but a memory, can it be logic
even a nominal value. We have been unable to an
in the affirmative."

46 I find but one other recent case which app
the propriety of valuing secondary franchises. T
railroad case heard by Special Master W. A. Guntl
have rested his case entirely on a misinterpretation
Consolidated Gas case. That case certainly did n
rule, as the Special Master seemed to think, that fra
are to be valued in rate cases.

47 A great confusion has grown up on the s
franchises from the fact that the court of last re

[1] 154 Wis. 181 Appleton Water Works Co. *vs.* The Railro
31, 1913.

[2] Italics not in the original.

allowed value in rate cases for franchises. But a careful study of these cases shows that they were all cases where the rate was fixed by franchise contract.[1] In such cases of valid contract the value is properly calculated on the assumption of that rate. Justice Swayze, in a recent case,[2] says: "But where, as in this case, the rate is not fixed and may be changed, there is no stable basis upon which to calculate the value of the franchise. Since that value is dependent upon the rate, the rate must indeed be reasonable, but to assume a value for the franchise, in order to determine the reasonableness of the rate, is to reason in a circle. The value and the rate are mutually dependent, and cannot be fixed independently, if the one is to form a basis for the calculation of the other. . . . . That a special franchise, in the absence of an exclusive right, is property, only in a qualified sense, is the result of the right of the State not only to regulate rates, but also to authorize a municipality to supply itself, and thereby destroy the value of the special franchise. . . . . This argument is inadmissible where the public-service company has an exclusive right." This is very true, but it rests on the existence of a contract, which is not subject to regulation. While the Supreme Court must necessarily uphold valid contracts, the most significant thing about recent cases, on this point, is the length to which that Court will go in resolving all doubts as to the power of a municipality to enter into a valid contract against the city, and thus to declare that the contract claimed as a defense against regulation and as a basis for valuing franchises was never a valid contract for lack of power on the part of the city. Truly, where the right to compete, either through the municipality or another company, exists or is reserved, the property is taken subject to this easement, restriction, or limitation. The genius of our Government and law are against granting exclusive franchises. Such are frequently forbidden by the Constitution, are frowned upon by the common law, and are always looked upon with suspicion by American legislatures. Justice Swayze, in the case cited, makes an interesting comment on the statement that the franchise ought not to be valued because it was originally a free gift. He necessarily sticks to the constitutional idea that it is property and may have a value, irrespective of whether it is a gift or not, but calls

[1] Justice Swayze, in Public Service Gas Co. *vs.* Board of Public Utility Commissioners, 85 N.J. 658–660, points out the error in interpreting the Wilcox case.

[2] Public Service Gas Co. *v.* Board of Public Utility Commissioners, 85 N.J., 568–660.

attention with great emphasis and with some ing
that a franchise, as a free gift, has a direct beari
franchises, for the simple reason that as franchis
gifts, investors in a competitive company would no
the franchise or pay for it, but would merely seel
franchise and make their rates accordingly. They
destroy the original franchise. This is the heart
monopoly, and brings to light the fact that recog
not only an advantage to the company in prevent
it lessens the risk and therefore lowers the return r
new capital to enter the undertaking. Justice Sw
this point is as follows: "Since it is in the power o
about a supply without compelling the public to p
value beyond the actual cost of procuring it, it wo
so, and the effect would be to destroy the value c
chise of the existing company." These considerat
conclusion that logically no allowance should be r
of the special franchise in a case where it is not le
where the State still retains the right to fix rates.
new set of investors would pay for an old franch
exclusive.

48 This marks the line of cleavage between t
property and of public-utility property in valuat
tainly is the attitude of the Supreme Court of the
though a franchise is property. The difference
property subject to a servitude and a free propert

49 I think I have made it plain that we mak
not because justice or economic policy requires i
Supreme Court, in the case of Smythe *v.* Ames unf
that the company is entitled to a fair return on t
property. We are therefore compelled to make
ask how valuations of public-utility property diff
of non-public-utility property. This question nec
consideration of the elements influencing the price o
and a careful analysis of the elements by which p
differentiated from public-utility property.

50 It may be admitted that valuation is a cra
from Smythe *vs.* Ames with its shibboleth of a fai
value. Is the dictum the laying down of a sound
it a mere legal fiction, convenient enough in its wa
cal relation to present social and economic conditi

DETERMINATION OF A FAIR VALUE

51 The question then arises what is a fair value, and what is the correct method of arriving at the sum of it. What, in fact, is the utility in a given case entitled to? We know that these industries are vitally necessary to our present civilization. From this it follows that, if they are to be left in private hands, they must be so dealt with by the public as to make possible investors willing to contribute their money for what is considered a chance of receiving as high a rate of income as can be obtained in other somewhat analogous industries in relatively the same territory and accompanied by substantially the same degree of risk. In the long run, if this appeal is not made to investors, they will fail to contribute sufficient money to meet the increasing needs of a progressive society. In other words, the estimated income from public-utility investments must be the normal rate in the same community for investments accompanied by like risks. But if we follow the history of regulation, we see that the object as well as the result of regulation has been to limit risk by preventing speculation. It also appears that value in ordinary language means exchange value. Under a system of competition it means market value. In a case of monopoly value it would be that amount which would be paid for like property, under competitive conditions, if competition actually existed. It could be easily shown, if time permitted, that public utilities of significant size are so rarely sold under normal conditions as to make it impossible to determine their value, save on the basis of earnings. But as I have pointed out so often, to use this basis is to assume the justness of present rates. In fact, it would be entirely impossible for us to arrive at the value of a large public utility with a fair degree of accuracy, under regulation, even if it were desirable to do so. For so far as value has any significance, and so far as the word is used in its ordinary economic sense, value is determined on the basis of estimated income. But income can be estimated on the basis of a fixed rate only. Thus, in a rate case it is utterly impossible, logically, to arrive at a valuation that has significance. The courts have tried in vain to escape from this difficulty by using vague, undefined words to qualify the word "value," such as "fair value" and the like. These words interpreted apart from the speculative past mean simply that the company is entitled, in view of the unregulated results of bygone years, to what the court, under all the circumstances, regards as an adequate reward for the service and capital the owners have wisely and efficiently contributed.

Every tyro knows that this has no fixed, permanε relation to value in an economic sense. The phrase mean nothing else than exchange value based on attempt to read anything else into it is misleading. value," as used, leads, always has led, and always fusion. It is responsible for the introduction of the visionary theories by engineers, lawyers and accou important rate case. All these theories come from motive, that is, the desire and effort of the comp interests. But "to protect one's interests" in sucl tect the results of past speculation with this prope same extent as if it were strictly private and had ne by the law a public utility. Such theories have n or application to future expenditures on capital ac in all civilized states, hereafter to be capitalized at p basis for fixing rates on that investment for all tim

COST-OF-REPRODUCTION THEORY OF VAL

52 For a decade or more and until recently th reproduction actually dominated courts and commiss the last few years the courts are demanding more a of investment. They are not so much concerned what the property would have been worth if it ha remain private property and to enjoy its present 1 unrestrained, or unregulated. For to suppose su contrary to all the facts and the law. The prope property and it has not the right to make, or even t speculative monopoly gains that it could make as u property. Hence it has not the same value.

53 The courts fell into this trap because they and social knowledge, were over burdened with worl public side was never adequately presented to th conditions they were controlled in their judgment private property and decided, as if they were ign that, if this were private property, subject only to as private property generally has imposed upon it the Court would never have come before it at all.

54 Urged on by the companies, under the stress the courts have, in fact, depended in general upon cost of reproduction, trimming down the estim extreme cases to fit their own sense of justice. So fa

strictly applied, it gives the companies all the speculative gains of unregulated monopoly.

55 Cost of reproduction is merely an attempt to value the property as if it were private property. But, private property legally and customarily exacts the highest possible price — the pound of flesh — and we are here dealing with a monopoly. Furthermore, the sole object of calling it a monopoly — or public utility, which economically is the same thing — was to prevent such charges. But you say this is destroying value and confiscating property, as Justice Field said in his dissenting opinion in Munn *v.* Illinois.

56 Very true, but to destroy such part of profit, and consequently value in these industries as rests on monopoly and permits extortion, is the very object and purpose of regulation. Regulation is meant, as the late Mr. Eshelman[1] so well said, to substitute what the company *ought* to take, for what it would take, if it had the same monopoly free from regulation. But what it ought to take has no relation whatever to value, but depends upon sacrifice, effort, or cost, on the part of the company. So on the cost-of-reproduction theory, the company, in essence, is trying to put itself, by inflating values, back to where it can make a fair return on a strictly monopoly value, which monopoly value supposes the presence of monopoly accompanied by the absence of regulation. It is needless to call attention here to the fact that unregulated private ownership of these industries is unthinkable, and that the very suggestion of such a thing finds no place under our present commission laws.

57 The theories of valuation pressed upon regulating bodies, at such expense, and with such vehemence by the hired experts of the companies claim a value on the cost of reproduction as high as the public would be willing to give rather than to deprive itself of the service and undergo all the inconveniences of going without it until it could build a duplicate plant. The companies always forget that there are two sides to this, namely, that the public, in the absence of specific contract, has a right to duplicate the plant or to license someone else to do so; and that, if such duplication should actually take place, the value of the old plant would be marvellously lessened, or totally destroyed. To claim value analogous to that of private property, therefore, leads to a confusion of thought. For what the companies are really contending for is a value independent of cost, a strict monopoly value on the basis of unregulated private-property rights. They do not seem to remember how far the Court has gone in

[1] The Utilities Magazine, January 1916, pp. 5–7.

limiting and restricting the right of contract whe
is concerned, or monopoly is involved, as in th
utilities when the case is adequately presented to

58 Perhaps this point deserves further emp
tley[1] case the railroad company entered into a
1871, which the Supreme Court later declared
was entered into, to grant Mottley and his wife du
natural lives free passes over the railroad in co
sonal injuries received on the road by Mottley.
to issue the passes after the amendment, in 19(
Commerce Act (passed 35 years after the contract
this amendment forbade the railroad to receive a
for transportation, except in the cases provide
Supreme Court, February 11, 1911, sustained th
great emphasis said the contract, although leg
entered into in view of the right of Congress to
commerce, and that the risk of such regulation
contract, that is, that the right to regulate was
right to contract.

59 This seems to show that all property is su
and I repeat that the object of regulating utiliti
monopoly part of property devoted to a public u

60 The same principle is embodied in the de
fornia Railroad Commission. In this case[2] (Eshel
the commission found certain contracts, emb
crimination for the supply of water to private co
these contracts were entered into before the co
ment, 1911, authorizing the establishment of tl
necessarily, before the statute creating the com
mission voided these contracts and the case wa
company did not appeal the case.

61 The commission, after saying that the
been both proper and legal when entered into, rei
time, when changing conditions bring about the n
in rate, the commission should exercise its und
depart from the conditions of any such contra
such contract might have been in its inception."
is property and Justice Field was right when he s

[1] L. & N. R. Co. *v.* Mottley, Sup. Ct. Rep. 31, p. 26
[2] 2 Cal. R. C. R. 464 *in re* Water Rates and Servic
Diego.

of property, or the profits of the same, in whole or in part, is to take the property. But that is the essence of regulation of monopoly, in public utilities; to take that part of the possible profits which is considered unjust. It is exactly this part of the property, and of the profits, that competition would wipe out if competition prevailed, and it is the absence of competition that gives occasion for regulation.

62 The fact that in a majority of the older and stronger railroads and many other utilities the original water has been squeezed out by the investment of surplus earnings in the plant, does not simplify the problem of valuation but merely adds complications to it. It emphasizes all the more the speculative elements in the situation. In the earlier days, the companies claimed a return on all their watered stock, and opposed regulation that interfered with this claim. To this the Court answered with its "fair value." The effort of the companies was to inflate the valuation so as to bring about the results that the Court denied them in the case of Smythe *v.* Ames. Meantime, social growth was very rapid and regulation very lax. Therefore, when prosperity came, surplus was piled into the plants in addition to a fair dividend, and under the doctrine of valuation and freedom of contract the companies with such surpluses based their claim to these same old speculative gains — gains which may legally, if not wisely, be permitted to go to private property, but which the law of public utilities tried to prevent from coming into being, and, by its doctrine of reasonable rates, tries to claim a share in after they have been created. I am not, at this point, passing judgment on the question whether these speculative gains ought to be prohibited or not. I am simply trying to show that the idea of monopolies regulated by law, under the doctrine of public utilities, leaves no place for them. The main object of classifying any industry as a public utility is to prevent this very thing. But valuation under the cost-of-reproduction and similar methods now employed prevents the accomplishment of this object through regulation.

63 The closer we look into this point the more we are brought back to my main thesis, namely, that all our trouble with regulation comes from past speculation and not from an inability to agree about what is fair and just, so far as entirely new utilities and future investment in old ones are concerned. For it is admitted by all parties that, so far as the future investments are concerned, it is the right as well as the duty of the commissions to hold the capitalization of these new investments down to par, and to fix a rate that will give no more

than a fair return on such capitalization, irresp
values caused by social growth.

RESULTS OF CURRENT VALUATION MIS

64 The valuation of all the railroads now in
current valuations in rate cases, are worth more t
as a means of educating the public. They m
profitable as a basis for compromising with the
past efforts. But they do not arrive at value
market sense. They are misleading. For the p
on the theory of the cost of reproduction, the n
arises we must go all through the process again
speculative elements due to fluctuating prices, soc
increment, surplus, and the like. For if the real
always increases, in these utilities with social nee
On the other hand, if we do not mean value the
solely to the sacrifice the owners have made, a
whatever to value in an unregulated monopoly;
on the public's need and ability to pay.

65 Indeed, these valuations will not serve as
as a basis for future valuations and a means of la
future valuations for all time to come, but as a g
of vast claims based on past action. The valua
also in pointing out the method of compromise fo
utilities not covered by the federal valuation.

66 It may not be out of place to call attenti
several relatively recent street car franchises[1] ba
principle of cost of service. These franchises are
not only defective, but vitally defective. But the
have cost, and all they will cost in the future becau
tion to the particular point under discussion. T
once and for all, from these systems the bugbear o
future the amount on which the companies are en
the actual investment, or what is accepted as su

67 These franchises are meant to give a fair r
on public-utility investments, made on a monop
non-speculative conditions, and publicly controll

68 Unless an attempt at regulating capitaliza
should prove as futile as earlier attempts at prev

[1] I refer more particularly to the so-called street car
Cleveland and Kansas City.

ing have heretofore proved, investments made from now on will need no valuation whatever, in these cities or elsewhere in rate cases. Furthermore, it seems plain that all commission laws will, in the future, provide for controlling investments and capitalization. It is not too much to hope and expect that the Interstate Commerce Commission will, at an early date, be given power over these matters.

69 We all agree that the owners of utility property should have the firm hope held out to them of financial returns commensurate with the money and effort they contribute to the public service, always taking the risk into view. Now, if we could forget the entanglement of our jurisprudence and the chaos of conflicting testimony in rate cases, and really determine how much the owners have put in, I ask, if any conjectural measurement, any mere guess, any pure assumptions, can furnish as an effective or rational means of determining what the owners under this modified form of monopolistic, regulated private property are entitled to, as an actual record of the contributions they have made in time and money to the public service. These franchises rest upon the belief that original cost, in money, including services, is the only sound basis of valuation.

70 If this is not sound doctrine why has every reputable act and every act that authorizes valuation attempted to put all future investments on this basis? Are these acts all based on a false theory? Ought we to make no effort for the future to hold capitalization down to actual investment?

71 This is what the new model franchises call for. It is what sound regulation means. It still leaves the vexed question of pioneering, past wrongs and extravagant ventures to be settled by compromise, not by determining value in any proper or economic sense of the word. As previously stated, this is a matter of public policy, to be determined not by experts but ultimately by public opinion.

72 Any other method of trying to determine what is fair to the owners by valuation, or otherwise, plunges us at once into the question of how to deal with the results of past speculation, good, bad, and indifferent. How ought the punishments and the rewards of these past reckless, unregulated conditions of the competitive era to be apportioned between the owners and the public? What is a reasonable notice of a profound change of policy on the part of the Government? There can be no doubt of the legal right of a government to change its policy and as little doubt of the fact that by creating the class of public utilities and making them subject to such regulation as the

law calls for, we meant to change the relation o
government, and, to take away the chance of specu
fair then to enforce these laws strictly, or should we
adjusting the old conditions, to the changed ideas
In other words, when ought we really to begin to e
underlying regulation? I cannot undertake to a
tions at this time, but content myself with remar
allowed the owners in their attempts to safegua
interests through valuations, to make control of
practically impossible, while contributing almo
settlement, under a compromise agreement, for th

THE SURPLUS PROBLEM

73 If we could imagine all public-utility prop
to be wiped out by a miracle — an assumption not
theories of the cost of reproduction — and all oth
as they are; namely, knowledge and needs, does a
any question of valuation, or "fair value," woul
Under the present form of commission control the
simply see that the money to rebuild the system
talized, and the owners would agree that their re
based on this amount and no other. Everyone
raise any question of the value of this property
legal relation to the amount on which the company
income. No one would feel warranted in asking
rates so long as present rates gave him a fair retu
invested, simply because of social growth, increase
the multiplying of land values, or unearned incren
one suggest that valuation had anything to do wit
such circumstances? Would it not seem as irrati
increase of rates because of such increased valuati
a state-owned railroad to increase railroad rates
minals and right of way increased in value, becaus
As a matter of fact the public would look upon su
absurd, if not insane, and would properly say th
value had no possible relation to rates, and, that,
stable rates. And it is in fact stable and equal rat
low rates or low profits at which a sound regulatio
claims of the public to a share of the surplus, the
take charge of that at its source and, under its c

ments, see to it that only such future surplus ever came into being as was really needed to protect the credit of the company and to carry on the business effectively. We should have no such a condition as presents itself today in the Union Pacific Railroad. The late Mr. Harriman, beginning more than a decade after the United States had undertaken to regulate railroads under the Interstate Commerce Act, used that property for the most gigantic speculation. As a result of this successful speculation that road now, according to the *Wall Street Journal*,[1] holds securities of other roads to the amount of $175,819,947; and in addition, a surplus in cash, and cash items, of $35,000,000. If the road never did any more transportation business or utilized any of its transportation property for any purpose it could, from its other investments, pay a good dividend on its common stock in perpetuity. What relation has all this property to a fair rate? What is a fair valuation? What property ought to be included in a valuation of the property of this system?

74 Until the rights in this surplus are determined by court or compromise agreement, valuation means nothing, and regulation is a farce. The same kind of a condition prevailed in the Great Northern Railroad until the Northern Ore certificates were distributed as a sort of stock dividend. But the ore property was bought with the money of the company, originally contributed by others than the owners, and after the act to regulate commerce was passed.

75 The *Wall Street Journal* says of the Union Pacific: "The net surplus available for dividends is $125,000,000" and remarks that "Undoubtedly more or less of this surplus will be so distributed eventually"[2] but states that part of the surplus is probably being held to use in buying up the Central Pacific, if that road should, by court order, be taken from the Southern Pacific. This journal makes the following comment on the situation: "It does little good to remind government, and commissions, and labor organizations that Union Pacific is earning only 7 per cent or 6 per cent or 5 per cent on its investment from railroad operation, and that the balance comes from its operations as a banker, using wealth not earned in the railroad field. What labor and commissions and administrative officials

[1] *Wall Street Journal*, Oct. 7, 1916.

[2] It was to prevent just such speculations as that, out of which this surplus came that regulation was established. It would have been "fair" to prevent the surplus. What it is "fair" to do with it nobody knows: but it will not be determined by valuation, nor even by "fair value," but by a compromise far removed from value in a proper use of that word, and by the strength of parties in bargaining. Just as real value is determined in the case of unregulated property.

remember is that Union Pacific last year had a to
per cent for common stock, and it would take
patience and Solomon's wisdom to convince the pu
an income Union Pacific is entitled to consideratio
railroad rates." Is it not to stop just such specul
that all attempts at regulation are set up, and is it n
companies to get the full benefits of such success
the past, that causes all the trouble in fixing a fai

76 Bankers within the last generation, with al
traditions, dragged their institutions into the s
corporation and trust promotion. They became
promoting public utilities, particularly railroads.
of the spirit of socialization, and the coming of
of utilities, the bankers are again growing conserv
be really convinced that the degree of speculati
mitted in private property, and in public utilities
permissible or desirable. Witness the call of the W
(not hostile either to corporations or to large i
made in speculation on the exchange) crying out
of the New York Clearing House banks to preve
tors from buying control of an old New York ba
there is more than a sentimental objection, when a
established clientele falls into the hands of inter
from outside' . . . . . It would seem that the
ing House authorities goes further than to superv
its members. They owe it to the whole bankin
vent, if possible, a time-honored institution from
sponsible hands. Such an incident tends to unde
Just so the falling into irresponsible, greedy and re
railroads and other utilities with the inevitable sp
lowed led to public regulation of utilities. It cann
that the chief object was to prevent speculative p
tive values out of which such profits could be made
these.

PUBLIC UTILITIES SHOULD BE GUARANTEED AGA
LOSSES

77 I wish to discuss most briefly one other p
of losses under regulation.

78 It will be said that the doctrine I have s

[1] *Wall Street Journal*, Sept. 13, 1916.

succeed, would at least require a guarantee of all losses. To this it may be said that probably any system of effective regulation of socially important monopolies calls logically for a guarantee by the public of all losses necessarily incurred. Many people of various points of view admit this. It might be more advantageous to do openly and directly what we now do vaguely, indirectly and occultly, that is, to guarantee all losses by contract, rather than to inflate values under the heads of "going concern," "cost of developing the business," and the like, to cover these losses. Not the least of the evils of repaying losses under the system of valuations, is that the process is so indirect that the public does not know what is being done, and that the methods by which the process is accomplished opens the gate for the capitalization of these losses. The fact of capitalization, alone, introduces endless confusion, and prevents effective regulation.

CONCLUSION

79 In conclusion, we have seen that the right to class certain industries as public utilities and limit their profits is justified by the law and necessary for the public good; and that the reason for such a distinction among enterprises is that the public utilities, while necessary for the maintenance of our present civilization, are monopolistic. Therefore, in the absence of regulation, there would be no limit to the charges or profits except the ability of the public to pay. Under such circumstances, the value, in the ordinary sense of that word, would increase with the profits, for value rests on bargaining power. The essence of this bargaining power is the element of contingent gains and speculative conjunctures. On account of the importance of these industries to the public welfare, states undertake to destroy this speculative chance and risk and to prevent extortion and abuse by confining the utilities to a fair and steady rate under regulation. This, while not the only motive for regulation, is the chief one. But the fact of limiting the possible profits of an unregulated monopoly of this kind removes the enterprise at once from the realm of value by destroying those attributes of private property out of which value grows. At the same time, regulation destroys the customary means of measuring value, even if it existed in a utility. For value is measured by earning power. But the earning power depends on the rate of charge, and this is fixed by law and not by the economic forces and bargaining power that fix value in the only proper meaning of that term. Therefore, to attempt to discover value as a means of

fixing a rate of such a monopoly by public aut
economist knows and every court says, to reason

80 We have been led into this slough of des
over who ought to have the gains and who ough
of the era of competition. If that problem could
for all, there would remain no controversy ove
amount so agreed upon by the parties, plus any
by the owners, that is, the investment, is the just
the owners of public utilities should be allowed t
commensurate with their efforts and the prevailin
present commission system of regulation, with its
ments, capitalization, and accounts, provides full
nation of the investment and there is now no poss
sion for valuation for rate making under such an
as the controversy over the disposition of the spe
unregulated past and the uncertainty of the rec
make it necessary as a basis for compromise.

DISCUSSION

H. L. Gantt said that things had so chang
the last two years that a great deal that Prof
thought, was out of date. It might be good
and proper, according to our viewpoint of a
we were thinking in different terms today. V
with England, and had not learned from the wa
but if ever confronted with the same problem,
come through the men who knew how to do thi
of our problems would be in the hands of our en
our financiers.

The paper related to a condition of affairs
was the supreme power in the world. Today, ho
was successfully disputing that power with h
countries, and no theory of valuation would l
future unless it took account of his value, wh
had left entirely out of his calculation.

Charles Whiting Baker thought it had bee
travelers, the people who bought gas and electr
hind the great movement that demanded public

regulated, rather than innocent purchasers of worthless securities. The cost-of-reproduction theory of valuation seemed to him to lie at the base of our whole work of valuation. The earning power of a manufacturing property would vary from year to year, and every competent appraiser knew that the only safe and fair way was to value it at what it would cost to duplicate the plant as it originally stood. Why? Because a competitor might come along the next day and duplicate the property and go into the business.

L. K. FRANK[1] wrote that it was not improbable that industrial property might be subject to public regulation of return on investment before the close of the century. The Federal Trade Commission was advocating the adoption of accounting methods by the manufacturers of the country, which in many instances would call for valuation work, and upon the care and industrial statesmanship of the engineers involved would depend the future course of this movement for social control of industry. Social progress was inevitable, and it was open to the engineers to play a part in this movement, the importance of which could scarcely be measured.

M. E. COOLEY said that he felt a degree of responsibility concerning the use of the word "depreciation" in connection with appraisals. It came up in connection with the appraisal of the Michigan railroad properties in 1900, and had come to play a very important part in the decisions of the various state commissions since organized; an unfortunate part, he might add, for it was not entitled to have the use that had been given it, and should be discarded in certain classes of investigations. For example, in the case of one of the old plants of a certain complex electric light and railroad property, the value of the machinery to be discarded was estimated — by observation — at $180,000. But an investigation which considered its distance from the other plants, the necessity for constructing transmission lines, and the necessity for wiping out in a sinking fund or otherwise whatever value there might be in the old plant, showed the commercial value of the old plant to be about $360,000.

The word "depreciation" had no place whatever in the investigation of a rate case. For example, a railroad was made up of a vast number of elements, all new in the beginning and each having, say, 100 per cent value. After a few years the elements — ties,

[1] 15 Dey Street, New York.

rails, locomotives, cars — wore in various degrees, After, say, twenty-five years this property woul condition and would continue to exist in perpet less than 100 per cent for each of the elements might be expressed, say, by 80 or 85 per cent; tion it would render the maximum service whicl rendering. But there was 100 per cent investm and we should accordingly allow in a rate-makin the 15 or 20 per cent depreciation having no p the problem.

The word, however, might have value in where, for instance, a sinking fund was provid difference between the 100 per cent and the 80 that case we should perhaps take the 80 or 85 p making purposes rather than the 100 per cent.

Morris Knowles said that he differed fron respect, and that was that valuation in amount r on the purpose. The report of the recent case electric situation showed that if the amount could b for rate-making purposes, this value should be use Certainly in the case of a purchaser this value wc he would not want to pay any more for the pro thought he might get out of it if he were subject

Referring to Dean Cooley's remarks, he thou ciation fund was in existence, then the prope earn on 100 per cent; but if the utility had po would not be carrying out its trusteeship to the ciation should therefore be deducted.

William Kent stated that there were so where he thought the cost-of-reproduction theo very well. For example, what was the value of the Interborough Rapid Transit Company in 1 owners decided to throw out engines that were spend $2,000,000 to replace them by turbines not the cost of reproducing the plant going in Some allowance for valuation would have to be duction basis.

We had not arrived at any proper theory o we had no leaders of thought on this subject public. Some day, however, we would get authoritative statements from engineers rather than from lawyers. It was a question of strong common sense rather than of the traditions of law.

ALAN E. FLOWERS wrote that no one seemed to question the justness of allowing future returns on the valuation agreed on plus additional investments, or the necessity of charging in the rate an amount sufficient to offset depreciation. For the future, then, an amount for depreciation would be subtracted from the capital account only when a unit was withdrawn, and at the same time the capital account would have added to it the cost of the replacement, the latter being taken from the depreciation reserve.

Professor Gray assumed that the utilities, although subject to regulation, were free from competition. The facts were that many utilities were subjected at the same time to competition from commercial rivals and from municipal plants, and in addition had been put to the expense of inventory and appraisal hearings before state commissions.

If this depreciation reserve were invested in extensions, the ownership both legally and justly would be vested in the stockholders and not in the consumers; and if these extensions increased the output the operating expense therefore should be paid by those using it, and they should also pay a return on the investment needed to give the additional output.

He thought that the practice of commissions requiring the filing of installation estimates and costs gave a basis which made reappraisals unnecessary, so that the author's fear of future additions to capital account by future appraisals which included values due to social growth seemed to be unfounded.

Professor Gray suggested that commissions might guarantee returns, ignoring the repeated declarations of courts that no such guarantees could be legally made.

ROBERT L. HALE[1] wrote that, unfortunately, people had been permitted to invest in public-utility companies with the expectation of getting more than was necessary to induce investment. As a result, justice might compel us to permit these unwarned investors to continue in the enjoyment of the returns for which they had paid, but this did not prevent us from adopting a policy of warning future investors as to just how much they would be permitted to

[1] Instructor in Economics, Columbia University, New York.

earn, and of preventing the value of existing
coming more swollen than it now was.

In regard to past investors he thought so:
unsatisfactory compromise necessary. If th
they were entitled to a fair return on the ex
entire business, then no reduction in net earn
But if they meant to permit a fair return on t
the "property," and meant to distinguish the
from that of the entire business, then the c
physical property with an equally efficient sul
light in some cases on the value of the prop
from that of the entire business.

When it had been decided how much return
to any public utility, there was no reason why
that return should go to it for accidental r
seemed best to permit a rate which would be
sult in a return as great as that which had be
missible, and then to hold the company resp
back into the public treasury, or by putting it
without adding to the amount on which it sho
the future.

R. B. SHEPARD, JR.[1] wrote that the import
costs of reproduction arose from the fact that
position open to public utilities in the protect
seemed to be that confiscation resulted when ear
by arbitrary action of regulating agencies tha
capital employed were denied, and the cost of re
afford the readiest measure of the sum of capit
lent, engaged in the service.

The term "cost of reproduction" should, i
terpreted to mean the total investment which
replace the object under consideration, includ
development costs, etc.; that was, the absolute
terms of money, which would be required for
in its entirety, of the completed and operating
somewhat more comprehensive definition tha
tached to the term, but it was justified by th
of the estimated cost of replacement.

The true measure of "value" was said to be

[1] Office Engineer, Atlantic Coast Line R. R. Co., W

of the business, and the true criterion by which to determine the fairness of regulatory mandates, the extent to which the regulated utility was permitted to participate in the prevailing prosperity of tributary country.

THE AUTHOR, in closing the discussion, began by saying that anyone who thought that railway capitalization did not affect rates, should recall the traction history in New York and Chicago between 1899 and 1907. The things that started the movement toward regulation were, first, fluctuation in rates, and second, the enormous fortunes made by speculators.

As to the cost-of-reproduction theory, he did not accept it, first, because to accept it was to put the value up with the social growth, and therefore to give the benefit of the unregulated industries to the people who were fortunate enough to have bet satisfactorily upon the rapidity of the social growth. This was absolutely inconsistent with any regulation at all. In the second place, he did not accept it because, if he understood the development of human society, in the long run the latter would conform, not to the precedents of the law, as someone had said, but to the necessities of economic and social life. The cost of reproduction arrived at something that nobody who understood the situation and was honest wanted to find. If our social system was to be organized on the basis of mere speculation, we would go to destruction. It was safer to get down to the facts. He was opposed to the cost-of-reproduction theory because no one wanted to reproduce one of these important properties. In the next place, there was no human experience on which to base an intelligent estimate as to how long it would take to reproduce the property, or how much it would cost.

We all agreed that the public utilities were necessary for the maintenance of our civilization on its present basis. The company wanted its monopoly guaranteed, and wanted its monopoly profits. In the final analysis we must have the industries carried on, and if we allowed private capital to do it, we should allow a rate of income that in the long run would induce men to put in capital enough to meet the requirements of a growing civilization. This, however, had nothing whatever to do with value. It had to do with the contribution of money and services, and services could be measured in money. The State did not allow the service to be discontinued. When land went up in price the land was not sold. The land was dedicated to that particular purpose, and in many

utilities it had been acquired by the company
pose and limited to that purpose only. It had
in many instances, at least, as Mr. Baker had
of the right of eminent domain. If so acquire
was not controlling, nor was the right to regu
right of eminent domain.

Replying to a question by Clarence H. Tolr
capitalization of losses, he said that if we wer
company down on the present law to what wa
both the custom and the tendency were to mak
The tendency, undoubtedly, was to hold the con
perous times to an average rate of income. In
there would be losses, and in that case he would
ought to guarantee the losses, provided they
fraudulent, or grossly reckless, for if they wer
for in some way, private capital would cease t
service would suffer. The viciousness of the
proposition and of all our present methods c
tangibles was not that the utility ought not to
sation, but in the method of giving it. The c
in his judgment, had been the reason for one of
movements toward public ownership, and he a
knew that the present traditions and the organiza
ment were wholly inadequate, and the Governm
quated to manage business effectively; but he t
should be set up in an amortization account, an
utility should have temporarily a larger rate c
losses should not be allowed to disturb the capita
fuse the public mind as to the nature and conditi

No. 1576

THE RELATION BETWEEN PERPETUAL-INVENTORY VALUE AND APPRAISAL VALUE

By Charles Piez, Chicago, Ill.
Member of the Society

"What is your plant worth? You should know — *exactly*. You should know for insurance purposes, for financial purposes, for every purpose that has anything to do with the safe conduct of your business. You should know — *must* know — before you can calculate costs, overhead, profits; before you issue securities, make loans, place insurance. Your annual statement has a hollow foundation if its estimate of your assets as a going concern is based on the accountant's guess — a guess that has no better foundation than an estimate of costs at some past period, from which certain arbitrary percentages have been written off each year."

2 This statement, taken from a publication of one of the appraisal companies, can be accepted as sound without committing ourselves to the conclusion which the appraisal company is anxious to establish: that the real worth of a plant for all purposes can be established only through the work of professional appraisers.

WHERE APPRAISALS ARE OF VALUE

3 Most plants grow from small beginnings, and during their early life expand as the needs dictate. The organization is necessarily small, because the most rigid kind of economy must be practiced, and original costs and the costs of additions are frequently so completely submerged in the total assets that no safe records of these costs can be established. Many plants never outgrow this lump-sum treatment of assets. In these cases depreciation of plant

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

and equipment is either wholly disregarded, ‹
amount is charged against profits, the amount c
ing inversely as the size of the profit it is desire

4 The annual statements of plants so cond
hollow foundation, for not only their assets but
are based on the accountants' guesses.

5 Industries so managed need the assista
appraisal company to inform them of the value
basis for embarking on a sounder and safer s
methods. Practically all successful industries, a
of rule-of-thumb accounting, start with an ap
maintain their inventories that additions to the
recorded, and depreciation is based on judgm
But even such industries will find it to their adva
of appraisal companies for the purpose of deter
records of the cost of reproducing plant and equi
insurance or overinsurance may be avoided an
requirements of coinsurance complied with.

6 Appraisals are also valuable in estab
values of plants that are about to merge, or in se
scheme of financing. But the claims that an a
for figuring overhead costs and the selling price
factured articles, are, to say the least, sadly ove

7 It has been my experience that in well
appraisal values are usually considerably abo
values established by the management. With th
prices that has been going on almost continuously
five years, it is but reasonable to suppose that re
substantially higher than initial costs, and it is
amount of such excess if a proper insurance cove
such advance is wholly speculative and has abso
the value of the plant from the standpoint of the
management.

8 The establishment of true costs is an esse
success of a business, and true costs can only be ‹
item of expense is included. Buildings that we
and convenience of arrangement when they were
ally outgrow these initial advantages as the busin
spite of substantial increases shown in the ap
buildings, they have less value for the purposes to
than they had when they were first constructed.

prove so unsuitable to the increasing needs of the business as to justify demolition and reconstruction.

NEED FOR DETERMINING A PROPER RATE OF DEPRECIATION

9 Few owners are astute enough to foresee their needs for ten years to come, and fewer still have the means to build or expand along the lines that will give ample opportunity for future busin es growth. It is safer by far, therefore, to provide proper sinking funds through an ample rate of depreciation, so that when buildings that have outlived their usefulness require reconstruction, funds have been provided out of profits for rebuilding along more modern lines.

10 Machine tools have changed very considerably as a result of the development of the Taylor-White and other high-speed steels, and companies that followed appraisal methods of depreciation find themselves with obsolete equipment and no funds to replace it with modern equipment.

11 Patterns and small-tool equipment often have but temporary value and should disappear wholly from the inventory when they have served their purpose, yet these two items are fertile sources for inflation of values through appraisals.

12 What the management of an industry is chiefly concerned in, is to provide a fund through a proper scale of depreciation which will reimburse it for the difference between the cost price of a piece of equipment and its fair cash selling price when sold either because it is ready for the scrap heap or because some newer form or method has made a change desirable. This difference is properly a part of the cost of the product, but becomes so only by charging depreciation against the expenses of operation.

DEPRECIATION NOT PROPERLY DETERMINED BY APPRAISAL COMPANIES

13 Has any appraisal company ever investigated the subject of depreciation from the operating standpoint and recommended a schedule of depreciation for adoption? Has any appraisal company ever advocated that depreciation be distributed as an operating expense against the product? Can any appraisal company claim with any justice that it can determine proper rates of depreciation without close contact with and full knowledge of the operating conditions and operating needs of an industry? Certainly, without such contact and without such knowledge the claim that successive appraisals are

essential factors in the determination of costs, p
to say the least, pure buncombe. The primar
praisal company is to determine an authoritativ
and its entire organization is trained for this pur
appraisers enter the field of depreciation, operati
they are playing wholly out of their class and
They are doing their clients positive harm an
straight to the shoals of financial disaster; for
viously explained, have a distinct upward tenden
in value which they show as the result of wh
ditions have the effect of lulling the manufactur
of financial security.

14 All of those with whom I have been a
for the last quarter of a century have been radi
of depreciation, but with all of this strong leanin
be considered an excessive write-off, we frequen
ready to discard a tool or reconstruct a buildin
additional amount must be charged off to profit

15 The great majority of industries charge
than too much, and the appraisal companies, if
ing, unconsciously, of course, in increasing this
times disastrous habit.

16 I had occasion recently to go over the fi
a manufacturing plant which had delegated the
of depreciation to an appraisal company. The
annually was less than one-half of the proper am
the owner said, to the constant and considerabl
placement value of the property. Here was a
ducing the operating burden of a plant by credi
speculative and unrealizable increase in prop
case the appraisal company specified the amou
each year, and was therefore responsible for this
unscientific procedure. The owner is about to b
I take no chances in prophesying that he has so
ments awaiting him in unforseen shrinkages of a
dons the old plant.

17 The problem of determining an adequa
tion is by no means a simple one, and it goes ha
problem of distributing depreciation against the
It is astonishing to find how widely the prac
manufacturers in the same line varies.

STANDARD DEPRECIATION RATES ADOPTED BY MANUFACTURERS' COST CONFERENCE, FEB. 25, 1916

	Per Cent on Cost	Per Cent on Reducing Balance
BUILDING AND ACCESSORIES:		
Reinforced concrete or steel and tile	2	3
Brick and steel with non-combustible roof and concrete floors	2.5	4
Brick, steel and wood	3	5
Brick and wood	3	5
Steel frame, wooden roof and corrugated-iron walls	3.5	7
Steel frame, non-combustible roof and corrugated-iron walls	3	6
Concrete block, with wooden roofs and floors	3.5	8
All-wood structures, well built (20 years)	4.5	10
All-wood structures, cheap (20 years)	5	12
Sprinkler system (20 years)	4	7.5
Heating and ventilating system (20 years)	4	7.5
Water and sewer piping and sanitary fixtures (where separate)	4	7.5
Tanks and reservoirs, steel	4.5	10
Tanks and reservoirs, wood (10 years)	9	20
Note: All Repairs and Maintenance to be charged to Account 8059		
MACHINERY AND LARGE EQUIPMENT:		
Boilers, pumps, feedwater heaters and air compressors	6	15
Power piping	6	15
Switchboards, main wiring and conduit	6	15
Engines and dynamos	5	10
Machinery, motors, machine tools, traveling cranes, etc	4.5	10
Punch presses, bending rolls, power shears and drop hammers	4.5	10
Shafting, pulleys, hangers and belting	50	....
Machine-tool accessories—Boring bars, drivers, key-seating broaches, etc. (All renewals to Repairs)	50	....
Cupolas, converters, melting furnaces and accessories	5	10
Annealing and heating furnaces, ovens, forges, etc	5	10
Motor trucks	20	60
Storage-battery locomotives (battery renewals to repairs)	10	30
Horses and wagons	12	35
Steel shelving, lockers, etc	5	12
For items below a single write-off at the rates specified is made and the balance carried as a part of the inventory without further reduction. Only items actively used in fabricating standard product, and described in schedule as net items, should be so treated, all other items being charged off wholly to expense.		
SMALL TOOLS:		
For machines, net additions	50	....
Hand tools, net additions	50	....
Punches and dies (Standard), net additions	50	....
Chills, iron and steel flasks and accessories, net additions	50	....
FIXTURES, FURNITURE AND MISCELLANEOUS EQUIPMENT:		
Mechanical appliances, net additions	60	....
Departmental wiring and electric fixtures, net additions	60	....
Miscellaneous items (wood), net additions	70	....
PATTERNS (Standard):		
Metal, net additions	75	....
Wood, net additions	100	....
All patterns required for a particular order or contract to be charged to the job.		
DRAWINGS:		
All new standard drawings to be charged to expense.		
All drawings required for a particular order or contract to be charged to the job.		
MISCELLANEOUS REAL ESTATE IMPROVEMENTS:		
Pavements, sidewalks, fences, retaining walls, roadways, tracks, yard drainage, general conduits, tunnels, vaults, etc	4.5	10

PROPOSED STANDARD RATES OF DEPR

18 Largely as a result of the recommend Federal Trade Commission that the cost-acc various lines of industry be standardized, the m veyors and elevators have made a determined eff standardized form of accounting procedure. Th ing of the manufacturers and their accountant that, out of nine manufacturers, two disregarded preciation entirely, five charged off depreciation and only two charged depreciation against operati it thereby a component part of the cost. The r varied widely, and the first steps taken by the in determining a standard schedule of rates of d item of interest in connection with the subject o an incentive for other lines of industry to detern similar standards, I have given the schedule on The rates are but compromises growing out of th perience of the individual members of the conv conference, but their correctness can later be verif perpetual inventory values which these rates w the actual experience of loss in cash value when ings are discarded.

19 They establish, therefore, a broad bas operating values and the shrinkage in these valu tear and change in style, can be computed, and nary steps to the equitable distribution of thes preciation in value, over the cost of the product.

20 I recognize the value of the work don companies in establishing authoritative replacen poses of insurance, for the purpose of comparing of various plants about to be purchased or merge tion upon which a perpetual inventory kept by th be based; but I am convinced that only a perpetu ing for a sound schedule of depreciation, and inte the management of the industry, is of value in dete of the product.

DISCUSSION

Carl G. Barth, in answer to a query by A. E that if a machine worth $1000 was to be depr cent on the reducing balance, it meant a depr the first year, $90 for the second, $80 for the third, and the amount would never be entirely extinguished. On a flat rate of 10 per cent a year, however, its value would be wiped out in ten years. He was opposed to including interest on investment in cost calculations.

Harry Barker wrote that the fundamental term "depreciation" had become so involved in a maze of differing definitions, that even when engineers endeavored to use it quite technically it conveyed different ideas to different persons. In an attempt to harmonize some of these differences he had been led to study the various meanings attaching to the term, and had found that it was being employed in six different ways, each definition having different shades of meaning. (These definitions are given in The Journal, March, 1917, p. 215.) Specific terms were proposed, each of limited use, in place of the wide use of "depreciation."

L. S. Randolph stated that he had abandoned the use of the word "depreciation," and attacked problems in the beginning by assuming that a machine or plant could be so repaired or renewed as to do the work as well as when first erected. After that it was a question, first, of using the word "obsolescence," and then the term "productive life," or, in other words, what would be the productive life of a plant, and that, he thought, would have to be laid aside to return the value of the original investment of the plant when the productive life had ceased. He had come across cases where apparently the productive life was infinite, and where there was actually appreciation of value. Sometimes legal enactments would abruptly cut off the productive life of a plant.

Oberlin Smith said that his rule governing inventory work was that things were worth what it would cost to reproduce them in their present condition tomorrow if they were burned today. On everything valued we should first find the percentage of obsolescence and take off that percentage, but in a machine shop it did not pay to have that value too low, as the tools were not efficient unless they were kept up to 80 or 85 per cent of new value, so that they would do full work, and to rate them at that was evidently fair. The principle of constant depreciation might put them down to nothing, although of nearly full efficiency.

There were two kinds of obsolescence. One kind resulted from the tool not being up to date, so if it were likely that the tools would have to be altered or repaired, that should be taken into

account in the valuation. The other kind wa
in the demand for the product. Certain things
one season and not wanted thereafter, so we h
sideration the obsolescence of tools which in th
date and efficient, but for which there was no e

Some people depreciated machine tools ten
and that soon made them worth one-half or
original value, when they were as good as ever
to allow a certain small amount for depreciation
the tools in good order. If a tool was run down
or an additional part was applied to it, then i
the end of the year when repaired, and perhap
original value spent on it, and it was not right
it right along. It was worth as much as when
method of allowing for depreciation the invento
up and down.

Public utilities were an entirely different
things which had a value outside of the actua
plant sold out for 200 per cent of its cost, it
earning power, and this had to be taken into a
the plant for the market, or perhaps for taxing

Robert J. Hearne, in a written discussion
jects of inventory, cost and valuation were so i
should be treated together. Too many valu
were colored by future prospects. On the ot
not ignore present conditions. While abnorma
basis for valuation would arise with a change
had to be recent to be of any good.

In actual practice, extending over many ye
practical and necessary for the purpose of inven
estimate of actual cost, (2) a conservative valuat
ance, and (3) a valuation for insurance purpose
the work attempted by appraisal companies sho
own employees. At the best, outsiders could on
facts; they could not possibly know the busin
have a value, however, in detecting fraud and i

Properly kept, a perpetual inventory was
daily corrector of values and costs, and kept ev
It took some trouble to install and some though
it paid. He had had over 15 years' experience

was adapted to almost any business, that saved time, trouble and expense and that could be applied to a new business very easily and to an old one a little at a time. (The details of this method are given in THE JOURNAL, March, 1917, pp. 215–217.)

JOHN L. HARPER said that financiers often ask their engineers to present to them certain factors for determining the amounts to be written off each year, depending on the conditions existent at the time, and it was along that line he wished to inquire of the author as to his method of determining these factors in relation to the salvage values and the fluctuations in actual values of the materials covered in the several items.

THE AUTHOR. The table of depreciation rates in the paper offers the choice of two schedules, the first representing a definite percentage of depreciation on the original cost, and the second a percentage of depreciation computed on the reducing or depreciated balance. The two schedules are offered as substantial equivalents based on an assumed life of each class of items. The assumed life was determined by the members of the conference, and has behind it long experience fortified by actual records. The rates are conservative because it would be unfair to assess excessive depreciation against costs.

The rates provided under the per-cent-on-cost schedule extinguish the entire cost at the end of the assumed life, while the equivalent rates under the per-cent-on-reducing-balance schedule are supposed to bring the items to a scrap value at the end of the same period.

In the case of small tools, punches and dies, chills and flasks, fixtures and furniture and patterns, only the additions actually made for the purpose of fabricating standard product are to be inventoried: these are to be depreciated as indicated under the per-cent-on-cost schedule, and are thereupon to be subjected to no further depreciation. Care must be exercised particularly in these items that all replacements are charged to maintenance and all other obsolete items are charged off entirely.

These items rarely have much cash value upon sale or liquidation, and care must therefore be exercised to prevent inflation of values in their inventory. An occasional check by actual count, and a reappraisal of the value of the active items on the basis provided in the schedule of depreciations is strongly advised.

In order to compare the two schedules presented by the table, a condensed depreciation statement for a 34-in. boring mill costing

$1318 and purchased Jan. 1, 1894, developed
rate of depreciation of 4½ per cent on the original
ately below on a 10 per cent rate on the reducin
herewith:

INVENTORY VALUE AT END OF YEA

Depreciation	1894	1895	1904	1909
At 4½ per cent..........	$1258.69	$1199.38	$665.69	$369.04
At 10 per cent..........	1186.20	1067.58	413.60	244.23

The rates are those provided in the schedule
In the first example the amount of depreciati
each year; in the second the first year's depre
from the original cost and the second year's de
puted on the remainder, or, as it is termed in th
ducing balance.

The amount of the second year's write-off i
10 per cent than that of the first year, and the
each succeeding year continues to decrease by th
The original cost is never wholly extinguished b
the amount of depreciation thus written off eac
more nearly the shrinkage in value that actually
under normal conditions, loss in the selling val
equipment is more rapid in the early years of i
later years. Then, too, there is usually some s
value at the end of the period, and this scrap v
or smaller, depending upon the nature of the raw
the item is composed. It is for these reasons th
to the method of computing depreciation on the
and recommend the rates of the second schedule f

As pointed out in the paper, the method e
manufacturers of charging depreciation to the prof
is wrong, for while this method accomplishes the
the book values of assets in line with actual value
depreciation a part of the cost of production.
all equipment, jigs, templates, or patterns espe
particular order should be wholly charged to th
reduction in value of all other buildings and eq
mined by the schedule of depreciation, must b
legitimate expense of the business and charged

the product. The easiest method of accomplishing this is to estimate in advance the depreciation for each department of the plant for the ensuing year, and then assess one-twelfth of these estimates as monthly expense charges against the departments, making these depreciation charges in this wise components of the departmental expenses and factors in the departmental overheads. Depreciation charges that cannot properly be assessed against any particular department should be assessed against general expense and distributed over the product through the general expense factor.

Any differences between the estimated depreciation and the actual depreciation as revealed by the final inventory must of course be adjusted before closing the books.

Two methods of treating the inventory of buildings and equipment can be followed: First, the inventory can be carried at the net figure with depreciation deducted, or second, the inventory can be carried at the original value and a depreciation reserve account created which will be credited with the amount charged off each year. In the second case, the records of original costs are preserved, the total amount charged off to depreciation is always available, and the net value equal to the difference between the two is readily determinable. The second method commends itself, therefore, as a more complete record of actual procedure than the first.

No schedule of depreciation, no matter how carefully and intelligently framed, will cover all possible conditions and provide for all possible contingencies. A fixed schedule is always lacking in that elasticity which would render it universally applicable, and must on that account be supplemented in unusual cases by exceptional treatment born of intelligent analysis and experience.

In discussing allowances to be made out of profits to cover loss or shrinkage of value, it is well to bear in mind that relatively few manufacturers regularly make such allowances for depreciation, and that even these few have never adopted or even discussed any standard set of rates.

That loss of value from any one or all of the causes enumerated by Mr. Barker does occur is universally acknowledged. Academic definition of the generally used term "depreciation" does not stay its inroads on profits nor make the method of providing for these inroads any clearer. Why add such new terms as "retirance" and "renewance" to an already overburdened industrial vocabulary?

What is needed is a presentation of records and experiences, so that out of these we may develop a schedule of rates which can be

looked upon as a standard for determining annual
loss of value due to wear-and-age deterioration ar

The loss of value is legitimately a part of the
a fact which is at present overlooked in many b
and to correct this omission or oversight, it is t
more effective to start with a sane schedule of ra
than to start with a definition. I realize, of cou
ker's discussion pertains more particularly to the
lic-utility properties, where the allowances to be r
units may be so large as to justify division into
causes; but in the mechanical industries the ter
ciation" and "total depreciation" contracted fro
ciation allowance" and "total depreciation al
widely understood as to require no further definiti

What is needed is to have some association of t
prestige of this Society investigate this subject
schedule of rates of annual depreciation for adopti

With the administration proposing a further ta
on all profits in excess of eight per cent on the i
necessary that the engineering societies shall defin
ant a factor in the expense of production as dep
to annually and how provision for it shall be mad

Mr. Hearne has outlined very clearly a sys
which a perpetual inventory of work in process,
other assets of a manufacturing plant, can be intr
tained.

I am a strong advocate of the value of run
inventories for work in process and materials in
for buildings and equipment, and Mr. Hearne's di
because it outlines the possibilities in these d
valuation of small tools I believe it safer and near
assume that this equipment averages about half-
and worn out, and to charge off a flat depreciati
instead of a third. I believe his method of valui
at full cost is dangerous, and the same must be s
in respect to drawings.

I have never heard of purchasing drawings a
than a merely nominal sum upon the liquidation o
patterns in the average industry are subject to
terioration in value through obsolescence as to jus
of depreciation named in the schedule.

No. 1577

VALUATION OF INDUSTRIAL PROPERTIES VERSUS VALUATION OF INDUSTRIAL METHODS

By Walter N. Polakov, New York, N. Y.
Member of the Society

The question of determining the value of the physical properties of our industrial establishments and public utilities has been recently brought to public attention. The purpose was primarily to justify the increased cost of commodities, rates and transportation as gravely influenced by heavy investments in the machinery of production. Discussing the subject at the conference on valuation held in Philadelphia in November, 1915, the writer said: "The loss to the companies from the undervaluation of their property is insignificant in comparison to the actual losses due to the lack of proper operating methods. The subject of correct methods of appraising a property is completely overshadowed by the importance of determining the proper methods of using this property. . . . What is needed most is the correct valuation of operating and managerial methods in vogue, and overestimating the importance of property valuation is like trying to trace old sins instead of preventing the committing of new ones."

2 It is a fact, that the fixed portion of the capital invested in real estate and equipment has materially increased since the tractor replaced the ox plow; modern factory equipment has taken the place of artisans' tools and transportation is done by rail and automobiles instead of horses. The correspondingly increased financial burden has created a new problem: how to absorb the expense of wear and tear, obligations to money lenders, etc., increased by adopting expensive machinery of production as a result of engineering progress.

Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

3 The solution offered by financiers and accou
school was to include in the manufacturing cost all
of a plant and all the financial obligations of a com
of whether these investments or expenses contrib
the value of the commodity or in any way benefite
If the instruments of production were completel
process this theory would be correct. However,
the value of the instruments of production passe
the cost of commodities as they are produced. Ap
evident fact that the equipment not used in produ
any part of its value into the product was usual
consideration that it loses value during idleness a
cumulates such charges as rent, interest, insurance

4 The absurdity of including these expenses
cost of an article already bearing its share of fixe
clearly exposed at the Buffalo meeting of this Societ
Mem.Am.Soc.M.E., that any further argument s
Present practice manifestly tends to charge against th
the cost of overequipment and mismanagement, thus i
the penalty for tolerating these conditions in the indu

5 The paradox that adopting improved mac
designers reduces the rate of profit of the enterprise
either by the fact that the improvement is merely i
improper use is made of the improvements. No
would approve a replacement or new installation
less study proved beyond any doubt that the adva
such a change are ample to more than pay for the in
the cost of production increases after an investmen
cates plainly either that it was poor judgment to
or that the method of management does not offe
to secure the advantages inherent in the new impr

6 Nevertheless, the tendency of the average ra
is only a manifestation of the development of the p
of society. Automatization, growing speed and
chinery cause the same number of laborers in t
convert a larger quantity of raw and auxiliary mate
In other words, the fall of the rate of profit is not
but only to a relative increase of the capital inv
production over its part invested in labor. This
dom recognized, is actually followed in our indust

[1] Trans.Am.Soc.M.E., vol. 37, p. 109.

It requires an increasing mass of total capital for the purpose of using the same quantity of labor to secure the same mass of profit though at a falling rate of profit. If the mass of profit is to be increased through the introduction of improved machinery (*i.e.* with corresponding increase of investment) without curtailing the wages or lengthening the working day, the intensity of labor must be increased.

7 In other words, to keep pace with industrial progress in order to meet the requirements and competition, improved instruments of production were adopted. Increase of investments is followed necessarily by reduction of the rate of profit, and in order to secure the same or larger earnings the manufacturer must expand the production, ergo, increase the investments. This is the process we actually observe in our industrial development. No amount of pains given to the subject of proper appraisal of physical property can alter the tendency for the rate of profit to fall. Salvation is sought therefore through raising the commercial profits by increasing the selling prices. This method, although temporarily expedient, overtaxes eventually the purchasing capacity of consumers and creates a so-called business depression.

8 At the outbreak of this war, our country was rushing headlong into a period of industrial depression caused by surplus production that could not be consumed in the United States on account of the small purchasing capacity of the people, and could not be exported because poor shop methods prevented us from producing as cheaply as other countries. The undeveloped stage of our credit system contributed a further obstacle to the development of export. There was no incentive for us to plunge into the war, since there were no foreign markets to fight for; neither could anything be gained by a territorial expansion. A crisis seemed imminent, but the loading up of our industries with European orders postponed it.

9 Similar circumstances in Europe culminated in the present great war when German methods of production took away foreign markets from the countries whose workers could not consume at home all they produced. If it were not for the war between the nations, a war within the nations might have been imminent. In the course of this struggle the most amazing reshaping of principles underlying the industrial life is taking place. Various governments have been forced to admit that the industries should serve the country, not the individuals. Handicapped by unscientific, disorganized production of the most necessary commodities, they have

come to realize that the right to control and direct production involves the responsibility of applying proper methods to secure beneficial results. Furthermore, precedents have been established where those who were not carrying on their business for the common good were denied the privilege of running it at all.

10 The misuse of or failure to use the expensive machinery of production is not a sufficient reason to either advance prices or to lower wages, as both undermine our future. Neither is it necessary. A manufacturer adopting superior methods of planning and managing his business, working with improved methods of production that have not yet become general, can sell below the market price but above his individual price of production. In this way his rate of profit rises until competition levels it down. During this period the second requisite comes into play — the expansion of the invested capital, facilitated by the previous period of accumulation. According to the degree of his expansion he will be enabled to employ all of his workmen or more; in other words, he will be enabled to produce the same or a greater mass of profits. As long as industrial development along these lines is still possible, and this possibility is accentuated not only by European experience but by our numerous experiments and successful demonstrations of the adaptation of scientific principles to the management of various industries, the revision of our methods is a paramount problem. Therefore the problem of determining the real value of our industrial and managerial methods, if properly solved, offers the opportunity of rectifying the errors of the past.

11 The appraisal of industrial property when accurately made, discloses the fact otherwise overlooked that, as a rule, there are more means of production than are made use of and the idle capital investments represent also idle labor. *The revenue-producing factor in our mode of production is not the investment but the method of its use.* It is the business of financiers to invest money in industries and they want to know what this investment is and what return they are getting. Ultimately the question they will ask is: *What is the value of the methods and what do they lead to?*

12 This statement does not imply suggestion to appraise property according to the capitalization of the earnings. The use value of the plant has nothing to do with its cost value and is in the highest degree unfit for the purpose, as it varies with personnel, policy and methods. Evolution in our industrial relations will eventually reverse the present situation, and all charges for unprofitable invest-

ments and non-productive forces will not be borne by the consumers, who as yet have no power nor means to take an active part in the management of the industries. Unfit methods and incapable leaders are equally harmful to both investors and consumers, and their interests unite in demanding that methods of the highest value be developed and put into use.

In presenting his paper the author said:

The ancient Chinese custom was for a bridegroom to pay ransom in proportion to the bride's weight. Modern financiers seem to have difficulty in getting rid of a similarly senseless habit of valuing their plants on the basis of the equipment they contain, irrespective of whether it is used to good advantage or not.

The time has come to realize that if our means of production are improperly used or not used at all, they are a liability, not an asset. Yet we still tax the production and protect the idleness and mismanagement, worrying by the force of habit about the valuation of properties and passing lightly on the valuation of methods.

Among the warring nations the cry "Idle hands assist the enemy" became a slogan, and we are gradually forced to recognize the significance of it.

The other day my advice was sought as to what additional equipment was needed to increase the output of a mill from 13 to 40 tons per day. Careful inspection showed that with the present equipment the mill could produce at least 60 tons daily if only the management would not make a mess out of routing of material and handling the men.

Examples of this sort can be multiplied at will and they all prove but one thing — that the methods of production have a real value, while the value of the means of production is a fetish.

Our present methods are not such that we should boast about them. Our munitions manufacturers, farmers and railroaders furnish ample proofs of that.

The coming of a new industrial era exposes our economic unsoundness, in that commodities are dear and men's work is cheap. In the new order of things *deeds* and not the *possession* of things will count.

The war is teaching us many lessons, and the keynote of them all is that those who do not carry on their business for the common good shall be denied the privilege of running it at all.

DISCUSSION

William Kent thought that the whole qu ment of capital and the returns which should b distribution of profits, had got to come up a future, and he wished to offer an idea as to Suppose the Government should pass a law, whic by the community at large, that in capitalizing connection with the public — what might be utilities, there should be $1000 issued in bond capital stock, the latter being entitled to a six earned, the former to three per cent interest. A be allowed to earn, without any further investm meet extraordinary charges due to accidents, et to compete with concerns employing superior r cess earnings should be divided, one-half going and one-half to three different parties: the G sumers (by lowering the price or rates), and helped make the money.

W. S. Rogers referred to the statements in revenue-producing factor in our mode of prod vestment but the method of its use," and "Wh methods and what do they lead to?" Now, bankers in investigating the affairs of a compa had not only called for facts regarding the work that the methods used were the best. They w the value of the methods and Mr. Gantt in hi foundation in detail to lead up to that.

The Author. In closing the discussion I de statement. Admitting that the method of use, and tools of production, is the revenue-produc nomic soundness of industry must have a bro *the production is carried out so as to make the m modity cheap.*

If we fail to accomplish either and both of the country either by producing things that ca by limiting the production on account of the low of consumers.

No. 1578

PRODUCTIVE CAPACITY A MEASURE OF VALUE OF AN INDUSTRIAL PROPERTY

By H. L. Gantt, New York, N.Y.
Member of the Society

Some months ago a professor of political economy in one of our most conservative universities admitted to me that the economists had been obliged to modify many of their views since the outbreak of the European war. My comment was, that the professors of political economy were not the only people who had been obliged to modify their economic and industrial views.

2 The war has taught everybody something. Military methods have undergone radical changes, but industrial methods are undergoing changes which promise to be even more radical than the military developments have been.

3 If there is any one thing which has been made clear by the war it is, that the most important asset which either a man or nation can have is the ABILITY TO DO THINGS. Our industrial and economic developments have in the past been largely based on the theory that the most important quality a man can possess is, his *ability to buy things;* but the war has distinctly shown that this quality is secondary to the *ability to do things.* The recognition of this fact is having a most far-reaching effect, for it makes clear that the real assets of a nation are properly equipped industries and men trained to operate them efficiently. The money which has been spent on an industrial property, whether it has been spent wisely or unwisely, and the amount of money needed to reproduce it are both secondary in importance to the *ability of that plant to accomplish the object for which it was constructed,* and hence cannot be given the first place in determining the value of the property.

4 Inasmuch as every industrial plant is built to produce some article of commerce at a cost which will enable it to compete with

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[1] Presented at the Annual Meeting, December 1916, of The American Society of Mechanical Engineers.

MILL, Textile June, 1916.

SYMBOL	DEPARTMENT OR MACH CLASS	% OF CAPACITY USED ON Day TURN	TOTAL EXPENSE OF IDLENESS	DETAILS OF IDLENESS EXPENSE DUE TO: LACK OF WORK	LACK OF HELP	LACK OF AND POOR MATERIAL	REPAIRS	POOR PLANNING	REMARKS
	Spinning		18 70	18 70					
	Winding		118 74		103 74		15 00		
	Doubling		10 61	10 61					
	Twisting		17 95	17 95					
	Quilling		20 67	10 67	10 00				
	Warping		390 75			390 75			Lack of Wound Yarn
	Weaving		915 25	75 00		840 25			Lack of Warps
	Finishing		210 72			210 72			Lack of Woven Goods
	Inspecting		49 70		10 70	39 00			Lack of Woven Goods
	Shipping		216 17	66 00		150 17			Lack of Woven Goods
	Total		1969 26	198 93	124 44	1630 89	15 00		

other producers, the value of a plant as a producing unit must depend upon its ability to accomplish the object for which it was created.

5 To determine the value of an industrial property, therefore, we must be able to know with accuracy the cost at which it can produce its product, and the amount it can produce. To compare two factories on this basis, their cost systems must be alike; for, if there is a lack of agreement as to methods of cost accounting, there will necessarily be a lack of agreement as to the estimated value of the properties. There are many methods of cost accounting; but there are only two leading theories as to what cost consists of. They are:

First, that the cost of an article must include all of the expense incurred in producing it, whether such expense actually contributed to the desired end or not.

Second, that the cost of an article should include only those expenses actually needed for its production, and any other expenses incurred by the producers for any reason whatever must be charged to some other account.

6 The first theory would charge the expense of maintaining in idleness that portion of a plant which was not in use to the cost of the product made in that portion of the plant which was in operation; while the second theory would demand that such expense be a deduction from profits. When plants are operated at their full capacity, both theories give the same cost. When, however, they are operated at less than their full capacity, the expense of carrying the idle machinery is, under the first theory, included in the cost of the product, making the cost greater; while under the second theory, this expense of idle machinery is carried in a separate account and deducted from the profits, leaving the cost constant. It is most interesting to note that, *when costs are figured on the second basis, great activity immediately ensues to determine why machinery is idle, and to see what can be done to put it in operation.* It is realized at once that this machinery had better be operated, even if no profits are obtained from its operation and only the expense, or even part of the expense, of maintaining that machinery is earned.

7 Fig. 1 illustrates this subject most clearly, and is an indication of the efficiency of the management as contrasted with that of the workmen, about which we hear so much. It is interesting to note

that charts of this nature, which are being mad
eral large plants, have already had a very educat
the managers of those plants. They show tha
which cannot be used should be disposed of, and th
and the space occupied, put to some useful purpos

8 A little consideration of the method of ge
this chart will make its value more apparent. I
growth of the paper I read at the Buffalo meetin
between Production and Costs,[1] and is based on th
ownership of a machine costs money, inasmuch
from available assets. For instance, if we buy a
we lose the interest on that $1000, say at 5 per c
we have taxes on the machine at 2 per cent and i
cent. Further, the machine probably depreciate
per cent per year, and we must pay $50 or more
rent of the space it occupies. All these expenses,
on whether we use the machine or not. Thus, the
having bought this machine and kept it takes f
assets approximately one dollar per day.

9 If now the cause for idleness is ascertained
find the expense of each cause of idleness as sho
That part which is due to lack of orders points o
policy is wrong, or that the plant is larger than
other words that somebody in building the plant
the demand. It is clear, however, that no con
based on the figures for one month, but on the res
months during which the problem has been caref
mistake has been made in building too large a plan
be made to determine the proper disposal, or util
cess, in order that the expense of idleness may be t
if no profit can be made.

10 The next column shows the expense due t
which means that we must investigate the labor p

11 The next column showing the expense due t
material, is an indication of the efficiency of the
and storekeeping system. The next column refle
maintenance department.

12 If in any case the expense of idleness is gr
attributed to all of these causes together, it mu
column as poor planning.

[1] Trans. Am.Soc.M.E., vol. 37, p. 109.

13 We can hardly claim that such a chart gives us a *measure* of the efficiency with which the above functions are performed, but it certainly does give us an *indication* of that efficiency.

14 In several cases, the first of such charts gotten out resulted in the scrapping of machinery which had been idle for years. The space thus saved was used for a purpose for which the superintendent had felt he needed a new building.

15 In another case it resulted in the renting of temporarily idle machinery at a rate which went far toward covering the expense of carrying that machinery.

16 Under the first system of cost accounting the facts brought out by this method are not available and the increased cost that a reduced output must bear is a great source of confusion to the salesman. The newer system with its constant cost shows that non-producing machinery is a handicap to the industry of a company, just as workmen who do not serve some useful purpose in a plant, or industry, are a handicap to that plant or industry. *Similarly, plants or people, therefore, who do not serve some useful purpose to a community are a handicap to that community, for idle plants represent idle capital, and idle people are not producers but consumers only.* The warring nations have recognized these facts, and put both idle plants and idle people to work wherever possible.

17 The statements so far made concern principally the operation of industrial plants and the production of articles of commerce; but they are none the less true concerning the construction of industrial plants.

18 We may ask the same question about construction that we ask about operation; for instance, should the cost of a railroad include all the money spent by the people engaged in building it, or should it include only such money as contributed to the building of the road? As an illustration, is the cost of a piece of road which was built and then abandoned for a superior route before being put in operation a part of the cost of the railroad built, or is it an expense due to improper judgment on the part of the builders?

19 I am not discussing the question as to whether the public should be called upon to pay interest on the money uselessly spent through improper judgment, but I do think that *in all construction it should be possible to separate those expenses which contributed to the desired result from those which did not so contribute.* A comparison of these amounts will give a measure of the efficiency of the builders. On this knowledge, proper action can ultimately be taken.

20 Still another factor enters into the value
We all have known cases where the same plant
manager was a failure, and under another a v
The value of a going plant, therefore, consis
namely, the value of the physical real estate
the value of the organization operating it. In c
of an organization we should realize that it lies
personality of the managers or leaders (who may
as in the permanent results of their *training and*
go on with the business, and are therefore an
accident.

21 We have the authority of no less a p
Carnegie, Hon.Mem.Am.Soc.M.E., for the sta
ganizations were of more value to him than
we can determine exactly the value of a going
must find some means of measuring the value
which operates it, for this is an integral factor
an industrial property, which is just as real a
brick and mortar of which buildings are compos

22 Our charts showing the expense of idlene
rough indication of this value, for they show t
cient management.

23 If the above premises are correct, the f
important elements in determining the value of an

The cost of the product
The capacity of the plant
The portion of the plant operated, ar
The expense of maintenance of the id

In presenting his paper, the author said:

Inasmuch as the value of an industrial plan
ability to produce values, such a plant, which i
not actual but only potential value; actual
upon it only when human intelligence and indu
to produce the results for which it was creat
industrial plants are created in order to make
can be sold at profit or to render some service v
profit, amply justifies the statement that their
upon how they perform their functions. You n

the value of a going industrial property into two classes — that of the potential or static value of the plant itself, and that of the dynamic value of the organization operating it. In the past the former has been given attention by accountants almost to the entire exclusion of the latter, in spite of the fact that the stock-market prices and sales values of plants actually, though indirectly, reflect both values combined. If we would get a clear conception of the value of a going plant, we must find the means of measuring the value of the operating organization, which has often far more to do with its successful operation than the particular constructions and equipments of which it is composed.

In the operation of an industrial property, it is seldom that all the expense incurred during any one month can be utilized to advantage; but if we can find how much was utilized and how well it was utilized, we can get some idea of the efficiency of the management. How well the money used was used, is indicated by the cost of the article produced; but there has, in general, been no indication as to what became of expense incurred, but not utilized. How much expense was incurred but not utilized, and why it was not utilized, are vital factors in any attempt to measure the efficiency of the management; hence it is to these questions that we give special attention in this paper.

At the Buffalo meeting last year I read a paper on the relation between production and cost, wherein I contended that the true cost should include only those items needed to produce the article, and that any other money expended while these articles were being produced, for any cause whatever, should be charged to some other account. People are pretty generally keeping an account of the money expended for producing articles. *In but few cases are they taking care to find out how much money was expended that did not produce anything.*

We are all familiar with the fact that machinery costs money, even when it is idle. Interest on the money invested, taxes, depreciation, insurance, etc., all go on whether the machinery is used or not. The value of a going plant is very materially influenced by the amount of the above items that are unused or unusable.

This is a factor which is directly in the control of the management, and does not in any way concern the inefficiency of the workman; but it is a real factor in the valuation of a plant. Hence, a chart of the nature shown in my paper is a very real help in determining the value of a going plant.

DISCUSSION

W. S. Rogers thought that if Mr. Gantt's would show the unbalanced condition of the p just as important as anything. The divisions c further, and show plainly and clearly inefficier would point to and almost name the man, up i who was killing himself at his desk, absolutely u get a meal, and the biggest leak in the plant, about it.

William W. Crosby, speaking of machine i in textile mills parts of looms would be in the sto year at a time. Looms would be put into operat because certain grades of cloth were demanded, they were not required. With a sufficiently wo could keep our mills running at somewhere ne which they were designed, but they could not b nomically unless there were orders to run them c

H. B. Cheney[1] thought that every mill had running on a particular product, was the contr cotton mill it was probably the looms. In mak Mr. Gantt had, the idleness portion of the chart had arranged it, upon the 100 per cent method. calculations of cost, the controlling factor of th — should be taken into account, and all the were necessary to supply the looms be consid partments; if the looms ran 100 per cent of the cent of all the expense of having that departm be consumed in the cost; but if the looms, which factor, only ran 85 per cent, then only 85 per ce the overhead should be consumed. All the de chart on a 100 per cent method would unque with the greatest clearness every point that wa that time. But it might be an entirely differen latter. Profits, he believed, instead of being fi made, should be figured upon the running of other words, one article might earn $1 a day pr

[1] South Manchester, Conn.

another one $5 a day, and the former might make 50 per cent profit and the latter but 10 per cent.

H. M. Wilcox told of results secured at the Winchester Arms Company from a set of charts similar to Mr. Gantt's, only carried out in greater detail. They had studied the idle-machine time in the cartridge-manufacturing department for a year or more, and determined how much time they were losing from the possible productive time of the machines due to tools, to machines, to labor and to excess equipment. This had directed their energy toward the points where they were weakest in eliminating the idle-machine time by giving their foremen specific information in regard to where time was being lost. The work became more interesting to the men without any effort other than pointing out where they were losing time, and a big gain in production was secured from the activities of the shop men themselves. The cost of scrap was added to the cost of the idle time and called the cost of lost effort, a report of which was given to the foremen weekly. The foremen had responded by attempting to reduce this waste, which was absolutely all waste as far as operating interests were concerned. Mr. Gantt's paper showed a way by which the efficiency of the management of an industrial plant could be measured periodically, and there seemed no very good reason why this should not be used as a basis for the extension of credit to that organization.

William Kent wrote that in a textile mill, if many different styles were made, the demand for which varied with the season and with the fashion, part of the plant might be running full time and not be able to take the whole capacity of the remainder, some of which would therefore have to be idle part of the time.

The "expense actually needed" for the production of a variety of articles might thus include a part of the idle time of machines necessary to have on hand to care for a varying demand, but which could not be kept continuously employed; and in such a case it was right to charge some of the cost of idleness into the "normal burden" which was distributed in the machine-hour rate to the cost of the goods. A modification of Mr. Gantt's chart was thus suggested to show how much of the idleness of a machine or department was normal and necessary to the conduct of the business, and how much was abnormal and excessive. This might be done by drawing vertical lines on the percentage portion of the chart indicating the normal percentage of full capacity which each machine or depart-

ment was expected to run during a month of go
thought the expression "value of the property" v
was ambiguous and that its meaning should be def

JOHN E. MULLANEY stated that after his co
Mr. Gantt's chart for about a year, he thought i
while to know that if during the time they had be
ing expenses to the operation they had been getting
at the fullest efficiency. In order to show this t
idea of drawing a red line adjacent to each black li
to see if the black and red line agreed. If they di
in question had been made at 100 per cent efficien
being based on the standards they had set for
The red ones which extended above the black w
him, because they showed that they were getting
tive efficiency out of these particular operation
previously planned. He thought this particular
working ideas could be applied to an industry
each factory produced an article, and made its
central head.

WILLARD C. BRINTON stated that the discussi
of the advantages which large businesses obtain
figures relating to operation, sales and finance. A
a business of, say, $5,000,000 or more a year can
higher net-profit return from a graphic-control dep
any other way in which an equal amount of money
To get the best results from a graphic-control d
grade man should be in charge of the departm
should have no routine duties to prevent his g
charts all of the study which their importance just
corporations having a national sales distribution
three thousand curves plotted continuously in ord
trol the business. To insure that the man at the h
control department might be free to report in an
on the results of his studies of the charts, he shou
to that officer of the company who was most a
affairs as a whole.

H. V. R. SCHEEL said that there had been
sideration a principle which was something more
a discussion of ways and means. That we now ha

was a definite measure of the efficiency of the management. Considerations of where responsibility lay had been broadened; the efficiency of the individual workman on a machine had become only a part of the efficiency of that whole for which the management, particularly, had to accept responsibility. In other words, Mr. Gantt's paper recorded a criticism of the management, but a constructive criticism, inasmuch as its form enabled the management to correct errors, modify plans and policies, and lay out a course of procedure more intelligent than any routine way of handling facts heretofore employed.

Keppele Hall wrote that he believed that most management engineers would agree that the vast majority of failures to succeed in business were due to the lack of what Mr. Gantt characterized as the "ability to do things" on the part of those really responsible, rather than to any other causes. One could well go a step further and state that in many instances a lack of real knowledge of the vital elements of operation precluded the exercise of this ability by those who really possessed it.

Harrington Emerson wrote that industrial property, according to the point of view, had three values: namely, the cost of its reproduction, the amount it would bring under the hammer, and its going value — the real value. This latter was difficult to determine, not because it was impossible on the basis of present or past returns and with due allowance for a definite amortization, to convert net income into capitalized value, but because no one could see into the future far enough to set either any real valid amortization or to determine what next year's profits would justify as increased value.

If of two plants one was able to operate materials, labor and capital valuation at standard, and each combination of materials, men and capital was yielding the largest margin between cost and sales price, while in the other plant operation was below standard and without reference to difference between cost and sales price, the first plant might very easily have many times the value of the second plant, although inventories were identical.

Edward W. Bemis[1] thought that Mr. Gantt's paper did not have a large bearing on the work of the appraisal engineer in rate cases, for in these the attempt was to determine a fair value, which

[1] City Hall Square Building, Chicago, Ill.

might be quite different from the value based
earnings. The productive efficiency should l
rate cases; not, however, in determining the
but in leading commissions to allow a higher ra
pany of large productive efficiency, attributab
the management, as compared with a business

Stuart W. Webb[1] wrote that from the op
Mr. Gantt's suggestions regarding idleness exp
portant. It did not seem to him, however, t
he had usually heard referred to as "load fa
accurately the value of the plant, due to th
different things might cause a low load factor.
would probably be due to poor management,
factors, such, for instance, as a distinct chang
the management would have absolutely no c
throw a plant that had been operating effici
balance. It seemed to him, therefore, essenti
ments of which Mr. Gantt spoke at the end o
for the product and relations to the market (g

Henry P. Kendall wrote that at the Plir
of the burden which was not earned through
ating of all or any of the machines, was cal
If the plant ran its full equipment full time, it
wise there was a certain amount of unearne
deducted from the profit and loss for each d
of every four weeks. He thought that if the net
were based on the actual net investment and
allowing a proper return on net investment, a
mined on a basis of direct expense with only
of burden applied regardless of whether the enti
or not, we would then have a common and st
mining these three points, which at the presen

Robert B. Wolf said that he would like t
fact that Mr. Gantt had primarily made a str
tion unity. The records exhibited formed a
track of the economic forces in the industry pri
of enabling the management to have some re

[1] Vice-President Old Colony Trust Company, Bost

but unless such records were furnished to the management they could not possibly use these forces intelligently. The management should be given the necessary information to enable those in charge to become efficient.

F. J. Cole wrote that one of the principal expenses in the production of manufactured articles resulted from not having on hand material in sufficient quantity or of proper quality. He showed that scheduling was most essential to anticipate correctly dates when things must be done in their proper sequence, and called attention to the fact that while all of the machines going all the time was the most economical way, a surplus of machines is sometimes required to take care of variations in product.

Arthur C. Jackson said that he hoped Mr. Gantt would eventually bring before and show the Society the necessity of charts for a whole industry, and select and define and portray on these charts the definite essentials for the economic operation of that industry, so that each member, each corporation within that industry, would have set up before it the beacons without which it would be very apt to run upon the rocks.

R. S. Hale wrote that many of the questions about cost and value would become simpler if we would give up the idea that there was any abstract "cost" or "value," and instead should work on the basis that the business of the accountant and engineer was to provide data which would make it easier for the executive to answer certain questions, or rather to enable the executive to take action.

Mr. Gantt's charts were exceedingly valuable, because they helped the executive to decide what he could have done if he had had more material, or if he had had more orders, or had had more help. Likewise they helped show what could be done if certain idle machinery were disposed of, etc. The cost figures they showed were, however, more than useless in some cases; but the same was true of all cost figures.

A. C. Jewett wrote that the expense of idle equipment shown by Mr. Gantt's chart must be combated, so far as idleness due to lack of orders was concerned, by an intense study of the sales problem. The engineer must direct the sales policies. He must direct the distribution of the products of industry. The consuming power

of mankind was not limited. It was the d
that was at fault when machinery in the mills
and many people lacked sufficient food and c
new, but it was new for the engineer to gi
such matters as a part of his work of indus

J. B. MILLIKEN[1] wrote that his company
of industrial property to mean the value a
manufacturing property of a corporation wa
Mr. Gantt's paper evidently contemplated
viz., one including the efficiency of the plan
value of the organization operating it, which
integral factor in the valuation of an indust
agreed with Mr. Gantt, if by industrial prop
values which were represented by the capital
in which case the value of the management
profit and loss account and in the market val
shares; but the value of an organization cou
flected in the physical assets of the corpora
books.

Their view was that the valuation of land
ment should be shown on the books of a corpo
cost, less a depreciation for use or obsolesc
their valuations and depreciation ratios, they
by professional appraisers at intervals of ap
and compared results carefully with their
lieved that appraisals should be made on the
to replace, less proper allowance for age or f
than on the basis of original cost, as the latte
determine at the time of the appraisal and
represent more or less than real value, even a

J. H. WILLIAMS[2] wrote recommending the
rates in cost accounting as a means of increa
duction, as well as Mr. Gantt's "constant" c

The use of standard cost rates, or rates
cost should be as distinguished from what it
daily comparison of the aggregate of the ar
cost records (at standard rates) with the aggre

[1] Treasurer, The Yale & Towne Mfg. Co., Stamfo
[2] 772 Park Avenue, New York, N. Y.

Through their use it was possible to determine profit or loss due to *volume* daily; and by analysis of actual cost and comparison with standard cost, to determine their source.

On the other hand, through using the same standard rates in keeping production-cost records, it was also possible to determine profit or loss due to *efficiency* daily by jobs as they were completed. By comparing the actual with the estimated cost of production, the operations involving profit or loss could be determined.

The Author. It is not pretended that my paper contains a complete solution of all our valuation or accounting problems, but it does point the way to detect many of the sources of waste and inefficiency which have heretofore been disregarded, and which cannot be detected until we get a proper appreciation of values.

Among the discussions which are particularly significant I may single out that of Mr. William Kent, who seems to be troubled about the difficulty of fixing a valuation for taxation purposes, if this method of valuation be accepted. This is particularly pleasing to me because, if the methods which are proposed did harmonize with the present system of taxation, I should feel very much discouraged, for there is nothing which is so detrimental to our industries and to prosperity in general as our system of taxation, by which the energy, initiative and business success of the individual are taxed for the benefit of the community, and the wealth created automatically by the community is allowed to go, without any return, to individuals who, as a rule, are contributing no equivalent return to that community. If the wealth created automatically by the community should be claimed by the community, it is highly probable that it would be unnecessary to tax any of the industrial activities of individuals, and Mr. Kent's troubles would absolutely vanish. If the proposed system of accounting has a tendency to make that fact clear, it will do much to lift a burden from our industries and enhance the prosperity of the workers.

If we would meet the competition with which some of us think we are so direly threatened after the war, we must encourage industry and discourage idleness, for the warring nations, having found what an enormous increase in strength such a procedure has given them, will hardly return, when the war is ended, to the other method which we seem to cherish so highly.

Arbitrary laws based on opinions inherited from a bygone age are not suited to an age like this, when the struggle for existence, which

is so keen in Europe, threatens, perhaps in anothe
for the same causes that were active there are ac

Ninety years ago Thomas Carlyle said, "th
that can wield them."

It is a reversal of this policy which, more t
combined, has brought Europe to such dire ex
trol of the implements of production fell into the
who saw more profit in the control of markets
efficiency, which they did not understand.

Competition for the control of markets is at
cause of the great war, and the fact that Germa
clearer comprehension of the importance of prod
the necessity for the control of tools by the hands
is the explanation of her tremendous industrial a

During the last eighteen months England ha
and through her Minister of Munitions has tak
from stock and bond holders, and placed it in the
can "deliver the goods." The development s
this change is so phenomenal as to be almost unbe

In her attempt to save her life she has learn
in productive efficiency. The other European
doubtedly learned the same fact.

In the face of these examples are we still go
to market control until aroused by a catastrop
from the fate of others and begin at once to
efficiency? We have been talking efficiency in t
ten years, but so far the results have been lame
is not the fault of the workmen, for wherever we
at the top we have had but little difficulty in t
be efficient.

For years, with lack of efficiency at the top st
and hampering me at every turn, I have labored
measuring that efficiency, as it is perfectly ev
efficient direction, efficient workmen are ineffe
possible to get them, which it usually is not.

If we can measure and evaluate the product
manager as we now measure that of the workma
better results.

The only men organized for the promotion of
are the engineers, and it is on your shoulders, ge
fall the burden of showing what can be done.

I offer as a part of the work of measuring executive efficiency the chart shown in my paper. It is an attempt to measure the efficiency of the executives, and to indicate in a general way their value, which we know is an integral part of the value of any successful industrial property. This is only a first attempt, and I note that already one engineer has taken a step beyond what I offer.

No. 1579

TRIBUTES TO JOHN E. SWEET

As a tribute to the late Dr. John E. Sweet, Past-President, Honorary Member and a founder of the Society, who died on May 8, 1916, the regular business of the Annual Meeting was suspended at 12.30 p.m. on Wednesday, December 6, when a memorial service to Professor Sweet was held in the Auditorium of the Engineering Societies Building.

Mr. John H. Barr presided at the exercises, and the members of the Council and of the Council-elect, and the past-presidents of the Society took seats upon the platform. The Chairman announced that the ceremony would open with an address by Mr. Worcester R. Warner, Past-President of the Society, and Capt. R. W. Hunt escorted Mr. Warner to the platform. Mr. Warner paid the following tribute to the memory of Professor Sweet:

TRIBUTE OF WORCESTER R. WARNER

In honoring the memory of Prof. John E. Sweet we stand at the shrine of one whom to know was to admire and love — of a personality instinct with qualities that live to bless the world endlessly, through perpetuation in the lives of others.

To all members of The American Society of Mechanical Engineers he was known as the revered founder and third president of this Society.

As long ago as 1880 I, myself, received a letter from him in which he outlined the scope and benefits of such an organization and invited me to become a charter member.

In those remote days the term Mechanical Engineer had a very limited application. It did not appear in college courses or degrees. There were some master mechanics and master builders and draftsmen; but to secure sufficient members as a nucleus for his proposed organization, Professor Sweet had to invade the ranks of foremen and machinists — and there, I may add, is where he found me and some others.

Recently he wrote: "Likely the most important thing I have done was to set the ball rolling for the organization of The American

Society of Mechanical Engineers, of which ther
and five thousand members scattered throug
surface of the world." As he was proud of the
may we be proud of the founder.

Professor Sweet left a brief and very chara
outline of his life history, from which it appears
plain English parentage, in the village of Pom
His educational advantages were few, but he and
marked taste and aptitude for mechanics. "I wa
something," he says. Not unnaturally, there
learned the carpenter's trade, from which, as he sa
life ran into architecture," and he adds that th
several buildings designed by him. Whimsical
as "the Government at the seat of war at th
jobs to those who wanted to stand up and be sl
job he 'didn't hanker for,'" so, in 1862, he went
London, Paris, Switzerland and Italy." For nea
after he had employment in England. On his re
invented his very ingenious but commercially im
to supersede movable type — the forerunner of t
he undertook bridge construction and tells how
buiding a bridge at Ithaca that "President Whi
came around one afternoon" and asked him "t
Board," from which resulted his appointment a
in Sibley College. The six years, from 1873 to
he taught there, were full to overflowing of acti
of his "boys," between whom and himself existe
bonds of regard and affection expressed in their l
gatherings with him in recent years. "At Cor
were constantly developing new things: absolu
plates, straight edges, squares and angles, stand
use of the measuring machine which we built (tl
this country) and which read to the ten-thousan
enabled us to judge the forty-thousandth. V
versity were always shown our shop, and the won
were doing interested the greatest of them all.
constant, interested visitor while he lived. G
Hay and hundreds of others came." "In conve
Ward Beecher," he adds, "I casually remarke
college graduate was not much use in a machine
guessed I had never seen one in a pulpit!" An

diverted the young professor, he thus describes: "We built certain lathes, and, among other new things, all the bolts and nuts that had to be changed to meet conditions were made to be operated by the one wrench. Showing it off to a friend, he said, 'What in the world would you do if you lost that wrench?'" With deep feeling he sums up that period thus: "The best thing we made or helped make at Sibley, I suppose — or hope at least — was a lot of valuable men." From these fine men, "Sweet's Boys," have proceeded such answering testimony to the worth of their instruction by this tireless, inspiring teacher as constitutes, it seems to me, his most perfect tribute and imperishable monument.

One of them, Dean Smith, of Sibley College, writes me: "My memory pictures Professor Sweet always at work upon something that should contribute to the training of the students in his charge; and no matter how busy he was, he was always ready to give his attention to any one who sought his instruction or advice upon any matter whatever." "One thing he said has always stuck in my memory," adds Professor Smith. "It is so characteristic of the man. He said: 'When you go out into practice you will not be paid for what you know but for what you can do.'"

Another of these former students, Mr. J. E. Johnson, Jr., has written a masterly appreciation of Professor Sweet which will shortly appear in the *American Machinist*. To it I am indebted for certain details and the extracts from Professor Sweet's reminiscences. Much stress is laid by his students on Professor Sweet's ingenious applications of graphics to replace the higher mathematics of his subject. Mr. Johnson mentions this interesting instance: On one occasion during a call from some professor the latter spoke of the very difficult problem in geometry which the mathematicians had just succeeded in solving, namely, that if three circles of different diameters are drawn in any position in a plane and a pair of tangents are drawn to each side of each pair of circles and prolonged to their intersection, the intersecting points of the three pairs of tangents will lie in a straight line. Professor Sweet thought this over for a few minutes and said: "Yes, certainly, I can see that that is true." The other professor said: "I guess you don't understand, Professor. This is a very difficult problem and we have just finally accomplished its solution. I don't think it is as obvious as you think it is." "Why, yes," said Professor Sweet, "of course it is obvious. Instead of three circles in a plane, take three balls lying on a surface plate. Instead of drawing tangents, imagine a cone wrapped around each pair of balls. On top of the three balls lay another surface plate. It will rest on the three balls and will necessarily be tangent to each of the three cones. The apexes of all the cones must lie in the intersection of two surface plates, and as the intersection of two planes is always a straight line, the apexes of the cones will lie in a straightline. It seems to me that this is perfectly obvious." So it was to a man who could think in those terms, but to how many of us, no matter what our mathematical training, would it be "obvious?"

At the Centennial Exposition in 1876, the notable feature of the Sibley College exhibit was Professor Sweet's "Straight Line" engine, constructed at the college shop. As a marked departure in design from previous types, it attracted much attention; the more so since, when tested out, it developed a high efficiency. Encouraged by these facts, the inventor decided to leave the teaching profession and to devote himself to the manufacture of his engine in his home city of Syracuse. So the Straight Line Engine Company was organized in 1879, with Professor Sweet as president — an office he held until his lamented death — and there Sibley College post-graduates and others of "Sweet's Boys" received their advanced practical training.

Many present, beside myself, must remember the kindly sentiment, "Visitors Always Welcome," cut in the stone over the entrance door of the factory, and, like myself, must have put it to the test more than once, only to find that welcome ever ready. And the Straight Line engine itself was a familiar friend, too. We installed one in our first shop, Swasey and I, and used it until we needed more power. Then it passed to Brashear, in whose shop it is working steadily still — a tribute to the inventor's sound sense and feeling for right design.

Professor Sweet's name and fame became so well known that engineers and inventors made "a beaten path to his door."

It was his pleasure to commend and aid those who possessed merit. Just as ready was he to wisely direct those who came bringing heretical ideas in opposition to the fixed laws of mechanics. In such cases his keen sense of humor often played a kindly but convincing part. On one such occasion an inventor, after demonstrating an elaborate model, asked, "Now, Professor Sweet, what do you think of it?" Instantly came the characteristic reply, "Well, it seems to be a mighty good way to do a thing that does n't need to be done." I recall, in the days when inventors tried to apply ball bearings to everything from bicycles to the axis of the earth, one such went to him to explain the merits of his new ball bearing for buggy axles. Professor Sweet listened attentively and then said to the proud inventor: "The ordinary buggy axle is of steel, about $\frac{7}{8}$ in. diameter, nicely fitted to a cast-iron sleeve and, presumably, well oiled. If you place such a buggy on a new, smooth floor, and with a sensitive spring scale, determine how slight is the power required to move it, you will see that, if your ball bearing saved it all, it wouldn't be worth while." He followed this statement by saying, "The resistance to moving vehicles is mostly on the ground and can be best overcome by making better roads." This now seems like prophecy, in view of the fact that the State of New York has, since then, spent over one hundred millions of dollars in improving its roads.

Thousands of young mechanical engineers discover the great loss of power by the use of the crank in the steam engine and are then subject to attacks of the rotary-engine fever. Most of my hearers have doubtless prescribed for this malady and accomplished cures. So did Professor Sweet. For, when one thus afflicted came to him with a splendidly worked out design, the cylinder but half the usual circle, and enthusiastically invited his approval, Professor Sweet said cordially, "It is the very best rotary engine I have ever seen, for you have at the outset thrown away half of it. Now all you have to do is to throw away the other half." Then we may be sure he gently put that young enthusiast back on the right track to real progress. This characteristic helpfulness to others held good to the very day of his death. For another of his "boys," Mr. Wm. C. Brown, tells me that on the morning of that day — it was the eighth of last May— a man from Texas, whom he had never heard of before, walked into the Professor's office to ask advice about a valve motion he had invented. When it had been explained but before Professor Sweet had an opportunity to express an opinion, his carriage came and he was obliged to go home to lunch, making an appointment for the man to return at a later hour. That afternoon the fatal stroke occurred and Professor Sweet was taken home by Mr. Brown, in the latter's car. On the way, though suffering intense pain, he told Mr. Brown about the stranger's errand and begged him to see the man when he returned and explain to him just why his valve motion was of no value. Here is the great human note, without which even the harmonies of Heaven would be poor.

In these latter years Professor Sweet had enj
Of this he writes: "I have crossed the Atlantic
been in ten seas, twenty-two countries, and in
bays and cities to enumerate. I have been fo
South of the Equator and half way to the North
a warm interest in world politics, especially as affe
war of nations. This is the theme of several persor
one of his very favorite "boys," Mr. F. G. Tallma
fessor Sweet's sentiments concerning the Civil W
to find him saying: "I have said, and said it ye
pare for war in times of peace' is the worst sentin
language, and if anything was necessary to pro
war does it. Men fight more for glory than
there would be no glory in taking an unprotec
disgrace to the people who would do it. I believe
be in a safer condition without a cent's worth of
and forts than with the best possible of those elen
vigorous words from a man of eighty-two. Sever
same vein. All express, at more or less length, h
of war and the specious excuses other men make

As I turn the pages of this very private corr
sent me by Mr. Tallman, I find one letter in q
very amusing vein. I quote: "Some two weeks
about what was the first thing I ever made that
about — and it was a boot jack that I made
1840, or when I was eight years old. And I call
that I did then what I have always tried to do sin
than the ones made before." Then the writer t
at Pompey a few days before and mentioning t
one "went out in the wood-house and brought i
tinues, "I have it now in my office. Wherein it
is this: the old, or, in fact, all others, have a cl
end, to hold that end up. In the place of the
holes through and put in two pins which I let
each side, so the jack is right side up whiche
floor. As an evidence that it was used in the
back end is worn tapering and the forked end
the boot heels. The point is that I did then w
tried to do since — *make the thing better*. But g
to adopt my notions has been as barren of resu
about preparedness."

Likewise, in his reminiscences, Professor Sweet writes: "I cannot call to mind ever starting on a job without thinking out how to make it better than it had been done before." The engineers of the world can set themselves no higher task than such a motto imposes, nor hope to accomplish it more fully and nobly than did Professor Sweet that which he undertook.

Simple and modest and self-effacing as he was by nature, honors sought him — not he them. In 1914 the John Fritz medal was awarded to him "for his achievements in machine design and for his pioneer work in applying sound engineering principles to the construction and development of the high-speed steam engine." In the same year Syracuse University conferred upon him the honorary degree of Doctor of Engineering. But, as has been truly said, these honors, great as they are, are less lasting than will be "his influence over men, to cause them to think straight and live honorably."

The man of whom such things are true need make no claim of his own. In life he is revered. In death we bring him the homage of our hearts.

The Chairman expressed the Society's indebtedness to Mr. Warner for his appreciative estimate of Professor Sweet. He then introduced another pioneer of the Society who was associated with Professor Sweet in his early days, Capt. R. W. Hunt. Captain Hunt said in part:

TRIBUTE OF CAPTAIN R. W. HUNT

Blessed are those whose works are such that every day and every moment of that life has been to make their fellow-men better and happier, and such was Professor Sweet. His life was as simple as his mechanical ideas, and just as accurate in its greatness and in its purity. To know him was to be blessed, and certainly his influence exerted upon those who came in contact with him, and left as a beneficence to those who follow him, is of a character that makes the world worth living in.

His achievements were not of a startling kind. His successes did not bring to him great fortune, but each one made men better and happier, and the fact that following the years of his teaching he established a brotherhood of love among men who continued to pay tribute to him until the day of his death was an achievement of which any man should be glad and proud, and sufficient to content any man.

The Chairman then called upon Prof. Alb of Sibley College, Cornell University, who w "boys" in the early days of Cornell, and wa tendent of the Straight Line Engine Works.

TRIBUTE OF DEAN ALBERT W. S

Professor Smith said he counted it one o privileges of his life that Professor Sweet was College and his employer in the Straight Line that there were two things which characteriz Professor Sweet: *first,* his clearness of thought, human sympathy. Dean Smith continued:

"As I said once before in public, Professor carried a little sharp needle to prick hot-air ball he saw the balloon collapsing, his sympathy fc brought it to him was so great that he would e to blow balloons so that they could not be punc

"I want to tell one story that has not been ing. The Professor was especially proud of tl gine because, although it was a high-speed en One day a man from Syracuse came to the w Line engine was on the skids ready to be ship not an engineer. He came in and the Professor said to the Professor, looking at this engine, making drop hammers.'

"It seems to me that in human life there sin, and that is selfishness. All others are si modifications of this one sin. It is the one of w was guiltless all his life, for he was the kindes man that ever offered help and sympathy t mortals. I am proud of the fact that it was gi of 'Sweet's Boys.'"

The Chairman in conclusion said that it see who knew Professor Sweet and loved him so de rial should be characterized by extreme simplici with these tributes the service closed.

No. 1580

NECROLOGY

CHESTER BIDWELL ALBREE

Chester Bidwell Albree was born at Allegheny, Pa., on April 8, 1862. He received his preparatory education in the Western University of Pennsylvania and his technical education at the Worcester Polytechnic Institute in Massachusetts, from which he was graduated with the degree of B. S. in 1884. Following his graduation Mr. Albree spent a year in travel, visiting manufactories in various cities, and selling lubricating oils. Before entering into business with his father, he worked in the drawing room at Thomas Carlin's Sons in Allegheny. The remainder of his business life was spent in establishing and managing The Chester B. Albree Iron Works.

While his principal business was ornamental iron, Mr. Albree started to manufacture pneumatic compression riveters in 1900, and did all the designing of these machines himself. He originated the very successful universal bail, whereby a suspended machine may be turned in any position by merely swinging it through a bail, so constructed as to keep always the center of gravity of the machine at the same height, and thus preserve stable equilibrium. Among his other valuable patents was one covering an automatic pneumatic compression riveter which does away with the adjustment screw entirely. He was widely known as a manufacturer and designer of bridge railing, and many of his beautiful designs can be seen in all parts of the United States.

Mr. Albree was a past-president of the Engineers' Society of Western Pennsylvania and a member of the American Association for the Advancement of Science. He became a member of this Society in 1886. He died on May 27, 1916.

ROBERT ALLISON

Robert Allison was born at Middletown, Durham County, England, on December 25, 1827, but was brought to this country by his parents when very young. He received his early education in the schools of Shamokin, Northumberland County, Pa., which were taught by farmers for four months during the year.

At the age of fifteen Mr. Allison was appointed
neer at the Shamokin furnace, and the next year h
to the firm of Haywood & Snyder, Pottsville, Pa., t
of machinist. For about two years following the
apprenticeship he worked as a journeyman, follow
cepted a position as foreman in the shops of T. H.
Carbon, Pa., where he introduced and developed ma
in the shops, one in particular being a tool for boring
machine did five times as much work as that done b

About 1863 Mr. Allison in partnership with
Pottsville, Pa., started up the Franklin Iron Work
which had been idle several years. One of the most
tions of this company was the Allison Cataract St
revolutionized the system of freeing the undergr
water. Another important invention was the h
diamond drilling machinery.

In 1878 Mr. Allison purchased Mr. Bannan
works and operated them alone until the time of
1901.

Mr. Allison was a member of several masonic
of the Historical Society of Schuylkill County, the
Club and the American Institute of Mining Engin
a member of this Society in 1884. He died at his
bon on February 3, 1916.

DARWIN ALMY

Darwin Almy was born in Tiverton, R. I., on F
and died in Providence on March 9, 1916. He a
schools in Tiverton until about sixteen years of age
work on his father's farm until 1868. Afterward
terested in menhaden fishing and was master of one
steamers out of Tiverton.

In 1874 he went to Providence and formed a p
manufacture of jewelry, which was abandoned, ho
years later. After this, he returned to the fishing

In 1878 he entered the employ of the Herresh
Co. at Bristol, R. I., and had charge of the boile
also had the advantage of taking part in numerou
steam yachts. During the first part of this time
conducted there a series of experiments under A

Admiral Isherwood on engines and boilers, and through this association he became very much interested in steam engineering.

In 1890 Mr. Almy took out his first patents on the boiler which bears his name, and organized the Almy Water-Tube Boiler Co., of which he was the head until his death.

He was a member of the American Society of Naval Engineers, Society of Naval Architects and Marine Engineers, Providence Association of Mechanical Engineers, Chamber of Commerce of Providence, Central Club of Providence, Bristol Yacht Club and Rhode Island Yacht Club. He became a member of this Society in 1893.

JOHN VANDERVEER BEEKMAN

John Vanderveer Beekman was born in Peacock, Morris County, N. J., on February 7, 1842, and received his education in the district schools near his home. For four years he served an apprenticeship at the Dunham & Staats Iron Works, at Raritan, N. J., and then went to Brooklyn, where he remained for six years with the South Brooklyn Engine Works. For two years following this he engaged in business for himself, having a general machine shop in Brooklyn. This was afterwards organized as the Pioneer Iron Works, of which he was manager. Finally, he became interested in the Lidgerwood Manufacturing Co., manufacturers of hoisting engines and boilers. He was well known in engineering circles and was the inventor of many devices now in use on hoisting engines.

Mr. Beekman was a member of the Machinery Club, the Plainfield Country Club, and the Hydewood and Stamford Golf Clubs. He joined this Society in 1889. He died at his home in Plainfield, N. J., on September 11, 1916.

THOMAS HATTERSLEY BELCHER

Thomas Hattersley Belcher was born in Newark, N. J., in December, 1876. He was educated in the Newark High School and later in the technical and mechanical drawing schools. His early experience in mechanical work was obtained while in the employ of Cyrus Currier & Sons, Newark, N. J. In 1900 he affiliated with A. & E. Brown Co. as engineer and superintendent of installations of power-transmission equipment in manufacturing plants, and in 1904 he became assistant chief engineer of that concern. Between 1906 and 1913 he was associated with the Chicago Coated Board Co. as general mechanical superintendent. In May, 1913, he became engineer and

representative for the Black-Clawson Co., Han
after two years, to become manager of the Car
of Carthage, N. Y. His death occurred Nover
acting in this capacity.

Mr. Belcher became a member of this Society

JUDSON H. BOUGHTON

Judson H. Boughton was born at Rochester,
10, 1881. He received his early education at th
Academy and was graduated from Cornell Univ

From 1907 to 1915 he was interested in the
water, gas, and street railways in the South and
that time he was secretary and treasurer of the
provement Co., a New Jersey holding company;
of the Fort Worth Light and Power Co., of Fort
dent of the Citizens Railway Co. and the Wac
Tex.; consulting engineer for the Hot Springs
Springs Gas Co., and the Hot Springs Street R
Springs, Ark.; president of the National Light an
Jersey holding company; president of the Fulto
Power Co., of Fulton, Ky.; president of the Hickn
Co., of Hickman, Ky.; president of the Prairie C
Co., of Prairieville, Mo.; consulting engineer for
road of Missouri; and consulting engineer on vari
During the last two or three years Mr. Boughto
his time to the boat industry, and in this connecti
was president of the St. Louis Yacht and Boat
Lakes Boat Building Corporation of Milwauke
giving all of his attention at the time of his deat
Milwaukee, Wis., on July 29, 1916.

Mr. Boughton was a member of the American
cal Engineers, the American Street Railway
National Electric Light Association. He was al
Racket Club, the City Club of St. Louis, the Uni
waukee, the Milwaukee Yacht Club and the C
He became a member of this Society in 1903.

J. LINWOOD BROWN

J. Linwood Brown, who joined the Society i
England in 1846. His experience as an engine

service, in which he progressed to responsible work in mechanical and construction engineering. He built and operated the Southern and West Wisconsin Railroad; held the position of engineer of construction with the Mexican Central, the Veronej and Rosstoff Railroad and the Panama Railroad, and was subsequently superintendent of motive power and maintenance of the Ohio Southern and Pittsburgh & Western Railroads. His later work was as superintendent of the Bureau of Water Supply and Distribution of the city of Allegheny, Pa.

He died on September 11, 1916.

WILLIAM LAING BUSS

William Laing Buss was born at Corry, Pa., on November 26, 1867, and died on May 3, 1916. He received his early education in the public and high schools of Troy, N. Y., following which he engaged in the drug business. In 1888 he moved to Chicago and entered the Rush Medical College, leaving at the end of two years to go into business with his father who was western manager for the Chapman Valve Co., of Springfield, Mass. Upon the death of his father in 1903, Mr. Buss succeeded him as manager and remained there until 1909.

In 1910 he became general manager of the Coffin Valve Co., of Neponset, Mass., which position he held until 1912, when he returned to Chicago to accept a position as sales manager for the Pittsburgh Valve, Foundry and Construction Co., of Pittsburgh, Pa., remaining with them until 1915. At the time of his death he was a manufacturers' broker, with offices in Chicago.

Mr. Buss was a member of several masonic lodges. He joined this Society in 1912.

ROBERT CAIRD

Robert Caird, a managing director of the old family shipbuilding and engineering company of Caird & Co., Greenock, Scotland, was born on May 22, 1852. He was educated at the Grange School, Sunderland, at the Greenock Academy, and at Glasgow University. He entered his father's works as an apprentice-engineer, continuing at this time for eleven years in the business. He was associated with the Maritime Construction Co., of Havre, during 1881 and 1882, and in the two following years was engineer-in-charge of construction for the Pullman Co., of Chicago. In 1888 he returned to his father's business and became one of the managing directors, taking special charge of the engineering department.

In 1900 Glasgow University conferred upon
degree of Doctor of Laws. This was a well-meri
in bestowing it upon him the Faculty of Law
reason: "His eminence as a shipbuilder has alrea
gold medal of the Institution of Engineers and Sl
land and the presidentship of that important bod
which possesses the only British Chair of Naval Ar
associate herself with the Royal Society of Edi
her appreciation of Mr. Caird's valuable contribu
of applied science. We cannot keep out of view
Mr. Caird has displayed in the present movemer
ment and extension of the scientific side of th
Caird is not merely a practical man. Following
of Scots students, Mr. Caird completed on the C
tion he began here. In the course of several year
he acquired an intimate and extensive acquaintanc
and literature of France, Germany and Italy, and
leged to carry on important investigations in the
and art under the direction of Mr. Ruskin. His la
ment were recognized in 1895, when he was creat
Crown of Italy. His papers, lectures and addre
in which he is interested display not merely s
marked literary charm."

He was a close student of the work of James
delivered the Watt Anniversary Lecture before th
sophical Society. This lecture was a characte
memoir of Watt's contributions to the advancem
Dr. Caird was much interested in technical educ
author of several monographs on this subject, incl
Science in Relation to Education and Technical E
an active supporter of institutions in the city of C
prominent in the founding of the James Watt Me
took the part of a captain of industry in connectio
local business and philanthropic institutions.

Dr. Caird was a member of the Royal Society
Institute of Metals, the Philosophical Society of
Philosophical Society of Greenock, and became a
stitution of Civil Engineers in 1900. From 1899 to
dent of the Institution of Engineers and Shipbuild

He was elected to membership in this Society
at his home at Greenock on December 1, 1915.

ALPHONSO H. CARPENTER

Alphonso 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 fourteen 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, December 24, 1915, he was vice-president of the Acme Machinery Co., of Cleveland, O., which position he had held for many years.

Mr. Carpenter was elected to membership in the Society in 1895.

GEORGE NEWBOLD COMLY

George Newbold Comly, whose death occurred in Syracuse on March 22, 1916, was born at Byberry, Philadelphia, Pa., on August 6, 1851. He was educated at the Friends' School at Byberry and also at the Friends' Central School in Philadelphia, and served an apprenticeship in the machine shop of Wm. B. Bement & Sons in Philadelphia for four years. Following this, he was foreman of the machine shop at the Edge Moor Iron Works, and later was put in charge of the drawing room.

In 1890 Mr. Comly started an engineering business for himself, doing considerable business for the Newport News Ship Building and Dry Dock Co. In 1896 he became connected with the Solvay Process Co., of Syracuse, where he was in charge of the engineering department for 14 years. For over two years preceding his death he was Deputy Commissioner of Buildings for the city of Syracuse.

Mr. Comly was a member of the Citizen's Club, The Technology Club, the Syracuse Country Club, and the Social Literary Club, all of Syracuse. He was also a director of the Society for the Prevention of Cruelty to Children. He became a member of this Society in 1880.

NAPOLEON DUBRUL

Napoleon DuBrul was born in Montreal, Canada, on June 22, 1846, and died in Cincinnati, October 23, 1916. At the age of fourteen he became apprentice in the Gilbert Machine Shop. In 1866 he went to Chicago, where he invented the tin cigar mold and later the tin-lined wooden cigar mold. He removed to Cincinnati in 1872 and in 1879 became a member of the firm of Miller, DuBrul & Peters Manufacturing Co. In 1893 the members of the firm bought and reorganized the Anniston Pipe & Foundry C which was later consolidated with the Ameri Co., of Chattanooga, Tenn., and in turn was s Iron Pipe & Foundry Co.

Mr. DuBrul devoted his inventive skill tc machinery for making cigar molds. He .des marketed many different machines and appliai business in all countries of the world.

He became a member of this Society in 1900 the Business Men's Club and the Queen Club c

CLAUDE M. DUGAN, JR.

Claude M. Dugan, Jr., who was elected t Society in 1913, was born in Philadelphia, Pa., He received his education at the Boys' Manual at the University of Pennsylvania, from which la in 1898.

In 1899 he entered the employ of the Cement Co. as a chemist, and later became c with the Wm. Krause Son Cement Co. and the in Philadelphia. In 1905 he became assistant Kosmos Portland Cement Co., at Kosmosdale, manager of the manufacturing department a manager of the company. In 1913 he became p Engineering Co., which was afterwards absorbe neering Co., of which he was vice-president.

He died on April 11, 1916.

JACQUES HENRI EDWARDS

Jacques Henri Edwards was born in Paris, 1889. He was graduated from the University the degree of B. S. During the following year h technical man with the firm of Jean Horowit automobiles, steam trucks and herringbone g then accepted a position with the American Tob for the purpose of improving his knowledge of and business methods. He also worked in De Carburetor Co. as assistant manager and alsc Corporation and the Northway Motor & Ma was a member of the firm of Breeze & Chapma of Detroit. In October, 1913, he went to St. Louis on special efficiency work and in June, 1914, entered the employ of Stanley G. Flagg & Co., who sent him to London and Paris as sales engineer. He held this position at the time of his death, which occurred in Paris on February 24, 1916.

Mr. Edwards was a member of the Automobile Association of America. He became a member of this Society in 1913.

ALFRED C. EINSTEIN

Alfred G. Einstein was born in Hoboken, N. J., in 1866. He received his elementary education in the St. Louis Public Schools, later graduating from the Manual Training School. He began a course in Washington University, but left before its completion, to become manager of a mining property in Silver City, N. M., which position he held for several years.

In 1891 Mr. Einstein returned to St. Louis to become president of the Consolidated Engineering Co., holding this office until 1894. During that period he succeeded in building 40 different plants, including waterworks, steam plants and electric- and street-railway plants. Between the years 1894 and 1896, he was vice-president and manager of the Paducah Electric Light and Street Railway Co., of Paducah, Ky. He sold out his interest in this company in 1896 and returned to St. Louis, where he purchased the Suburban Electric Light and Power Co. He also organized the St. Louis County Gas Co. In 1904 he sold the Suburban Electric Light and Power Co. to the North America Co., and in 1906 purchased the King Electric Light Co., which later was merged with the Suburban Electric Light and Power Co. In 1911 he became vice-president and general manager of the Union Electric Light and Power Co., which office he held at his death, on November 20, 1916.

Mr. Einstein had always taken an active interest in the business and civic affairs of St. Louis. He was the fourth vice-president of the Business Men's League, and was largely instrumental in bringing many conventions to St. Louis. He was elected to membership in this Society in 1905.

THEODORE NEWELL ELY

Theodore Newell Ely was born at Watertown, N. Y., on June 23, 1846, and died at his home at Bryn Mawr on October 28, 1916. He received the degree of C. E. at Rensselaer Polytechnic Institute in

1866, the honorary degree of M. A. from Yale University in 1897, and the degree of Sc. D. from Hamilton College in 1904.

Mr. Ely's entire business career was spent in the employ of the Pennsylvania Railroad, the engineering department of which he entered in 1868. From 1903 to 1910 he was chief of motive power of the lines east and west of Pittsburgh and Erie, and at the time of his retirement in 1911 was chief of the whole system.

He was also a director of the Pennsylvania Steel and the Cambria Steel Companies, and was a member of the Permanent Commission of the International Railway Congress, a member and president since 1904 of the Eastern Railway Association, and a member of the American Railway Association, the American Society of Civil Engineers, the American Institute of Mining Engineers, the Institute of Civil Engineers of Great Britain, the American Philosophical Society, the American Historical Association, and The Franklin Institute.

Mr. Ely was a fellow of the American Association for the Advancement of Science, vice-president of the American Academy in Rome, and an honorary member of the American Institute of Architects. He belonged to the Century, University, and Engineers' Clubs of New York, the Philadelphia and Engineers' Clubs of Philadelphia, and the Metropolitan Club of Washington. He was known for his interest in art and music, and was a director of the Philadelphia Academy of Fine Arts and a trustee of the Drexel Institute and the Philadelphia Commercial Museum. Mr. Ely became a member of this Society in 1880 and served as Manager from 1880 to 1881 and as vice-president in 1882.

CHARLES ALBERT FRANCIS

Charles Albert Francis was born in Mansfield, Mass., July 19, 1866. He received his early education in the schools of Mansfield and served an apprenticeship there with the Mansfield Watch Tool Co. From 1884 until 1890 he was machinist, tool and model maker for L. L. Pollands, of Worcester, Mass. In 1890 he was mechanician for the Department of Physics in Clark University in Worcester. He remained there until 1894 when he was chosen for a similar position at the University of Chicago, where he remained until 1899. Between 1899 and 1903 he was tool maker for the Veeder Manufacturing Co., at Hartford; special mechanician at the Yerkes Observatory; foreman of the Meter Department of the Diamond Meter Co. in Peoria, Ill., and with the Pratt & Whitney Co. in Hartford.

In 1903 he became instructor and master mechanic at the Higher Technological School in Tokio, and in 1913 he accepted a position as foreman at the General Electric Co., of Pittsfield, Mass., leaving in 1915 to enter the employ of the Remington Arms Co., of Bridgeport, Conn.

Mr. Francis became a member of this Society in 1906. He died at his home in Bridgeport, Conn., January 4, 1916.

WM. H. FRANCIS

Wm. H. Francis, who was elected to membership in this Society in 1884, died in Philadelphia on March 16, 1916.

GEORGE GILMOUR

George Gilmour, born at St. Petersburg, Russia, on December 25, 1864, received his preparatory education in the public schools of Russia and England and his technical education at Anderson College, Glasgow, Scotland.

He learned his trade as mechanical engineer with John Elder Co., shipbuilders and engineers, of Glasgow, Scotland, and was subsequently sent by them as an engineer to South America. On his return by way of Panama, in 1883, he was engaged by the French engineers to take charge of the mechanical department of the work on the Panama Canal, later holding a position with the Atlas Line of mail steamers.

When the West India Improvements Co. bought the Jamaica Government Railway in 1891, Mr. Gilmour was appointed master mechanic of the Jamaican system, and later became superintendent of motive power and consulting engineer to the Jamaica Light and Power Co. After eight years of service in Jamaica, he was appointed mechanical engineer to the New York Telephone Co., of New York City. After eight years of experience in this capacity, he resigned in 1905, to become chief engineer of the Travelers Insurance Co., which position he held at the time of his death. In 1910 the New York Commission on Employers' Liability and Causes of Industrial Accidents appointed him its consulting engineer and in that capacity he made an extended tour abroad to investigate and report upon the European methods for lessening the number of industrial accidents.

Mr. Gilmour was a charter member and trustee of the American Museum of Safety, and was also a member of the American Railway Master Mechanics' Association, the International Boiler Makers'

Association, the American Society for Testing Materials, and the Engineers' Club of New York. He became a member of this Society in 1909. Mr. Gilmour died at his home in Brooklyn, N. Y., on June 15, 1916.

CHARLES H. GUCKEL

Charles H. Guckel, a member of the Society since 1901, was born in Düsseldorf, Germany, December 21, 1876. His family moved to this country shortly after, locating in Pennsylvania.

In 1891 he entered the employ of the Pennsylvania Plate Glass Company, Irwin, Pa., as an apprentice, remaining with this company until 1894. In that year he became identified with the Westinghouse Electric & Manufacturing Company, Pittsburgh, Pa., in whose employ he continued until 1898. He then assumed the duties of general superintendent of the New York and Queens Electric Light Company at Flushing, Long Island, N. Y., where he remained for more than three years. He spent eight years in the Panama Canal Zone, and in 1908 he accepted the position of general manager of the Dover Electric Light Company and the Rockaway Light and Improvement Company, of Dover, N. J.

In June, 1912, he became affiliated with the Springfield Gas and Electric and the Springfield Traction Companies, Springfield, Mo., in the capacity of general manager, resigning about one year prior to his death, to enter the electrical field as a general contractor.

Mr. Guckel was prominent in Masonic circles and was also a member of Florence Lodge of Elks. He died on November 1, 1916, as the result of injuries sustained in an accident.

HARRY A. HAAS

Harry A. Haas was born at Tamaqua, Pa., on March 31, 1887. In June, 1902, he was graduated from the Tamaqua High School and became a clerk in his father's general store. He entered Mount Hermon School in Massachusetts in the summer of 1906, graduating in August, 1907. He then entered Lehigh University and graduated with first senior mechanical honors in 1911.

During the summer of 1909 he was employed as a teacher of science at Mount Hermon School, and during the summer of 1910 he was employed in the gun shop of the Bethlehem Steel Co. Following his graduation from Lehigh University, he entered the employ of the Winchester Repeating Arms Co., in the capacity of equipment engineer for the cannon, case, annealing, washing and packing shops and for special machinery, which position he held at the time of his death.

Mr. Haas was elected to membership in this Society in 1915, and was also a member of the Lehigh Alumni Association. He was drowned in Lake Bantam on July 15, 1916.

EDWARD THOMAS HENDEE

Edward Thomas Hendee, who was born at Claremont, N. H., February 22, 1880, died at Minneapolis, Minn., November 12, 1916. Upon graduation from New York University in 1900 he received the degree of B. S., and immediately assumed the duties of instructor and assistant professor of chemical and mechanical engineering at New York University, receiving the degrees of M. E. and M. S. in 1901, when he also received the degree of Sc. D. at Columbia University.

In 1902 he associated himself with the firm of Joseph T. Ryerson & Son, Chicago, Ill., as mechanical engineer, and in 1906 became manager of the machinery department. Between the years 1909 and 1913 he acted in the capacity of assistant to the president of the company and in 1913 became secretary, continuing as such until his death. Under Mr. Hendee's management both the domestic and foreign machine business and the railway-supply business of the company were very widely extended. Besides his affiliation with Joseph T. Ryerson & Son, Mr. Hendee was vice-president and director of the Lennox Machine Co. and director of the American Glyco Metal Co.

He was a member of the University Club, of the Alumni Board of Trustees of New York University, and of a number of athletic clubs in Chicago. He was elected to membership in this Society in 1908.

EMIL HERTER

Emil Herter was born in New York City on January 16, 1860, and received his education in the public schools of New York City. He served an apprenticeship with Gustav Burger from 1874 to 1880, and from 1881 to 1883 held a position in the drawing room of the Rider Hot Air Engine Co., at Walden, N. Y., and with J. H. Wright, at Bridgeport, Conn. From 1883 to 1884 he worked in the shop of the Colt's Patent Fire Arms Company.

His other practical experience included positions as patternmaker and machinist with the Goshen Foundry & Gas Machine Co., at Goshen, N. Y., draftsman with Brown & Hall, New York City, and

Adams & Richards, New Brunswick, N. J. He had charge of erecti machinery at the Niles Tool Works, Hamilton, O., and followi that worked as draftsman with the E. W. Bliss Co., in Brookly N. Y. From 1890 to 1892 he had charge of machine-shop work in t Schenectady branch of the Edison General Electric Co., after whi he held the position of chief draftsman with the New Jersey a Pennsylvania Concentrating Works. At the time of his death was chief mechanical engineer with the Edison Storage Battery C working on many of Mr. Edison's most important inventions.

Mr. Herter became a member of this Society in 1894. He di on August 23, 1916.

HOWARD DRYSDALE HESS

Howard Drysdale Hess was born at Philadelphia, Pa., Septemb 6, 1871. He received his education at the Central High School Philadelphia and at Lehigh University, from which he was graduat with the degree of M. E. in June, 1896.

Following his graduation he was appointed general manager the Eastern Steel Co., at Pottsville, Pa., which position he held f one year, resigning to accept a position on the faculty of Drexel I stitute, where he served two years. During 1904 he served on t faculty of the University of Kansas at Lawrence, and from there we to Cornell University as assistant professor of machine design. 1910 he was appointed professor of machine design, which position held at the time of his death.

Professor Hess was a member of the Engineers' Club of Phil delphia, Sigmi Xi honorary fraternity, and the Town and Gow University and Cosmopolitan Clubs of Ithaca. He became a mer ber of this Society in 1903. He died in Buffalo on April 22, 1916.

JOHN ALEXANDER HILL

John Alexander Hill, president of the Hill Publishing Compa at the time of his death, January 24, 1916, was born in Sandgate, ne Bennington, Vt., on February 22, 1858. He received his ear education in central Wisconsin and when he was fourteen years of a started work in a country printing office, later becoming half owner o machine shop. In 1878 he removed to Colorado and ran a locomoti on the Denver and Rio Grande R. R. He was soon made a roun house foreman and later assistant superintendent of motive power.

Mr. Hill had a great fondness for journalism and in 1885 founded the *Daily Press* of Pueblo, Colo. At this period he contributed a number of articles to *Locomotive Engineering*, published in New York by a company which also published the *American Machinist*, and in 1888 he was invited to come to New York and take charge of *Locomotive Engineering*. Shortly after this he associated himself with Angus Sinclair, purchased the journal from its owners, and undertook to carry it on as a separate publication. In 1896 he sold his interest in *Locomotive Engineering* and purchased the *American Machinist*. He later bought other publications, including *Power*, *Engineering and Mining Journal*, *Engineering News*, and *Coal Age* and organized the Hill Publishing Co. to carry on these various publications. The printing and publication of the company's magazines was all carried out in one building, which was completed in 1914 and was planned and built not only to suit the convenience and economy of the printing and publishing business but also to provide for the safety, comfort and health of the army of workers housed in it.

Mr. Hill was vice-president of the Machinery Club, and a member of the Engineers' Club of New York, the Railroad Club, and the Campfire Club. He became a member of this Society in 1913.

JOHN W. HILL

John W. Hill was born in Liverpool, England, on July 12, 1865, and received his early education there. From 1880 to 1884 he served an apprenticeship as machinist with Daniel Adamson & Co. and from 1890 to 1899 he was at the Watervliet Arsenal, with E. D. Leavitt, of Cambridge, Mass., and the General Electric Co. as draftsman and designer. After leaving the General Electric Co. he was superintendent of both the Steamobile Co., at Keene, N. H., and of the Roller Bearings Co. He was later employed with the Maxwell-Briscoe Co. in charge of the department of tooling up machines for rapid jig construction of automobiles. In 1911 he entered the employ of the Bantam Anti-Friction Co. as mechanical and sales engineer in charge of the Detroit office and the northwestern territory.

Mr. Hill was a member of the Society of Automobile Engineers, and a member of this Society since 1899. He died at his home in Detroit on February 12, 1916.

FREDERICK W. HOLMGREN

Frederick W. Holmgren, elected to membership in this Society in 1915, was born December 29, 1891, in Brooklyn, N. Y. He re-

ceived his early education in the Brooklyn publ
tended the Manual Training High School for a tin

After leaving school, he took an evening cours
Polytechnic Institute, working in the projectile c
E. W. Bliss Co., Brooklyn, during the day. He r
of M. E. from that institution in June, 1914.]
the E. W. Bliss Co. for a time after his graduation, r
a partner in the Berggren & Pearson Machine (
with which concern he was affiliated at the ti
December 29, 1916.

AMOS GRANVILLE HOSMER

Amos Granville Hosmer was born in Hubba
October 2, 1861. He received his early education i
and at the age of twenty learned the machinist's tr
F. Copeland, at Sterling, Mass.

He later became connected with the Arnold Pri
Adams, Mass., as mechanical engineer, where he r
For the following eight years he was at Clinton,]
caster Mills, also as mechanical engineer. In 1
Lowell, Mass., where he held the position of m
tendent for the Merrimack Manufacturing Co. .

Mr. Hosmer was a member of several lodges ar
Cotton Manufacturers' Association. He became
Society in 1902. Mr. Hosmer died following an
22, 1916.

WILLIAM HENRY JAQUES

Capt. William Henry Jaques was born in
December 24, 1848, and received his early educa
of New Jersey. He entered the United States N
midshipman in 1863, graduating with honors in
tailed for active service immediately. He becan
master in 1870, and lieutenant in 1871. At var
formed duties as aide to the President, the Secre
and the Commandant of the New York Navy Ya

Between 1870 and 1874 he was assistant in
Coast Survey; from 1874 to 1878 he assisted the N
Education in technical education; in 1881-1882 h
spector of ordnance; and from 1883 to 1885 he w
secretary to the Senate Committee on Ordnan

During this time he succeeded in introducing the system of fluid compression and hydraulic forging of heavy masses of steel, and was the inventor of many improvements in the manufacture of heavy ordnance and armor and the leading exponent of employing nickel in steel.

Captain Jaques resigned his commission in the Navy in 1887 to accept a position with the Bethlehem Steel Co. as ordnance engineer. In 1894, having successfully carried out the various developments he had advised, he retired. Soon after he associated himself with Horace See, eminent engineer and architect, in general engineering and consultation in connection with the manufacture and treatment of guns, armor, and other war material. In 1895, at the request of the governor of New Jersey, he began the organization of a naval reserve for that state and was commissioned captain, holding this command until 1898, when loss of health compelled him to resign. Although he had already done his full share in bringing the ordnance and armor of the United States to a high standard of excellence, he undertook in 1897 the development of submarine torpedo boats and accepted the presidency of the Holland Submarine Boat Co. In 1909 he became president of the Hampton Water Works Co., Little Boar's Head, N. H., and in 1913 president of the Progress Manufacturing Co., Boston, Mass., which offices he held at the time of his death, October 23, 1916.

Captain Jaques was the author of numerous monographs and books on heavy ordnance, armor, torpedoes, solar radiation, etc., and was an authority on water engineering. He was one of the international jury on marine transportation and war material at the Columbian Exposition of 1893.

Besides being a member of this Society, which he joined in 1893, he was also a life member of the Society of Naval Architects and Marine Engineers, a member of the American Institute of Mining Engineers, the American Society of Civil Engineers, the Iron and Steel Institute, the Institute of Civil Engineers (Great Britain), the Institution of Mechanical Engineers (Great Britain), the Institution of Naval Architects (Great Britain), and other organizations.

HARRY DIBROW JOHNSON, JR.

Harry Dibrow Johnson, Jr., was born in South Bend, Ind., on July 29, 1882, and died in that city on November 14, 1916.

He received his early education in the public schools of South

Bend, after which he entered Cornell University,
Immediately he was employed by the Gener
Schenectady, N. Y., and after two years returned
responsible work with Studebaker Brothers
During the first seven years he progressed from m
man and assistant master mechanic, having cha
sign and construction, power-plant operation, ma
tenance, repairs, methods, piece prices, factory
construction, to the position of master mechanic
director.

Mr. Johnson's varied interests included pub
general charities, as well as memberships in the
Chicago, the Indiana Club, the South Bend Cou
Chamber of Commerce. In 1912 he became a mer

CHARLES KIRCHHOFF

Charles Kirchhoff was born in San Francisco,
1853. After receiving his education in the scho
he completed his education in Germany, where he
the Royal School of Mines at Clausthal, Saxony,

Returning to the United States, he was emp
the Delaware Lead Refinery in Philadelphia u
year he became assistant editor of the *Metallurg*
following year he held the same position with the
he afterward became editor and part owner for
years. From 1881 until 1884 he was assistant e
Journal, and in the latter year became again asso
Age, this time as associate editor. In 1889 he be
chief, a position which he held until 1910, when h
ness. During the time he was in control of this
one of the leading engineering publications of th
this time he was also vice-president and gene
David Williams Co., retiring from this connectio

From 1883 until 1906 Mr. Kirchhoff was a
United States Geological Survey for the collectio
production of lead, copper and zinc. He was
on Some European Iron Districts (1910).

Mr. Kirchhoff was a member of the American
Engineers, of which society he was president fr
He also was a member of the American Iron an

Great Britain, the Verein Deutscher Eisenhuettenleute, and an honorary member of The Franklin Institute of Philadelphia. He was trustee and treasurer of the United Engineering Society of New York and in 1915 was elected a trustee of the Engineers' Club of New York. He was also a member of the Century Club. He became a member of this Society in 1882. He died at his summer home in Wannamassa, N. J., on July 23, 1916.

HAROLD B. KIRKUP

Harold B. Kirkup, elected to membership in this Society in 1914, was born on September 14, 1887. He received the degree of M. E., from Cornell University in 1912. In September, 1912, he entered the employ of the Cambria Steel Co., of Johnstown, Pa., doing general power-plant work, including steam-engine testing, indicating and valve setting. For two years preceding his death he was connected with the Ice Manufacturing Co., of New York City, as testing engineer.

Mr. Kirkup died in New York City on March 23, 1916.

ERASMUS DARWIN LEAVITT

Erasmus Darwin Leavitt, Honorary Member and Past-President of the Society, died in Cambridge, Mass., on March 11, 1916, where he lived for many years. He was born in Lowell, Mass., on October 27, 1836.

He received his education in the public schools of Lowell, and in 1852 entered the Lowell Machine Shop as an apprentice, where he served for three years, after which he was employed by Corliss and Nightingale for one year. In 1858 he was employed by Harrison Loring at the City Point Works at South Boston as assistant foreman, and had charge of the construction of the engine of the U. S. S. (flagship) *Hartford*. From 1859 to 1861 he was chief draftsman for Thurston, Gardner & Company, Providence, R. I., builders of high-class steam engines. He entered the United States Navy in 1861 and served during the Civil War, during which time he saw service on the gunboat *Sagamore* in the Eastern Gulf Squadron from September, 1861 until July, 1863, when he was promoted to the office of second assistant engineer, and afterwards was engaged in construction duty at Baltimore, Boston and Brooklyn. Two years later he was detailed to the Naval Academy at Annapolis as instructor in steam engineering. He resigned from the service in 1867 and immediately entered the practice of mechanical engineering.

Mr. Leavitt's fame as an engineer may be said to have begun with

the installation of the pumping engine at Lynn,
— a beam compound — marked an era in the e
engines throughout the world. It was officiall
Worthen, J. C. Hoadley, James P. Kirkwood, Ch
Joseph P. Davis, their final report being dated
duration of the test was 52 hours, without a sto
the duty was 103,923,215 ft-lb. per 100 lb. of
anthracite coal.

The Lynn engine was quickly followed by a pa
coupled together, but of somewhat larger size, fo
Lawrence, Mass. In the report of this test, i
96,186,979 ft-lb. of work per 100 lb. of coal was re
a combined indicator diagram, which was the first,
in this country. It was plotted by Mr. Hoadley,
caused much interest among engineers, and th
"adiabatic" in connection with it created a new
havior of steam.

Upon the recommendation of James B. Fran
draulic engineer of Lowell, Mr. Leavitt was appoi
mechanical engineer of the Calumet and Hecla M
1874, a position which he retained until 1904.
opportunities to display his powers as a designer
for pumping, air compression, general power pur
stamping.

While Mr. Leavitt was employed by the C
Company he was frequently engaged by other co
ipalities, and found time to act as consulting en
Worthington, The Dickson Manufacturing Com
lehem Steel Company — for the former compan
oped their high-duty direct-acting pumping engin
when the plant at Bethlehem was being modern
forging introduced. For pumping water for the f
steel works he designed a 3-cylinder 3-crank sim
vertical beam engine to drive pumps designed an
pany. For the Dickson Company he designed a
30-in. × 60-in. × 60- r.p.m. engines for the Wash
American Woolen Company at Lawrence, Mass.
continuously in operation since the spring of 188

He designed the first engines used for the c
Brooklyn Bridge; engines, boilers and other ma
Callao Mining Company of Venezuela; three imm

ing engines for the City of Boston, one being of 75 million gallons capacity in twenty-four hours; a pumping engine of 15 million gallons capacity for the Louisville Water Works which upon being tested for six days and six nights without stopping surpassed all previous records for economy in steam consumption; a large engine for the Cambridge Water Works; two engines for the New Bedford Water Works; a large engine for the Boston Water Works, etc.

After 1888 he made frequent visits to Europe, where his fame had preceded him. He became well acquainted with the leading engineers in several countries, among these being Professor Riedler of Berlin, from whom he acquired the right to use the Riedler pump valve and gear in this country.

Of Mr. Leavitt as a designer, it can be said that he did more than any other engineer in this country to establish sound principles and propriety of design. He appreciated the importance of directness and the absence of ornamentation in strictly utilitarian designs, and he firmly believed that beauty in machine design came from propriety. He bore the same relation to good taste in the design of heavy machinery that William Sellers did in the design of machine tools. He was among the very first engineers in this country to appreciate the importance of weight in machinery, and his view has amply been vindicated by the present status and tendency of design.

Mr. Leavitt received the degree of Doctor of Engineering from the Stevens Institute of Technology in 1884, and was the first recipient of this degree from the Institute. He was not only an original member of The American Society of Mechanical Engineers, but was made an Honorary Member on January 12, 1915; he served as a Vice-President in 1881–82 and as President in 1883, declining a second term in the latter office. He was a member of the Institution of Mechanical Engineers of England for thirty-three years, and was made an Honorary Member on February 18, 1916. He was a member of the following societies: Institution of Civil Engineers of Great Britain; American Society of Civil Engineers; American Institute of Mining Engineers; Boston Society of Civil Engineers (Honorary, 1908); American Society of Naval Engineers (Honorary); British Association for the Advancement of Science (Life Member), The Franklin Institute, and the New England Water Works Association (Honorary, 1906). He was a Fellow of the American Academy of Arts and Sciences.

He was for many years on the Visiting Committee of the Engineering Department of Harvard University and of the Observatory.

In general business Mr. Leavitt did not mak
but he was a director for many years of the Har
of Cambridge, and was very much interested in t
Men's Christian Association. He was also a r
having charge of the construction of the new
between Boston and Cambridge.

EDWARD I. LEIGHTON

Edward I. Leighton was born in Birmingham
8, 1850. He moved to this country with his pa
years old, settling in Cleveland, where he attend
Prior to 1880 Mr. Leighton was employed in the
land. In that year, with Fred W. Bruch, he s
Punch & Shear Works Co., operating under the
ton & Bruch. About 1890 Mr. Leighton pu
interest and later the business was sold. Alt
retired from business 15 years, he had been dire
his death in the following concerns: Acme Mach
City Live Stock & Fair Co., the Van Dorn & I
Dorn Electric Tool Co., and the Reliable Machi

Mr. Leighton became a member of this Socie
at St. Augustine, Fla., on February 26, 1916.

CLIFFORD E. LIPE

Clifford E. Lipe, who was elected to membe
in 1914, was born December 23, 1887. He secu
tion in the public schools of Syracuse, N. Y., and
University, being graduated from the school of
university in 1911.

On a year's trip abroad he studied the auto
an engineering and manufacturing standpoint, a
immediately engaged to look after the engineerin
Lipe Gear Co. He became chief engineer of tha
of experimental engineering, including develop
transmission devices, testing accessory devices b
pany by designers and inventors, and designin
company's line of four-speed transmissions.
signer with the C. E. Lipe Manufacturing Co.
variety of special devices, tools, jigs and fixture
automatic machine for drilling and reaming soli

He became vice-president of the Brown-Lipe Gear Co. and was also a member of its board of directors, being instrumental in dictating the business policy of that company.

Mr. Lipe died on February 7, 1916.

BRUCE C. MCALPINE

Bruce C. McAlpine was born on June 2, 1872, at Pierceton, Ind. He received his early education in the high schools of Peoria, Ill., and Charlotte, Mich. From the latter he entered the Michigan Agricultural College, and was graduated from the mechanical-engineering course in 1905.

He entered the employment of George D. Walcott & Son, builders of machine tools at Jackson, Mich., as draftsman, and later became successively chief draftsman and general manager. Owing to changes in the ownership of the Walcott firm, he moved to Detroit, where he was engaged in the design of special tool equipment. In May, 1913, he became chief draftsman and mechanical engineer for the Frost Gear & Forge Co., of Jackson, Mich., which position he held at the time of his death.

He became a member of the Society in 1914. He died on December 27, 1916.

JOHN E. MCINTOSH

John E. McIntosh was born at Cayuga, N. Y., on February 17, 1858, and died on September 17, 1916. He was educated at Cayuga, at St. John's School, Manlius, N. Y., and at Columbia University. His invention of a flywheel governor made possible the beginning of the McIntosh & Seymour Co., which he founded about 1885, and of which he was president until his retirement from business a few years prior to his death. He had great ability as a practical mechanic and also as a boat builder, having built several large boats for his own use.

Mr. McIntosh was a member of the Citizens' League of Auburn, the City Club of New York, City Club of Auburn, Owasco Country Club, Technology Club of Syracuse, Atlantic Deeper Waterways Association, Auburn Chamber of Commerce, and was National Chancellor of the Chamber of Commerce of the United States. He also served one term as Mayor of the City of Auburn. He became a member of this Society in 1901.

ROBERT COCHRAN MCKINNE

Col. Robert Cochran McKinney was born January 20, 1852, and died on October 3, 1916. moved to Cincinnati, where he attended the publ ward High School until 18 years of age. In 187 University, taking a partial course in mechanic lowing this he was employed at Hamilton, O., in t office of a company manufacturing steam pump

In 1877 Mr. McKinney became associated Works. Within two years he was elected secret and a little later treasurer and general manage management the company grew until the busi and capitalized at $2,000,000. Shortly after thi sary to secure additional property, the plant an Cope & Maxwell Manufacturing Co., whose p pumps, were bought by the Niles Tool Works Co.; and business of the Cope & Maxwell Co. were poration and became a part of the International

In 1898 the Niles Tool Works Co. purchased Machine Tool Co., of Plainfield, N. J., the pu mented by options on the works of Bement, N delphia, Pa., as well as the Philadelphia Engin company thus created is known widely as the Nil and Colonel McKinney was its president. Other made by this company, including the Pratt & W ford, the Bertram Co., of Canada, and the Ridg Ridgway, Pa.

Colonel McKinney was a member of the U Engineers' and Cornell Clubs of New York, th Hartford, and the Queen City Club of Cincinna of the Machinery Club of New York. He beca Society in 1890, serving as manager from 190 president from 1905 to 1907.

FRANK BARTLETT MCSOLEY

Frank Bartlett McSoley, who joined the S born at Providence, R. I., on December 11, 18 early education in the common schools and was Providence Technical High School, where he too cal, electrical and steam engineering course. He from the Rhode Island School of Design, where he specialized in mechanical and steam design.

In 1905 Mr. McSoley entered the service of the Narragansett Electric Lighting Co., of Providence, where he had full charge of the service department and of output computation of generating and sub-stations, also the estimating of construction and extension costs. He was, in addition, personal assistant to the general manager.

He was much interested in wireless telegraphy, and delivered a number of lectures on this subject. He organized the Rhode Island Wireless Telegraphic Association, of which he was first president. He was later made chairman of the advisory board of the association.

Mr. McSoley was a member of the National Electric Light Association, the American Institute of Electrical Engineers, and The Institute of Radio Engineers, and was also very active in the boy scout movement. He died at his home in Providence, R. I., on August 15, 1916.

WILLIAM L'E. MAHON

William L'E. Mahon was born in Detroit, Mich., June 19, 1861. His education was secured at the University of Michigan, succeeded by a course at the Massachusetts Institute of Technology. He specialized in marine-engine-construction work, beginning as an apprentice in the Dry Dock Engine Works in Detroit. He then became connected with the Frontier Iron and Brass Works, in the capacity of mechanical engineer and chief draftsman, later acting as assistant superintendent on construction of heavy marine engines with the same works, and subsequently with the Brown Hoisting Machinery Co., of Cleveland, O., continuing with this company for several years.

Mr. Mahon was also connected for several years with the New York office of the Taylor Wharton Iron & Steel Co., of High Bridge, N. J., later representing them in the Pacific Northwest. It was while engaged in work with this company, with headquarters at Butte, Mont., that he died, October 6, 1916, at Ogden, Utah.

Mr. Mahon was a member of the Technology Club of New York and became a member of this Society in 1889.

WILLIAM KNOX MILLHOLLAND

William Knox Millholland, for many years distinguished as a machine-tool designer, died October 9, 1916. He was born in Baltimore, Md., in 1856 and educated in the public schools.

In 1874 he started his mechanical training
drafting room and machine department of F
Baltimore, Md. He remained in their emplo
which time he completed an evening course in
of Arts and Design. In 1881 he became superi
Rivet & Machine Co., Cuyahoga Falls, O., wh
built special machine tools and brought out se
1887 he entered the employ of the M. C. Bulloc
of Chicago, Ill., as tool maker and designer an
company until 1893, when he became superinte
Whitcomb Co., designing and building coal-cutt
ting machinery, etc. In 1898 he became sales
holt Machine Co., Madison, Wis., in which wo
1906, when he organized the International M
dianapolis, Ind., and became its secretary. In
and became president of the W. K. Millholla
dianapolis, Ind., in which he was actively intere
his death.

Mr. Millholland was elected to membership i

AUSTIN D. MIXSELL

Austin D. Mixsell was born in Easton, Pa.,
and received his education in the schools of Ea
Charter School in Philadelphia.

In 1892 he went to Bethlehem and was em
office of the Lehigh Valley R. R. In 1898 he e
the Bethlehem Steel Co., working in the gen
office and in the sales department, and later was
tive of that company in New York City. In 1
general sales agent of the Company and in 191
president. At the same time, he became a me
directors of the company. He was also presi
& Harvey Machine Co. which became a subsidi
Steel Company.

Mr. Mixsell was a member of the Bethlehe
the Union League Club, Philadelphia, the M
Philadelphia, the Engineers' Club, New York,
New York, the American Society for Testing Ma
Iron and Steel Institute (as an executive membe
Steel Foundries Society. He became a memb
1913. He died at his home in Bethlehem on Ja

GEORGE MEREDITH PEEK

George Meredith Peek, a member of this Society since 1892, was born at Richmond, Va., on September 29, 1870. He received his preparatory education in the home schools and at the age of sixteen began his business life as an apprentice in the machine shop of the Baltimore & Ohio Railroad at Baltimore. In 1888, as traveling electrician for the Baxter Electric Motor Co., he had charge of their exhibit at the Milwaukee Industrial Exposition. From 1888 to 1891 he was engaged in machine work and general drafting with the Richmond Locomotive Works and also in the drawing room of the Newport News Shipbuilding and Dry Dock Co.

In 1890 Mr. Peek entered the University of Virginia at Charlottesville, remaining there for six years, during which period he received degrees in both mechanical and civil engineering and became an instructor under Prof. William H. Thornton. From 1896 to 1898 he occupied the chair of civil and mechanical engineering at the University of Arkansas.

After this extended period of study he engaged in consulting work for a year and then entered the employ of the Pelton Water Wheel Co. in New York as engineer, designing and installing water-power plants. While with this company he designed and installed plants in the United States, Mexico, Canada and Spain, preëminent among which were the Animos Power Plant in Colorado and the power plant for the Companie General de Asfeltos y Portlant in Spain. Mr. Peek also designed the motive-power equipment of the Niagara Falls pumping plant and designed and erected the regulating apparatus.

In 1910 he entered the employ of the St. Louis Water Department and devoted his energies to many improvements in the service. He was appointed engineer-in-charge of the construction branch, which position he held up to the time of his death which occurred on May 2, 1916.

HARRY SHAFER PELL

Harry Shafer Pell, elected a member of the Society in 1916, was born in Lykens, Dauphin County, Pa., on July 23, 1846. He received his early education in the public schools and in 1860 entered the employ of the Lykens Valley Railroad Co. as an apprentice machinist and blacksmith. From 1871 until 1873 he was master mechanic and later engineer for the Vulcan Iron Works, and from 1873 until 1881 he was shop foreman for Tolten & Co., of Pittsburgh. He was also engineer for the St. Louis Ore and Steel Co., in the machinery

and foundry business at Minneapolis, superint
the shops and all construction work for Jas. P
Castle, Pa., and engineer and superintendent f
of Barberton, O., manufacturers of the Stirling
the engineering department, shops and erection,
purchasing of materials. For a few years previc
Pell had been chief engineer of the water-tube-l
the Erie City Iron Works.

He died at his home in Erie, Pa., on March 1

WILLIAM ALFRED PERRY

William Alfred Perry, whose death occurred
York City on February 16, 1916, was born in B
1835. He received his early education at Mr. H
in Brooklyn and was graduated from Columbia C
following year he became a clerk in the firm of He
and 13 years later became first a partner and la
the same firm. He was also a director in the Un

The first propeller ferry boat with guards th
Bay was built under his direction by the firm of
of Philadelphia, in 1867.

Mr. Perry was a member of the University,
neers' Clubs. He became a member of this Soci

ALFRED WATERS PROCTOR

Alfred Waters Proctor was born in Needha
1878. Having from early youth an inclination
chanical, he determined to fit himself for mechani
in 1899 entered the Massachusetts Institute c
leaving the Institute, he acquired his first experi
designer and draftsman in the plants of the Bla
and the Burton Electric Smelting Co., and with
architect, all in Boston. He was later employed
chine tools with the Western Electric Co., and in
pyrographic tools with the J. G. Tyssowski Co., (
of his time was devoted to inventions and the d
chinery, for the manufacture of which he operat
in Washington, D. C.

From 1907 to 1915 he maintained an office
during which time he made a number of specia

cluding tests on inventions of pumps, tests with pumping and engine apparatus and tanks to determine the highest efficiency of differently-shaped siphon bowls, tests on speed of type bars for the Royal Typewriter Co., reports on inventions for the Crown Cork & Steel Co., chronograph tests on shoe machines for the Reece Button Hole Machine Co., and ball-bearing measurements for the Hess-Bright Manufacturing Co.

Mr. Proctor was for seven years an examiner in the United States Patent Office, and during that time attended and was graduated from Columbia University Law School, and was admitted to practice as a lawyer by the Court of Appeals of the District of Columbia and by the Supreme Court of the United States. Because of his peculiar qualifications, which now included a thorough training in the theory and practice of engineering and a wide knowledge of patents and patent laws, his services were sought as an expert in patent litigation. His success in this capacity resulted in his being retained in some of the most important patent cases before the courts in recent years, and his standing with the members of the patent bar and the confidence and respect with which the judiciary regarded his work were such that in several cases he was invited to sit with the court as a technical adviser. It had been with some reluctance, however, that he relinquished his active work as engineer, having always a preference for actual constructive work, and he continued to act frequently as a consulting engineer.

Mr. Proctor was elected to membership in this Society in 1916, and was also a member of the University Club, the Knickerbocker Field Club, and the Young Men's Republican Club, all of Brooklyn, and of the Technology Club of New York City. His death occurred on September 10, 1916.

ALASTAIR ROSS

Alastair Ross, who became a member of this Society in 1915, was born in Aberdeen, Scotland, on July 1, 1888. He received his early education in the high school, at Gordon's College at Aberdeen and at the West of Scotland Technical College in Glasgow. He served a four years' apprenticeship with J. Abernethy & Co., engineers, in Aberdeen, in fitting and erecting marine engines and in general mill work, and spent one year in the drafting room. From 1910 to 1911 he was a draftsman with the Harvey Engineering Co., in Glasgow, sugar machinery specialists, and left this firm to become chief draftsman for Catton, Neil & Co., in Honolulu, Hawaii. In 1913 he be-

came head of the engineering department of Fi
Co., of Manila, Philippine Islands, and the next y
chief engineer for The Guantanamo Sugar Co. He
1915, to take part in the War, and obtained a co
Lieutenant in the Royal Flying Corps. He was k
accident in Catterick, Yorkshire, on January 17,

HENRY FISLER RUGAN

Henry Fisler Rugan was born in Philadelphi
died in New Orleans, September 3, 1916. In
moved to Terre Haute, Indiana, where he recei
the public schools, later completing his studies a
Logansport, Ind. He left college to become an e
Haute and Indianapolis Railroad in Terre Hau
machinist, remaining with the company until 18
advancing to the position of foreman of one of tl

After two years spent in various parts of the
several railroads, he returned to Terre Haute to
instructor in the Rose Polytechnic Institute. H
Institute for nearly three years, leaving it to a
foreman of the railroad shops of the Texas and
Big Springs, Texas. He continued with this
advancing to a position where he had charge of
located in Longview and Marshall, Tex., and
He then left railroad work and became identi
industry as chief engineer and superintendent on
tions of Louisiana.

In 1895 Professor Rugan accepted the posi
mechanic arts at Tulane University of Louisia
Here he spent the last 21 years of his life. He w
the grade of assistant professor, later to that o
and finally to that of professor of mechanic arts.

He spent the year 1898–1899 in Europe, most
chester University, where, in collaboration with
of that institution, he conducted extensive rese
growth of cast irons after repeated heatings. Th
were published in the *Journal of the Steel and*
A further paper in the same journal in 1912 wa
of this work. Professor Rugan was also a contr
sion of similar problems in this Society.

He was devoted to the Masonic order, and received many honors from that organization. He became a member of this Society in 1900.

WILLIAM BARKER RUGGLES

William Barker Ruggles was born in Bath, N. Y., on December 17, 1861. He received his early education in Bath, and was graduated from Cornell University in 1883. He then became associated with the West Shore Railroad, working at Frankfort, N. Y., and elsewhere, and later with the American Casualty Co. He was the inventor of the Ruggles-Coles double-shell dryer and in 1893 founded the Ruggles-Coles Engineering Co., of which he was president up to the time of his death. He was also president of the Novella Cement Co., Niagara Cement Co., and a director of the Buffalo Potash and Cement Corporation.

Mr. Ruggles was a member of the Engineers' Club, the Machinery Club, the Psi Upsilon and Cornell University Clubs of New York, and a trustee of Trinity Church, of Bayonne, N. J. He became a member of this Society in 1905. He died at his home in Bayonne, N. J., on January 23, 1916.

PER H. SCHEDIN

Per H. Schedin, elected a member of this Society in 1913, was born in September, 1871, in Stockholm, Sweden. He received his education in the Technical School and College of Mining in Stockholm, and his mechanical training from the Stridsburg Saw Works, Trolhattan. From 1891 to 1893 he was chemist and foreman in the open-hearth department of the Gullofors Steel Works, Sweden. In January, 1894, he accepted a position with the Midvale Steel Co., Philadelphia, Pa., starting work in the machine shops and rising to the position of chief draftsman and designing engineer.

Mr. Schedin was a member of the Engineering Society of Western Pennsylvania and of the Midvale Engineering Society. He died at his home in Nicetown, Philadelphia, Pa., on January 22, 1916.

FRANK EDSON SHEDD

Frank Edson Shedd was born in Sharon, N. H., July 18, 1856. He attended the Conant High School of East Jaffrey, N. H., and was graduated from Dartmouth College in 1880. After a year of teaching as principal of a high school, he was in the service of the United States Coast and Geodetic Survey for a year, his work being on the charting of the coast of Maine.

In 1882 he left the government service to take up civil engineering in Lowell. In 1886 he had charge of the erection of the Washington Mills at Lawrence, Mass., at that time one of the largest mill-construction propositions that had then been undertaken, and later owned by the American Woolen Co. In April, 1887, he became a member of the staff of Lockwood, Greene & Co., designers of the Washington Mills, and two years later was made first assistant to Stephen Greene, then the sole member of the firm. On January 1, 1901, upon the incorporation of Lockwood, Greene & Co., Mr. Shedd became a director and the vice-president of the firm, both of which positions he held until his death, September 22, 1916.

Mr. Shedd was a civil engineer of high standing, had designed many large mills and hydraulic plants in various parts of the United States and Canada, and was considered one of the leading authorities in this country on hydraulic developments.

He was a member of the American Society of Civil Engineers, the Boston Society of Civil Engineers, and a member of this Society since 1906.

GARRETT W. SIMPKINSON

Garrett W. Simpkinson was born in Cincinnati, O., on August 17, 1860. He received his early education in the schools of Cincinnati and later received private instruction in certain branches of engineering.

He served four years as pattern maker and four years in the drafting department of the Lane & Bodley Co. He then entered the employ of The Stilwell & Bierce Manufacturing Co., Dayton, O., in charge of the drafting and pattern departments. Several years later he returned to Cincinnati to assist in building the first cable railway in that city, designing track and curve constructions as well as the mechanical details of the driving stations. After the completion of this work he returned to the Lane & Bodley Co. and for a time had charge of their drafting department and pattern shop. In 1888 he became associated with Bert L. Baldwin, Mem.Am.Soc. M.E., in designing and constructing inclined-plane and electric railways and plants in and near Cincinnati, and made a specialty of machine-shop and foundry design and construction, as well as structural and architectural engineering. He became a member of the firm of Bert L. Baldwin & Co. in 1900. For several years Mr. Simpkinson instructed evening classes in mechanical drawing and mechanics of engineering at the Ohio Mechanics Institute, Cincinnati, O.

He was a member of the Cincinnati Engineers' Club and a member of this Society since 1912. Mr. Simpkinson died at his home in Cincinnati, O., on January 22, 1916.

ELMER NEILL STACY

Elmer Neill Stacy, whose death occurred on September 9, 1916, was born on December 11, 1876, in Minneapolis, Minn. He received his early education at the district school and the central high school, Minneapolis, and was graduated from the College of Engineering, University of Minnesota, in 1907. Before entering college he spent about three years in the drafting department of the Plano Manufacturing Co., at West Pullman, Ill.

After graduation he was employed by the Minneapolis Threshing Machine Co. as draftsman and designer, and did considerable work on the testing floor. About June, 1908, he associated himself with the Decarie Incinerator Co. as assistant to the general manager, and in March, 1909, he became general manager of the company, directing the design and construction of refuse-disposal plants. In April, 1914, the Stacy-Bates Co. was organized for the exclusive sale of the Decarie Incinerator Company's plants, with Mr. Stacy as president, who performed the same work as with the Decarie Company.

Mr. Stacy was a member of the Sigma Xi and the Tau Beta Pi Fraternities and became a member of this Society in 1911.

FRANKLIN MCMILLAN STANTON

Franklin McMillan Stanton was born in New York City on May 23, 1865, and died on September 12, 1916. Following a course of public-school instruction, he entered Columbia University, graduating from the School of Mines in 1887. He immediately took up his profession, working as a surveyor and assayer for about two years, when he decided to embark in the mining field on his own account. The undertaking proved successful and led to his long connection with the Superior companies.

After a few years he accepted a position with the Atlantic Mining Co., soon advancing to the post of superintendent, in which capacity he served for 23 years, improving mining methods and creating what was considered one of the best organizations in the mining field. Because of failing health, Mr. Stanton retired in 1910, visited most of the European countries and availed himself of the opportunity to study the mines of the Continent. In 1914 he became treasurer and

director of the Mohawk Mining Co., the Wolv
Co., which controls some of the best territory i
copper district, and the Michigan Copper Mini
treasurer of the White Pine Extension Copper C
been organized.

In addition to his activities in the mining fi
identified with several other enterprises. He
director of the Ft. Mountain Talc Co., which
northern Georgia where talc abounds, and was
First National Bank, Houghton, Mich., the Ohi
road, which penetrates the coal region of thos
Range Co., etc.

A large number of professional, social and p
included Mr. Stanton in their membership. I
the American Society of Civil Engineers, Americ
American Forestry Association, American Insti
neers, American Revolver Association, Americ
American Society for Prevention of Cruelty t
Museum of Natural History, Burns Society, A
Association, Psi Upsilon Fraternity, Lambda C
Continental Guards, The Navy League of the U
Infantry N. G. N. Y., Regimental Mess, Seven
and Active League, Society of Upper Eighties of
St. George's Society, Sons of the Revolution,
Wars, United Engineering Society, Columb
Houghton Life Infantry, Horticultural Society
Superior Mining Institute, Michigan State Rifle
Security League, National Rifle Association an
versity Athletic Association. He was also a me
social clubs. He became a member of this Soci

THOMAS I. STEPHENSON

Thomas I. Stephenson was born in Lenoir Ci
5, 1863, and received his early education in the
He began work with the Knoxville Iron Comp
appointed secretary in 1891, vice-president an
1895, and president and general manager in 1
president of the Cross Mountain Coal Co. in 19

Mr. Stephenson became a member of this So
at his home in Knoxville, Tenn., on January 18

MAX M. SUPPES

Max M. Suppes, elected to membership in this Society in 1890, was born at Johnstown, Pa., February 18, 1856. He received his education in the common schools at Johnstown, and learned the machinist's trade in a machine shop there.

He went to Troy, N. Y., in 1879 as master mechanic for the Rensselaer Iron and Steel Works (later the Troy Iron and Steel Works) where he was associated with Capt. Robert W. Hunt, and helped develop the first automatic rail mill for rolling T-rails. From Troy he went to Braddock in December, 1887, as assistant master mechanic of the Edgar Thomson Works. He remained at Braddock about a month, returning to Johnstown on February 1, 1888, to assume the duties of master mechanic of the Johnson Steel Street Rail Co., which was being built by A. J. Moxham and Tom L. Johnson. In the fall of the same year he became manager of the rolling-mill department of the plant and assisted very largely in the development of the method of rolling girder rails.

Early in 1894, when Messrs. Moxham and Johnson decided to build a steel plant at Lorain, O., and move the girder-rail mill at Johnstown to Lorain, Mr. Suppes was appointed general manager of the new plant and put in charge of its construction. Active construction operations began at Lorain in July, 1894, and the plant was completed in a period of ten months. The plant was first known as the Johnson Co., then as the Lorain Steel Co., was later taken over by the Federal Steel Co., and subsequently became The National Tube Company of Ohio. Through all these changes and the continued expansion of the plant, Mr. Suppes continued as manager until his death, March 27, 1916.

He was a man of very marked engineering ability. He was of an inventive turn of mind, evidence of which is amply given in the large number of patents issued to him, many of which are in use in various steel mills. In 1883 he installed the first tables in front of rolls on a rolling mill. He developed the first arrangement for rolling steel out of heating furnaces. Other rail-mill and blast-furnace improvements originated by him are well known, such as his weighing device for rolling mills, leading spindle and coupling for rolling mills, roll-adjusting mechanism for rolling mills, stock-distributing and collecting apparatus for blast furnaces, etc. His expansion joint for engines is used on a great many reversing engines. One of his later developments was a stock-handling and storage arrangement for open-hearth-furnace plants.

He was a member of the Iron and Steel Institute, the American Iron and Steel Institute, the Masonic Order and the Royal Arcanum.

JOHN EDSON SWEET

Prof. John Edson Sweet died at his home in Syracuse on Monday, May 8, at the age of eighty-three. An appreciation of Professor Sweet is published elsewhere in this volume, and the facts of his life are only briefly summarized here.

He was born in Pompey, near Syracuse, N. Y., October 21, 1832, and his only opportunity for schooling was in the district schools of that time. As a boy he worked on a farm, and later became a carpenter's apprentice. He was an architect and builder in the South for several years, but on the outbreak of the Civil War in 1861 he returned to the North. He was an inventor and mechanical draftsman from 1861 to 1873, when he was appointed Director of Shops in Sibley College, Cornell University. In 1879 he left Cornell University to establish the Straight Line Engine Co. in Syracuse, of which he was president from its inception until his death.

Professor Sweet was one of the founders, and was the third president of this Society. In 1914 the John Fritz Medal was awarded to him "for his achievements in machine design, and for his pioneer work in applying sound engineering principles to the construction and development of the high-speed engine," and in the same year Syracuse University conferred upon him the honorary degree of Doctor of Engineering. Thus he received honors in recognition of his great ability as an engineer. But his great life work consisted in his influence over men to cause them to think straight and to live honorably. All who were his students at Cornell or elsewhere — for he was a teacher all his life — have always felt for him the highest respect and deepest affection; and his death brings to them all a sense of personal loss. The world can so ill afford to lose those rare men of sane brain, sound judgment and big heart who love to serve their fellow-men.

GEORGE W. K. TAYLOR

George W. K. Taylor, who joined the Society in 1907, was born in New York City on December 18, 1856. He received his early education in the public schools of Brooklyn and later attended Cooper

Union, receiving a B. S. degree in 1877 and a degree in mechanical engineering in 1907.

From 1874 to 1893 he was employed by the Eaton, Cole & Burnham Co. at New York City and Bridgeport, Conn., in charge of the designing, manufacturing and superintending the production of special valves, fittings, etc., especially adapted to high-pressure steam work for power stations. In 1893 he formed a partnership with Mr. McMann, specializing in designing, manufacturing and superintending work for power plants. In 1894 he equipped the steam-power plants of the Baltimore, Md., traction system, and the Columbus, O., Edison Electric Light Co. In 1910 the firm was incorporated as the McMann & Taylor Co., designing and manufacturing wrought- and cast-iron pipes, boiler tubes, fittings, valves, tools, etc., and all goods pertaining to steam, water, gas, etc.

Mr. Taylor was a member of the Machinery Club of New York, the Society of Old Brooklynites, the Chamber of Commerce, the Merchants' Association, the Old Union League Club of Brooklyn, director of Greenwich Bank, trustee of St. John's Methodist Episcopal Church of Brooklyn, and was a 32d degree Mason.

He died at his home in Brooklyn, March 5, 1916.

DAVID THOMPSON

David Thompson, a member of the Society since 1905, was born at Castlemaine, Victoria, Australia, on December 5, 1865. He received his early education in the state and public schools and in the School of Mines. He served an apprenticeship in the Castlemaine Foundry of the Thompson Co., manufacturers of mining and general machinery, which was founded by his father. From 1882 to 1887 he was engaged in making detail drawings for all classes of mining machinery, engine boilers and pumping gears, and in 1887 became assistant works manager at the Castlemaine Foundry. Since 1891 he had been managing partner of the Foundry, the largest privately owned ironworks in Australia. In 1915, he completed the first railway locomotive that was built in that part of the country.

Mr. Thompson was very much interested in the welfare of the working classes and sat on all wage boards in Melbourne in connection with the iron trade and was largely instrumental in fixing the rates of wages. He was also a member of the Chamber of Manufacturers and acted on various committees associated with that body.

His death, which was the result of an accident in the works at Castlemaine, occurred on February 6, 1916.

KENNETH TORRANCE

Kenneth Torrance was born in Brooklyn, N. Y., in 1863. He went to Stevens High School for one year and then to Stevens Institute, graduating with the class of 1884. He remained an active alumnus throughout the rest of his life and founded an enthusiastic association of Stevens alumni in Schenectady.

After leaving college he became connected with the Worthington Pump Co. and later accepted a position with the Brooklyn Water Works as superintendent of the Ridgewood and all the Long Island pumping stations. In 1906 he joined the General Electric Co., Schenectady, N. Y., to take charge of its power stations, pumping stations, water, steam and compressed-air systems, heating, etc. He carried out the reconstruction of the entire heating system for this works and was responsible for the design of all heating for the many new buildings erected at this works during his connection with the company. He carried out extensive additions to the power plants which resulted in very successfully meeting most extraordinary requirements in connection with emergency power supply, steam for testing, etc. He was actively connected with the development of the steam-flow meter.

Mr. Torrance had a rare gift for handling men. In Brooklyn and later in Schenectady he held the loyalty and respect of those under him as few men do. He had a host of friends, and wherever he went he made more.

He was elected to membership in this Society in 1885 and was also a member of the American Waterworks Association, the Society of Engineers of Eastern New York, the Schenectady Stevens Club, the Delta Tau Delta fraternity, the Mohawk Club and the Mohawk Golf Club.

He died on September 13, 1916, at Mount Kineo, Me., where he had gone to convalesce after a severe illness.

WILLIAM BAYLY UPTON

William Bayly Upton was born in San Francisco, Cal., on October 30, 1856, and died in Alameda, Cal., on July 30, 1916. He received his education in the home schools and served his apprenticeship at the Union Iron Works in San Francisco. He was associated with cable-road construction in San Francisco and Kansas City up to

1892, when he was called to Washington, D. C., for similar work with the Capital Traction Co. He served later as engineer in charge of the construction of several of the important underground and overhead electric lines now under the control of the Washington Railway and Electric Co., and was engineer in charge of the joint construction by the two traction companies of the lines approaching the Union Station. Among other of his achievements in Washington was the planning and installation of the lighting and wiring system in the municipal filtration plant. He returned to San Francisco to take charge of the construction of the Argentine Pavilion at the Panama-Pacific International Exposition in 1915.

Mr. Upton became a member of this Society in 1894.

FRANK HASTINGS VARNEY

Frank Hastings Varney was born at San Jose, Cal., on September 15, 1872. His parents later moved to San Francisco where he received his public-school education and where his business life commenced.

His engineering work began in 1894, when at the age of twenty-two he became engineer of the Harbor Light and Power Co., a position which included practically all duties from lineman to manager of the 25-kw. station and lighting system. The following year this concern was absorbed by the Edison Co. and he was transferred to the new Stevenson Street plant of that company, advancing to the position of station foreman in 1898. The plant was then purchased by the San Francisco Gas & Electric Co. and he was made chief electrician in charge of their three local steam-electric generating stations — in 1900 becoming superintendent of all steam and electric stations — and the distributing system. When this rapidly growing company absorbed the Independent Co., he was made superintendent of generating stations and sub-stations, and the company later becoming part of the Pacific Gas & Electric Co., he was, in 1907, made chief engineer of operating and maintenance of steam stations, the position he held at the time of his death. It is interesting to note that in his last position he had control of an output of 90,000 kw.

Mr. Varney became a member of this Society in 1909 and was also a member of the American Institute of Electrical Engineers, the National Electric Light Association, and the Engineers' Club of San Francisco. He died at his home in San Francisco on January 21, 1916, after several months' illness.

JAMES F. WALSH

James F. Walsh was born in Cleveland, O.
He received his education in the parochial scl
College of that city.

In 1871 he began railroad work as machini
Cleveland, Columbus, Cincinnati & Indianapo
successively locomotive fireman, engineer, and
and general foreman. In 1892 he became associ
Signal Oil Co. as mechanical expert, and in 19
superintendent of motive power of the Chesape
at Richmond, Va. In May, 1910, he became g
of motive power for the Chesapeake & Ohic
active service in July, 1912.

At the time of his death, which occurred c
Roanoke, Va., Mr. Walsh was doing special me
Galena Signal Oil Co. He became a member of

WILLIAM A. WARMAN

William A. Warman was born at Latrobe,
on June 28, 1861, and received his education at

From 1878 to 1886 he devoted his time to d
following which he had six years' shop experien
railroad shops. From 1892 to 1894 he was con
well Co., of Buffalo, and the following year he n
own at Niagara Falls. In 1898 he was engaged
Manufacturing Co., and from 1898 to 1901 he
the Ritter Dental Co. During 1902 and 1903 h
Press Guard Manufacturing Co., and became
velopment of press guards for stamping presses.
for the American Tobacco Co. and in 1905 forn
which soon thereafter consolidated with the K
graving Co., with which concern he held the p
special machinery. He also did much original r
nection with internal-combustion engines and
the inventor of a number of safety devices, also o

Mr. Warman became a member of this Soci
on July 2, 1916.

HUBERT LEIGH WATSON

Hubert Leigh Watson, who died in New Lo
11, 1916, was born in Philadelphia, Pa., Septe

educated in the public schools of Hackensack, N. J., and later entered Cornell University for a course in mechanical engineering, taking his degree in 1910. The following year he pursued advanced work in gas-engine design.

For practical experience Mr. Watson began with the summer of 1909 to employ his vacation time in apprenticeships, at that time as machinist's helper in the repair shops of the Delaware, Lackawanna & Western Railroad at Kingsland, N. J.; in 1910 he worked with the American Car and Foundry Co., at Berwick, Pa., in its steel car department, and in 1912 and 1913 was employed with the De La Vergne Machine Co., New York City, in erecting and testing oil engines, also doing some special tool designing.

From 1911 to 1915 he was instructor in practical mechanics in Purdue University and was in charge of the course in gas-power-plant design and taught all sections in mechanism. In July, 1915, he resigned from Purdue University to accept a position with the New London Ship & Engine Co., of Groton, Conn.

He published an article on Accelerating Force of Reciprocating Parts in *Machinery*, August, 1916.

Mr. Watson was a member of Purdue Cosmopolitan Club, and in 1913 became member of this Society.

DON JUAN WHITTEMORE

Don Juan Whittemore was born at Milton, Vt., on December 6, 1830. After his graduation from the Bakersfield Academy, Mr. Whittemore joined the engineering staff of the Vermont Central Railway, later going with the Western Railway of Canada and the Central Railway of Ohio.

He became chief assistant engineer of the La Crosse & Milwaukee Railroad, where he served for four years, and was then made chief engineer of the Southern Minnesota Railway and assistant chief of the Western Railway of Cuba. He later returned to the La Crosse & Milwaukee Railroad and in 1863, became connected with the Chicago, Milwaukee & St. Paul Railroad, where he remained until 1910, retiring as consulting engineer of the road.

Mr. Whittemore was a past-president of the American Society of Civil Engineers, and had received the degree of C. E. from the University of Vermont and those of Ph. D. and LL. D. from the University of Wisconsin.

He became a member of this Society in 1889. He died at his home in Milwaukee, Wis., on July 17, 1916.

WILLIAM C. WILLIAMSON

William C. Williamson spent his early youth in Philadelphia, attending high school there. After leaving school he began his engineering experience by associating himself with a jeweler and watchmaker in repairing watches. This work was not especially to his liking and so he apprenticed himself to the old firm of Reaney, Neafie and Co., at that time one of the foremost engineering firms of the entire Atlantic seaboard.

In 1861 he entered the United States Navy as third assistant engineer, serving throughout the war. He resigned June 10, 1866, to engage in the engineering business with his two brothers, founding the firm of Williamson Brothers. His engineering talents soon showed in the Williamson type of clutch and frictional-geared cargo hoists. In the early eighties he introduced in a ship building at the yard of the William Cramp & Sons Ship and Engine Building Co., a steam engine to supersede the old type of hand-steering wheel for the control of ships. It was an immediate success and entitles him to be classed among the foremost naval engineers of his day.

He was prominent also in financial affairs, being one of the oldest directors of the Kensington National Bank, and director and vice-president of the Industrial Trust, Title and Savings Company of Philadelphia.

He was a member of the Naval Order of the United States, Post No. 2, G. A. R., and the Engineers' Club of Philadelphia. He was a member of the Society of long standing, having been elected to membership in 1882. He died December 2, 1916.

CHARLES J. H. WOODBURY

Charles J. H. Woodbury, one of the original members of the Society who served as manager and vice-president, died on March 20, 1916. He was born at Lynn, Mass., on May 4, 1851, and resided there during his life. He received his education in the public schools of Lynn, and from there he entered the Massachusetts Institute of Technology in the course of civil engineering, being graduated in 1873.

He began the practice of his profession in the city engineer's office in Lynn during his vacations while at college, and was later superintendent of a mill at Rockport. He became engineer of the Boston Manufacturers' Mutual Fire Insurance Co. in 1878, and later vice-president. While with the company he made numerous investigations on fire hazards in mill construction, lubricating oils and electric lighting, invented several improvements in automatic sprinklers and also reorganized the company's methods of inspection and reports upon mill property. In 1894 he became assistant engineer of the American Bell Telephone Co., holding this position until the removal of the company to New York City in 1907, when he took up private practice as a consulting engineer.

Dr. Woodbury was the author of Fire Protection in Mills, 1882, Telephone Line Engineering; The Telephone System, 1899, and Bibliography of Cotton Manufacture, 1909. Also, he wrote numerous monographs and papers on fire protection, political economy and engineering, many of which are incorporated in the proceedings of the scientific and engineering societies.

He was the originator of a number of valuable inventions, and on three occasions received awards for them. The Société Industrielle de Mulhouse awarded him the Alsacian Medal in 1883, the city of Philadelphia conferred on him the John Scott Medal in 1885, and in 1910 he received the Association Medal from the National Association of Cotton Manufacturers. In 1893 he received the honorary degree of Master of Arts from Tufts College, and was made an honorary Doctor of Science of Union College in 1906 and of Dartmouth College two years later.

Dr. Woodbury was a member of the American Society of Civil Engineers, the American Institute of Electrical Engineers, the National Association of Cotton Manufacturers, of which he was secretary and treasurer since 1894, the Society of Arts, and the Sons of the American Revolution. He was a fellow of the American Academy for the Advancement of Science, and an honorary member of the New York Telephone Club. He was a non-resident lecturer at the Massachusetts Institute of Technology and also at Cornell University and from 1886 to 1895 he was chairman of the Lynn School Committee.

To him is due the initiation of our Library. At the meeting of the Council on February 15, 1883, he moved that the Secretary be instructed to request from members circulars of manufacturing establishments and reports of engineering operations, with a view to making a catalogue of contemporaneous engineering work, to be filed properly and placed at the service of members. The motion was carried and put into effect, and the response was prompt and liberal, many of the technical periodicals contributing copies of their publications, and some sending complete bound files of their back numbers. As the outcome a standing committee on the library was appointed, which recommended the definite organization of the Library.

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NOTE

1 Names of authors and discussers, also deceased members preceding an obituary notice, are in caps and small caps. A discusser 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 titles but under their subject-matter.

[1] See THE JOURNAL, March 1917, p. 201.
[2] See THE JOURNAL, March 1917, p. 207.

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