THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 42

ST. LOUIS MEETING
NEW YORK MEETING
1920

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CONTENTS OF VOLUME 42[1]

St. Louis and New York Meetings

[1] The Society shall not be responsible for statements or opinions advanced in papers or in discussion at its meetings (C 55).

OFFICERS
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

FORMING THE STATUTORY COUNCIL
1920

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Terms expire December 1920

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Terms expire December 1921

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NOTE:—Numbers in parentheses indicate the number of years the member has yet to serve.

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JOHN R. ALLEN .... WILLIAM B. GREGORY
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SPENCER MILLER

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Appointed 1909, for supervising all research activities, collaborating with committee of kindred societies, and obtaining results of researches conducted in other countries

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ALBERT KINGSBURY (2)

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Standardizes method of making and arriving at standards themselves. Endeavors to bring about a unification of the standards of the society. Coöperates with American Engineering Standards Committee.

HENRY HESS (2), *Chairman* .... WILLIAM H. KAVANAUGH (4)
WILLIAM F. KIESEL (1) .... WILLIAM P. EALES (5)
CLARENCE F. HIRSHFELD (3)

NOTE: — Numbers in parentheses indicate the number of years the member has yet to serve.

[1] Sub-Committees are published in 1920 Year Book,

SOCIETY REPRESENTATIVES[1]

Appointed by the Council on Committees, etc., of other bodies

AMERICAN ENGINEERING STANDARDS COMMITTEE

Aims to organize and control forming of engineering and industrial standards in United States. Began work in October, 1918

Henry Hess .... Henry H. Vaughan
Fred E. Rogers

ENGINEERING COUNCIL (1920)

A joint body on which are represented at present the Founder Societies and the A.S.T.M. Deals with problems of joint interest to these societies

Chas. Whiting Baker (2) .... L. P. Alford (2)
Arthur M. Greene, Jr. (3) .... Fred J. Miller (1)
Charles T. Main (3)

ENGINEERING FOUNDATION BOARD (1920)

Administers the Engineering Foundation. Established for the Founder Societies by Mr. Ambrose Swasey, Hon. Mem. Am. Soc. M.E. Income being applied to research

W. F. M. Goss .... E. Gybbon Spilsbury
David S. Jacobus

JOHN FRITZ MEDAL BOARD OF AWARD

Awards to John Fritz Medal Annually

John R. Freeman .... Ambrose Swasey
John A. Brashear .... Henry B. Sargent

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Administers the property of the Founder Societies (A.S.C.E., A.I.M.E., A.S.M.E. and A.I.E.E.)

W. M. McFarland .... Irving E. Moultrop
E. Gybbon Spilsbury

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Administers the joint library of the Founder Societies and the U.E.S.

L. P. Breckenridge .... Henry A. Lardner
Andrew M. Hunt .... William N. Best
Calvin W. Rice

Note: — Numbers in parentheses indicate the number of years the member has yet to serve.

[1] Partial list of Society Representatives. Complete list published in 1920 Year Book; also special committees and Local Sections' Committee.

SUMMARY OF MEMBERSHIP BY RESIDENCE

UNITED STATES

Alabama	72	Nebraska	25
Alaska	2	Nevada	5
Arizona	30	New Hampshire	46
Arkansas	15	New Jersey	940
California	420	New Mexico	10
Canal Zone	4	New York	3078
Colorado	71	North Carolina	40
Connecticut	650	North Dakota	5
Delaware	100	Ohio	950
District of Columbia	190	Oklahoma	65
Florida	36	Oregon	35
Georgia	75	Pennsylvania	1475
Hawaiian Islands	20	Philippine Islands	16
Idaho	4	Porto Rico	8
Illinois	815	Rhode Island	141
Indiana	220	South Carolina	18
Iowa	40	South Dakota	5
Kansas	45	Tennessee	50
Kentucky	40	Texas	105
Louisiana	70	Utah	20
Maine	35	Vermont	36
Maryland	175	Virginia	80
Massachusetts	1078	Washington	68
Michigan	485	West Virginia	35
Minnesota	115	Wisconsin	201
Mississippi	10	Wyoming	4
Missouri	195		
Montana	15	Total	12588

FOREIGN COUNTRIES

Africa	14	India	13
Australia	21	Italy	3
Austria	1	Japan	21
Belgium	4	Java	1
British West Indies	1	Mexico	21
Canada	185	Norway	10
Central America	3	Roumania	1
Channel Islands	1	Russia	7
China	16	Scotland	6
Cuba	47	Singapore S. S.	1
Denmark	4	South America	62
Domingo	1	Spain	3
England	104	Sweden	11
Finland	2	Switzerland	6
France	37	Turkey	1
Germany	17	West Indies	6
Holland	1		
		Total	631

United States (*continued*)

Membership in United States	12588
Membership in Foreign Countries	631
Present address unknown	32
Total Membership	13251

Membership by Grades

Honorary Members	13
Life Members	102
Members	6292
Life Associates	3
Associate-Members	2789
Associates	705
Life Juniors	1
Juniors	3222
B-16 Members	124
Total Membership	13251

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 42—1920

VOLUME 42 contains an account of the activities of the Society for 1920. In it will be found the papers and addresses presented at the Spring Meeting, held in St. Louis, and the Annual Meeting, held in New York, and the principal discussion thereon.

An attempt has been made in selecting the material published to include all papers having permanent reference value. For this reason it will be found that many discussions are condensed and that less technical interest has been considered as belonging more properly to MECHANICAL ENGINEERING.

FRED J. MILLER

FRED J. MILLER was born in Ohio in 1857. He had a common high school education, supplemented by personal study and special instruction. He served an apprenticeship of four years in the machinist trade, and for a number of years worked in various capacities in machine shops.

He became a contributor to mechanical journals and finally a member of the editorial staff of a prominent machinery journal, where he remained for twenty years, the last ten years of which he spent as editor-in-chief and vice-president of the publishing company. During this time he acquired an extensive acquaintance with machinery-building establishments in Europe, as well as in America.

After retiring from journalism, Mr. Miller was for nine years general manager of a group of typewriter factories, and was then commissioned Major of Ordnance in the U. S. Army, being stationed during most of his service at the Rock Island Arsenal, where he organized and was at the head of the Civilian Service Division of the Arsenal. Afterward he was attached to the office of the Director of Arsenals in Washington, being assigned to duty at the Bethlehem Steel Works, and was then attached to the office of the Secretary of War. He was also a member of the Technical Advisory Committee of the War Claims Board at Washington.

Major Miller became a member of the Society in 1890, and has always taken an active interest in its affairs, having served on a number of important committees, and as a member of Council from 1904 to 1907, inclusive. He also served two terms as the Society's representative on the Board of the United Engineering Societies, where he was chairman of three successive committees entrusted with the work of reorganizing and perfecting the Founder Societies' business affairs and finances. He was a member of the Council at the time THE JOURNAL of the Society was started, and has always been consulted on questions relating to its development, having recently devoted considerable time to the study of the Society's publishing problems and its office methods and organization.

No. 1738

MEETINGS JANUARY-JUNE

MEETINGS OF SECTIONS

THE following is a record of the meetings which were held by the organized sections of the Society and the Providence Engineering Society, an affiliated body, during the first six months of the year. A number of the papers of more general interest were published in MECHANICAL ENGINEERING during 1920.

ATLANTA

January 27: Address by Charles M. Rogers, Rogers, Higgins Co., on Oil as a Fuel.

February 24: Building a 30,000-kw. Turbo Set, by H. E. Bussey, General Electric Co.

March 30: Ball Bearings Applied to General Power Transmission, by W. H. Holby.

May 25: Manufacture of Cotton Goods, by J. T. Wikle.

June 10: Plate dinner, election of officers.

BALTIMORE

February 27: Council meeting. Informal dinner attended by the Mayor, City Superintendent of Schools, Council, Sections Committee, Executive Committee of the Baltimore Section, representatives of the Engineers' Club and members of the Philadelphia and Washington Sections. Evening meeting on Vocational Education.

February 28: Visit to Experimental Stations and Laboratories of the United States Naval Academy at Annapolis. Luncheon at Carvel Hall.

March 3: Reinforced-Concrete Design and Construction, Past and Present, by Ernest P. Goodrich.

March 10: Railroad Grade-Crossing Elimination, by Samuel T. Wagner.

March 17: The Commercial Side of Engineering, by Commander Walter M. McFarland.

March 24: Engineering and Construction Organization for Rapid Work, by I. W. McConnell.

April 2: Business meeting.

April 16: Joint meeting with Baltimore Section, A.I.E.E. The Super-Power System, by W. S. Murray.

May 7: Joint meeting with the Electrical Institute. Some Recent Developments in Heating and Ventilation, by Prof. John R. Allen.

BIRMINGHAM

April 23: Joint meeting with Atlanta Section. Export Business in the Near East, by Prof. John R. Allen.

BOSTON

January 22: Research. Speakers, Professors John R. Allen and Arthur M. Greene, Jr.

February 3: Dinner meeting of the A.S.M.E. and A.I.E.E. Large Turbo-Generator Units from the Operator's Standpoint. Discussion opened by Farley Osgood. Engineering Research, by E. H. Colpitts.

February 24: Joint meeting with A.I.E.E. and Boston Society of Civil Engineers. A National Department of Public Works, by Prof. George F. Swain.

March 30: Eleventh annual dinner of the engineers of Boston, sponsored by the Boston sections of the A.S.M.E., A.I.E.E. and the Boston Society of Civil Engineers. Some Economic Aspects of the Treaty of Paris, by Paul Cravath. Social Engineering, by Dean Roscoe C. Pound.

May 5: Methods of Airplane Production and Possibilities in Future Commercial Aviation, by Charles H. Tavener.

BUFFALO

January 6: Oxy-Acetylene Welding, by Major C. K. Bryce.

February 24: Wage Payment, by A. L. De Leeuw.

April 6: Educating Blinded Soldiers to Lines of Usefulness, by L. W. Wallace.

May 4: Business meeting. Address by Secretary Calvin W. Rice.

CHICAGO

January 23: Fuel for Internal Combustion Engines, the Lubrication and the Motor Itself, by William F. Parish.[1] The Proposed National Department of Public Works, by Arthur L. Rice.

February 16: Joint meeting with the Western Society of Engineers. Factory Fire Protection, by Benjamin Richards.

February 28: Joint meeting with the Western Society of Engineers. The Washington Award was presented to Mr. Herbert Hoover.

March 4: Joint meeting with A.I.E.E., Electrical and Mechanical Sections, Western Society of Engineers, and Western Railway Club. Some Comments on the Present Status of Steam Railroad Electrification, by S. T. Dodd.

March 19: Dinner meeting. Beauties and Power Development of Niagara Falls, by William M. White.

April 15: Joint meeting with other engineering societies in coöperation with the Western Society of Engineers. Teaching Shop Executives, by Prof. B. W. Benedict.

April 19: Council Meeting. Joint meeting of four Founder Societies with the Western Society of Engineers and the local members of other national societies. Banquet, preceded by informal reception and followed by brief addresses by the Presidents of the four Founder Societies.

April 20: Luncheon, at which were present members of the Board of Directors of the five societies, and the Executive Committees of the local sections.

May 21: Luncheon and excursion to the works of the Western Electric Company.

May 22: Entertainment of the members of the A.S.M.E. en route to the Spring Meeting.

June 8: The St. Lawrence River Project,[2] by Horace C. Gardner.

CINCINNATI

January 15: The Evolution of Automotive Electrical Ignition, by R. C. Fryer.

February 19: Joint meeting with Engineers' Club of Cincinnati. The Engineer and the Markets of the World, Dean Herman Schneider.

April 22: Theory and Practice, Prof. John R. Allen.

[1] See MECHANICAL ENGINEERING, March, 1920.

[2] Abstract published in MECHANICAL ENGINEERING, September 1920.

CLEVELAND

January 27: Rubber-Working Machinery, by John Schaub, and Clay-Working Machinery, by A. O. Austin.

February 24: Joint meeting with Cleveland Engineering Society. A Photographic Study of the Sounds of Large Guns, by Dr. Dayton C. Miller.

March 23: Methods of Measuring Flow of Fluids, by E. G. Bailey.

April 13: Is There a Cure for Class Differences in America? by Dr. Ira N. Hollis.

April 27: Joint meeting with the Cleveland Engineering Society. Grinding Developments, Achievements and Future Possibilities, by C. H. Norton.

May 25: Joint meeting with Cleveland Engineering Society. Gas Warfare Lessons, by Frederick Pope.

June 8: The Benefits of Engineering Associations, Col. Edward A. Deeds. Getting the Most out of Life, by E. J. Cattell.

COLORADO

January 23: What the United States Engineers did in France, by Major Charles Larsen.

February 27: Some Interesting Tests on a High-Pressure Automotive Steam Boiler, by E. H. Nate.

March 26: Colorado's Development and the Engineer, by Prof. L. D. Crain.

April 23: The Engineer and the Public, by Durbin Van Law.

CONNECTICUT

Bridgeport Branch

March 29: Dinner and meeting.

Hartford Branch

January 19: The Use of Fuel Oil, particularly in Steam Production, by Ernest H. Peabody. Certain Aspects of the Fuel Question, by John P. Leask.

March 15: Power Production, by George A. Orrok.

April 12: Drop Forgings and Die Castings, by representatives from the Billings and Spencer Co., the J. H. Williams Co., and the Doehler Die Castings Co.

Meriden Branch

February 10: Methods of Measuring and Manufacturing to the Millionth of an Inch, by W. H. Weingar.

April 8: Boiler Feeding, by Stanley Brown. Motion picture, The Twist Drill — Its Uses and Abuses.

May 7: Business meeting.

May 21: Machine Switching Equipment, by E. H. Everit.

June 24: Inspection of Aeolian Co. Factory. Manufacture of Talking Machine Records, by H. L. Thompson. The Manufacture and Use of Talking-Machine Spring Motors, by E. W. Carruth.

New Haven Branch

January 21: Joint meeting with Winchester Engineers' Club. Appraisal and Valuation Methods, by J. G. Morse.

March 8: Joint meeting with Winchester Engineers' Club. Research, by Professor W. J. Wohlenberg. Coöperation of the Engineering Societies, by Professors C. F. Scott and L. P. Breckenridge. Measuring and Manufacturing to the Millionth of an Inch, by W. H. Weingar.

April 16: Progress in Mechanical Engineering Research, by Prof. Arthur M. Greene, Jr.

May 14: Inspection of works of the Geometric Tool Co. Evolution of the Opening Die Head, by an engineer of the company.

June 2: Business meeting; election of officers.

Waterbury Branch

June 26: Business meeting; election of officers.

DETROIT

January 9: The Design and Application of Electric Propulsion Equipment for Submarines, by David Hall.

February 20: Joint meeting with A.I.E.E. and Detroit Engineering Society. Spectacular Illumination, by W. D'Arcy Ryan.

EASTERN NEW YORK

January 9: Gages, by C. E. Johansson.

February 20: Performance Tests on Airplanes, by A. R. Stevenson, Jr. Supercharger for Airplane Engines, by S. A. Moss.

March 26: Gas Warfare in the A. E. F., by Major Frederick Pope.

April 23: Bearing Metals, by C. H. Bierbaum.

ERIE

April 12: Is There a Cure for Class Differences in America? by Ira N. Hollis.

INDIANAPOLIS

January 23: Joint meeting with Indiana Engineering Society, A.I.E.E., A.A.E., Scientech Club and Indiana Society of Architects. The Department of Public Works, by Professor Newell.

April 22: The Manufacture and Uses of Malleable Castings, by H. A. Schwartz.

LOS ANGELES

February 10: The Aims and Activities of the Society, by Dean D. S. Kimball. The Design, Construction and Use of the 100-Inch Reflector, by F. G. Pease. Solar Work at the Mount Wilson Solar Observatory, by Dr. Charles E. St. John.

May 13: Recent Developments in Physics, Dr. Charles E. St. John.

June 1: Business Meeting.

June 24: Some Interesting Features of the 100-Inch Telescope at the Mount Wilson Observatory, by F. G. Pease.

MID-CONTINENT

May 28: Joint meeting with Oklahoma Section of American Chemical Society. The Preparation of Motor Gasoline from Heavier Hydrocarbons, Professor Fred W. Padgett;[1] Modern Water Purification, by W. R. Holway; Refining Crude Mineral Oils and the Action of Absorptive Clays on Same, by F. C. Thiele; Flow of Fluids through Pipe Lines and the Effect of Pipe Line Fittings, by D. E. Foster; Some Investigations in Briquetting of Oklahoma Bituminous Coal, Professor James C. Davis; Meta-Nitrophenyl Ether, by Dr. Hilton I. Jones; Mid-Continent Gasoline, by Dr. Chas. K. Francis; Coal *vs.* Oil Cost Performance Chart,

[1] See MECHANICAL ENGINEERING, July, 1920.

by L. C. Lichty; Research in the Mid-Continent Field, by Dean P. F. Walker;[1] The Treatment of Cow's Milk for Babies' Food, by Dr. Edwin De Barr; The Engineer and His Citizenship Responsibilities, by E. S. Carman; and Industrial Possibilities of Tulsa, by Clarence B. Douglas.

MILWAUKEE

March 17: Joint meeting with Milwaukee Engineering Society. Falk Gears as Applied to Marine and General Uses, by Percy Day.

April 20: Joint meeting with the Milwaukee Section of A.I.E.E. Address by Calvert Townley.

April 21: Joint meeting with Milwaukee Engineering Society. Engineering Opportunities, President Fred J. Miller.

May 19: Joint meeting with Milwaukee Engineering Society. The Coming Science of Acoustical Engineering, by Prof. Vladimir Karapetoff.

MINNESOTA

January 5: New Problems in Engineering, by Dr. Ira N. Hollis.

February 20: Coals and Coal Analysis, by W. D. Langtry.

May 17: Annual meeting and dinner. Election of officers.

NEW ORLEANS

January 12: Annual meeting and smoker.

February 9: Discussion of proposed National Department of Public Works.

March 17: Business meeting. Sub-Committee on Membership appointed.

April 17: Business meeting. General discussion of Section affairs.

June 4: The Recent Installation of a Large Turbo-Alternator at Niagara, by W. M. White.

NEW YORK

January 13: New Fuel Uses. Colloidal Fuel, by Lindon Bates; Fuel Oil, by E. H. Peabody; Pulverized Coal, by C. C. Trump.

[1] Published in MECHANICAL ENGINEERING, September 1920.

February 10: Alloyed Aluminum as an Engineering Material, by G. M. Rollason.[1]

February 19: Joint meeting with A.S.C.E., A.I.M.E. and A.I.E.E. A National Department of Public Works, by M. O. Leighton.

March 9: Symposium on the Application of Heat in Industrial Processes.

April 13: Symposium on Welding. Elements of Arc Welding, by O. A. Kenyon; Gas Welding, by A. S. Kinsey; Thermit Welding, by J. H. Deppeler.

ONTARIO

February 2: Industrial Relations, by R. W. Gifford.

April 8: Milling Machine Practice and Design, by B. P. Graves.

May 14: Inspection of works of Canadian Westinghouse Co., International Harvester Co. and the new generating station of the Dominion Power and Transmission Co.

OREGON

January 29: Talk by Dean D. S. Kimball.

PHILADELPHIA

January 27: Anti-Aircraft Artillery, Maj. R. L. Goetzenberger.

February 24: Navy Night. War Time Naval Engineering, by Commander J. S. Evans.

March 23: Safeguards in Industrial Plants, by L. A. de Blois.

April 27: Modern Practice in the Manufacture of Steel, by Col. W. P. Barba.

May 25: Export Trade, by Samuel Vauclain. Other speakers, Messrs. Carter, Aertsen, Kenney, Brinley and Liversidge.

June 4: Business meeting.

PITTSBURGH

January 16: Organization meeting.

PROVIDENCE

May 11: Providence Section formed. A Committee was appointed to plan an active session for 1920–21. Meeting of the

[1] Abstract published in MECHANICAL ENGINEERING, September 1920.

Providence Engineering Society followed. Apprenticeship Systems, by Mr. Goss.

PROVIDENCE ENGINEERING SOCIETY

April 27: Foundations in the Vicinity of Providence, by Wm. H. G. Temple.

ROCHESTER

February 4: The Engineer's Participation in Engineering Phases of Public Affairs, by Secretary Calvin W. Rice.

April 14: Is There a Cure for Class Differences in America? by Dr. Ira N. Hollis.

ST. LOUIS

January 2: New Problems in Engineering, by Dr. Ira N. Hollis.

January 28: Construction of the Old Hickory Smokeless Powder Plant at Nashville, Tenn., by A. J. Brandt.

February 20: New By-Product Coke Plant at Granite City, by P. E. Irvine.

April 7: Joint meeting with Associated Engineering Societies of St. Louis. Different Processes in the Manufacture of a Heine Boiler, by F. O. Pahlmeyer.

April 30: Development in Chain Grate Stoker, by H. P. Gauss.

SAN FRANCISCO

January 8: The Industrial Uses of Gas, by B. W. H. Henderson and H. M. Crawford. Use of Oil Gas in Manufacture and Heating of Rivets, by M. Balliet.

February 5: Talk by Dean D. S. Kimball.

February 27: Business meeting.

March 11: The American Plan of Industrial Relations, by Minor Chipman.

March 25: Development of the Diesel Engine, by Adolph A. Tacchella.

April 1: Meeting of Executive Committee. Commercial Development of the Airplane, by Dr. Durand.

April 8: Turbine Reduction Gear for Ship Propulsion, by W. J. Davis.

April 10: Visit to the Municipal Installment of the Diesel Plant at Palo Alto.

April 29: Inspection of the Moore Shipbuilding Co. and the Skandia Pacific Co.

SYRACUSE

June 1: Organization meeting and election of officers.

TOLEDO

April 9: Organization meeting; election of officers. America's Future in the World Wide Industry, by E. S. Carman.

June 3: The Relation of the Engineer to the Public, by President Fred J. Miller.

UTICA

March 18: Organization meeting.

VIRGINIA

January 26: Informal address by Earl F. Scott; election of officers.

WASHINGTON

January 29: Experience with the Ricardo Engine for Tanks in France, by Major Ralph Sasse; Experience with the Diesel Engine for Submarines, by Lieut. Comdr. M. C. Bowman; Artificial High Altitude for the Study of Aircraft, by Dr. A. C. Dickinson.

March 31: Some Commercial Heat Treatments for Alloy Steels, by A. H. Miller; Some Applications of Alloy Steels in the Automotive Industry, by H. J. French; The Application of Alloy Steel to Engineering Work, by P. E. McKenney.[1]

May 17: Gas Welding, S. W. Miller; Electric-Arc Welding of Steel, H. S. Rawdon; Metallic-Electrode Arc-Welding Process, O. H. Eschholz. The papers by Messrs. Rawdon and Eschholz are published in abstract in Mechanical Engineering, October 1920. Followed by a dinner, when a short talk was given on Should Junior Members Vote?

WASHINGTON STATE

January 27: Address by Dean D. S. Kimball to engineering students and to the combined engineering societies of Seattle.

[1] The papers by Messrs. Miller and French were published in MECHANICAL ENGINEERING, September 1920.

March 1: The Relation of the Engineer in Successful Industrial Management, by Stephen I. Miller.

April 19: An Investigation of Compressed Spruce Pulleys, by Professor G. S. Wilson.[1]

June 9: Meeting and banquet. Speakers, R. M. Dyer, C. W. Tubby and E. O. Eastwood.

WORCESTER

January 16: President's night. Speakers, President Fred J. Miller, Dean M. E. Cooley, Charles T. Main, Dr. Ira N. Hollis.

February 10: Industrial Relations, by Robert B. Wolf.

March: A Trip through the Bureau of Standards, Washington, D. C., by Dr. S. W. Stratton.

THE SPRING MEETING

St. Louis, Mo., May 24 to 27

The 1920 Spring Meeting was the first convention of the Society to be held in St. Louis in twenty-four years and so open-hearted was the hospitality, so well carried out were the excursions and entertainments and so thoughtfully was every detail worked out for the comfort of the guests that every member in attendance enthusiastically demanded an early repetition of a St. Louis Spring Meeting.

The total registration was 475 members and 330 guests, most of whom arrived for the first session and remained to the end. All of the professional sessions were held at the Hotel Statler, the headquarters of the meeting.

On Monday evening a reception for visiting members was held at the Hotel Statler. Charles H. Howard, president of the Commonwealth Steel Company of St. Louis, extended a warm welcome. President Miller responded with the thanks of the Society for the hospitality shown. Mr. E. R. Jackson displayed several reels of official moving pictures recording the First Transcontinental Motor Convoy. Mr. L. C. Nordmeyer, Chairman of the St. Louis Section, presided at this meeting, which was followed by dancing.

The Spring Meeting banquet, held Tuesday evening, May 25, in a large dining room of the Missouri Athletic Association, was attended by over 700 members and guests. After the dinner, Lt.-

[1] Abstract published in Mechanical Engineering, August, 1920.

Col. H. W. Miller, of the Ordnance Department, presented a paper on the German Defenses of the Coast on Belgium.

On Thursday afternoon the excursion steamer *St. Paul* was chartered for a boat ride on the Mississippi.

The ladies attending the convention were afforded very interesting trips through the residential section, through Forest Park and visits to the Art Museum, Shaw's Garden, Washington University and to Brentmoor, Westmoreland and Portland Places.

On Tuesday afternoon the plant of the Commonwealth Steel Co. at Granite City, Ill., was visited by a large number of members interested in the making of steel castings. The 122,000-kw. plant of the Union Electric Light and Power Company at Ashley Street was also visited on Tuesday afternoon, followed by a call at the docks of the St. Louis Boat and Engineering Company, where steel barges were under construction for the United States Government.

Wednesday morning the trips included the plants of the General Motors Corporation, Wagner Electric Company, Evertight Piston Ring Company and Heine Safety Boiler Company. Lunch was served at the Bevo plant of the Anheuser-Busch Association, following which the party moved on to the works of the Busch-Sulzer Brothers Diesel Engine Company, the Mississippi Valley Iron Company and the cokeplant of the Laclede Gas Light Company.

Eighty members of the Society availed themselves of the opportunity to inspect the hydroelectric developments of the Mississippi River Power Company at Keokuk.

Following the meeting in St. Louis about fifty of the attending members left for Tulsa, where, on Friday and Saturday a joint meeting of the Mid-Continent section of the Society was held with the Oklahoma section of the American Chemical Society.[1] A full account of this meeting, as well as more extended accounts of the business and professional sessions of the Spring Meeting will be found in the July, 1920, issue of Mechanical Engineering.

The convention was in general charge of the Committee on Meetings and Program, consisting of Dexter S. Kimball, Chairman, A. L. DeLeeuw, W. G. Starkweather, R. V. Wright and J. W. Roe.

The entire arrangement for the entertainments, excursions and publicity for the St. Louis meeting was under the jurisdiction of the General Committee, the personnel of which was as follows: M. L. Holman, Hon. Chairman; Louis C. Nordmeyer, Chairman; Lewis Gustafson, Vice-Chairman; Fred E. Bausch, Secretary;

[1] See p. 8 for program of meeting.

William P. Samuel, Treasurer; Ernest L. Ohle, H. Wade Hibbard, Chester B. Lord, Edward W. Schadek, Victor J. Azbe, and Edward R. Fish. The chairmen of the various sub-committees were: Reception, R. H. Tait; Women's, Mrs. Ernest L. Ohle; Hotel, W. H. Reeves; Publicity, V. J. Azbe; Finance, P. DeC. Ball; Program, E. L. Ohle; Entertainment, Fred. E. Bausch; Information, E. R. Fish; Transportation, W. E. Bryan.

PROGRAM

Monday Morning, May 24

Registration of members and guests at headquarters. Meeting of Council. Organization meeting of Machine Shop Section.

Monday Afternoon

BUSINESS MEETING

Report of tellers of amendments to constitution; reports of committees.

APPRAISAL AND VALUATION SESSION

(Discussion upon the following papers)

Appraisal and Valuation Methods, David H. Ray.[1]

Rational Valuation — A Comparative Study, James Rowland Bibbins.[2]

Data on the Cost of Organizing and Financing a Public-Utility Project, by the late F. B. H. Paine[2] (contributed by Dean M. E. Cooley).

The Construction Period, H. C. Anderson.[2]

Price Levels in Relation to Value, Cecil F. Elmes.[2]

Monday Evening

Address of welcome

Illustrated lecture: The First Transcontinental Motor Convoy, E. R. Jackson,[3] former 1st Lieut., Ordnance Dept., U.S.A.

Reception and dance.

Tuesday Morning, May 25

PROFESSIONAL SESSION
LOCAL SESSION

(Papers contributed by St. Louis engineers)

The Housing Problem in St. Louis, Nelson Cunliff.

Industrial Housing — A Financial Problem, Leslie H. Allen.[4]

[1] Published in Mechanical Engineering, December 1919.
[2] Published in Mechanical Engineering, October 1920.
[3] Published in Mechanical Engineering, March 1920.
[4] Published in Mechanical Engineering, June 1920.

Design of an Ore Fleet for the Upper Mississippi River, Wm. S. Mitchell.[1]

Mississippi Valley Rivers' Transportation Activities, E. W. Schadek.

Burning Eastern Coals Successfully on a Conveyer Feed Type of Stoker, Lloyd R. Stowe.[2]

Tight-Fitting Threads for Bolts and Nuts, Chester B. Lord.

ELEVATOR CODE SESSION

Tuesday Afternoon

Public hearing by Boiler Code Committee on rules for air tanks and pressure vessels.

Excursions.

Tuesday Evening

Banquet.

Illustrated address: The German Defenses on the Coast of Belgium, Lieut-Col. H. W. Miller,[3] Ordnance Dept., U.S.A.

Music and dancing.

Wednesday Morning, May 26

SIMULTANEOUS SESSIONS

AERONAUTIC SESSION

Physical Basis of Air-Propeller Design, F. W. Caldwell and E. N. Fales.

Analytical Theory of Airplanes in Rectilinear Flight and Calculation of Maximum Cruising Radius, A. Rateau, Hon.Mem.Am.Soc.M.E.

Aeronautic Instruments — General Principles of Construction, Testing and Use, Mayo D. Hersey.

Flow of Air Through Small Brass Tubes, T. S. Taylor.

CASTINGS SESSION

(Under auspices of Sub-Committee on Foundry Practice)

Malleable Castings, Enrique Touceda.

Die Castings, Charles Pack.

Aluminum Castings, Zay Jeffries.

Steel Castings, John H. Hall.

Gray-Iron Castings, Richard Moldenke.

Brass and Bronze Castings, C. H. Bierbaum.

Wednesday Afternoon

Excursions.

Wednesday Evening

Entertainment at Municipal Open Air Theatre, Forest Park.

[1] Published in Mechanical Engineering, July 1920.

[2] Published in Mechanical Engineering, May 1920.

[3] Published in Mechanical Engineering, June, 1920.

Thursday Morning, May 27

SIMULTANEOUS SESSIONS

SCIENTIFIC SESSION

(Joint session with American Society of Refrigerating Engineers and American Society of Heating and Ventilating Engineers.)

An Improved Form of Weir for Gaging in Open Channels, Clemens Herschel.

Simplification of Venturi-Meter Calculations, Glenn B. Warren.

Dissipation of Heat by Various Surfaces, T. S. Taylor.

The Thermal Conductivity of Heat Insulators, M. S. Van Dusen. (Contributed by A.S.R.E.)

Ship Ventilation, F. R. Still.

(Contributed by A.S.H. and V.E.)

POWER AND COMBUSTION SESSION

Pulverized Coal in Metallurgical Furnaces at High Altitudes, Otis L. McIntyre.

Efficiency of Natural Gas used in Domestic Service, Robert F. Earhart.

The Separation of Dissolved Gases from Water, J. R. McDermet.

Locomotive Feedwater Heating, Thos. C. McBride.

No. 1739

PHYSICAL BASIS OF AIR-PROPELLER DESIGN

BY F. W. CALDWELL [1] AND E. N. FALES,[1] DAYTON, OHIO
Non-Members

It is the purpose of this paper to summarize the conceptions of flight phenomena as developed by the use of the high-speed wind tunnel at McCook Field, Dayton, Ohio. The paper contains a description of the wind tunnel; a discussion of the phenomena discovered by its use; an analysis of propeller theory and application thereof to fan design; and a review of the methods employed for visualizing air flow.

The wind tunnel at McCook Field is notable, first, for the high velocity of air used (500 m.p.h.) and second, for means adopted to visualize the air flow. This latter is done by maintaining the humidity of the air in the tunnel at a certain predetermined level and by providing opportunities for its condensation where it shows the character of the air flow.

The study of air flow by these means gave new data on vortex formation created by the presence of aerofoils and made it possible to form a clearer conception of the conditions surrounding the operation of propellers.

The value of such investigations as can be made with the McCook Field equipment lies in the fact that hitherto the theory underlying flight phenomena has been purely rational and not directly applicable to engineering design because of the absence of empirical measurement of flight vortices. Because of this, it was impossible to predetermine the performance of aerofoils of new shapes, speeds, and sizes without first building a model and determining the particular coefficients applicable to the new design experimentally. Data obtainable in a wind tunnel having provision for visualizing the air flow are, however, the first step toward supplying this deficiency.

The authors show the application of some of the data obtained to the design of blowers and fans, and in an appendix the application of model wing tests to propeller design is discussed.

MEMBERS of The American Society of Mechanical Engineers as well as most of the older engineering professions have shown a keen interest in aeronautical problems and in the progress of the scientific side of aviation. This interest is no doubt partly due to the romance of seeing realized mankind's age-long desire to imitate

[1] Air Service, Engineering Department, McCook Field.

Presented at the Spring Meeting, St. Louis, Mo., May 1920, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

the birds in their flight; and partly to the patriotic hope that America will become predominant in a field which owes its existence to two American inventors.

2 This interest is very valuable to aeronautical engineers, as we feel that association with the older professions ought to enable us to keep at least one foot on the ground. At the same time it is felt that the information gained in aeronautical research may prove helpful in some ways to the mechanical engineers.

3 It is the purpose of this paper to summarize our conceptions of flight phenomena as developed by the use of the high-speed wind tunnel at McCook Field. The understanding of the fundamental aerodynamical phenomena upon which flight depends has been rather vague throughout the past, simply because of the invisibility of the air. Our knowledge of these phenonema has been based on deductions made from observing the forces produced. (The variation of the air resistance has been well understood for various types of wings, and also for solid bodies of geometrical shape.) To evaluate these forces, however, the results only have been analyzed — not the phenomena themselves. Measurement of the dynamic and static pressures in a given air flow has contributed somewhat to our knowledge of what goes on in the air. The use of threads, solid particles, such as smoke, etc., has also been universal for determining the direction of given portions of an air flow. Such methods, however, have barely scratched the surface of the problem and have been of small value to the aerodynamical analysis of the mathematicians.

4 A means of actually visualizing the air is therefore of greatest interest; and while its immediate application has been to the specific case of studying air flow past aerofoils, many other uses suggest themselves wherever air in motion constitutes an engineering or scientific factor. Thus, for instance, the study of meteorology involves a knowledge of the circulation of air currents on a large scale over the earth's surface; this knowledge, too, is imperfect because of the invisibility of the medium concerned, and can be greatly improved by application of means for visualization. Moreover, the flow of air in ventilation and other engineering processes is a question which may frequently require analysis by a method of visualization.

5 It is thought that the general engineering profession will quickly deduce from the experiments recorded in this paper principles which can be of service in their more commercial fields. There-

fore it has been our aim to summarize the main features of these experiments without going too deeply into the technical details necessarily of interest only to the aeronautical engineer. The experiments have resulted in discoveries which may be briefly summarized as follows:

6 The flow of air, whether in a duct or past an aerodynamic body, is subject to parasite energy losses connected with accidental

Fig. 1 Photo of Tip Vortex, Camera Axis being Coincident with Vortex Axis

whirls and eddies, and also to internal motion on account either of the shape of the containing duct or the aeronautical body. Analysis of these motions in the air can be facilitated by visualizing them. The writers' method for producing visualization involves condensation of the moisture contained in the air. This moisture, when turned into visible vapor by suitable means, takes up shapes dependent upon the motion and pressure gradient of the air and justifies us in considering that the air itself is thus visible. (See Fig. 1.)

7 When the air is made to flow past an aerodynamic body, special shapes of flow result and can be visualized by our method. It has been shown in our experiments that when the speed changes, the characteristics of the internal motion in the air alter. This alteration has been a source of much uncertainty to aeronautical engineers in the past; but the combination of visual study of these changes of air flow, together with quantitative analysis of the forces involved, has made it possible to throw much light on the underlying structure of the air as it applies to dynamic flight and propulsion.

8 In our experiments, then, we introduced small model wings

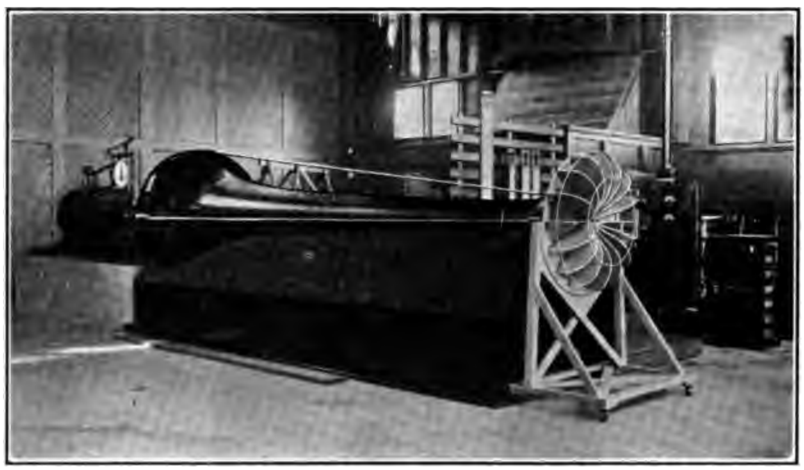


Fig. 2 McCook Field Wind Tunnel, General External View

(aerofoils) into the air current of the wind tunnel and studied their performance, intending to interpret the results for the purpose of propeller design.

WIND-TUNNEL APPARATUS USED FOR OBSERVING AEROFOIL COEFFICIENTS AND AIR FLOW

9 The McCook Field wind tunnel (Fig. 2) was designed by the writers for advanced study of aerofoil coefficients used in propellers. It is a departure from the accepted type of wind tunnel in two ways: first, it has a speed higher than elsewhere attained for similar purposes — 500 m.p.h. at 200 hp.; second, it makes possible the visualization of aerodynamic phenomena.

10 It seems wise to summarize briefly the general function of a wind tunnel, which is the laboratory apparatus of the aeronautical engineer. An airplane, when once in the air, is very difficult to analyze; neither the pilot nor an observer on the ground is able to gage accurately the speed, angular attitude, nor the air forces acting. So far, no thoroughly accurate instruments for accomplishing full-flight analysis have been perfected. Projects have been proposed for attaching a full-sized airplane to a moving electric locomotive properly equipped to record accurately all the forces produced; but such a plan has drawbacks. It is found that the best, most accurate and cheapest way to analyze airplane characteristics is by means of the wind tunnel. This is a confined artificial blast of air which is made to blow against a model wing or airplane. The model is supported in the air blast by a thin rod reaching into the wind tunnel from outside, where it connects to a delicate scale. Thus all forces created on the model are transmitted to the scale where they can be measured.

11 It is found that the forces due to the air current blowing against fixed objects are the same as though the object were moved through still air, provided the air current is smooth and without eddies. Elimination of eddies is therefore an important requirement in a wind tunnel.

12 The forces found in a wind tunnel apply to full-sized machines, and wherever aeronautical design is carried on the wind tunnel is essential. Besides the McCook Field wind tunnel there are in America about a dozen others. In Europe there are many more, the British National Physical Laboratory employing seven. The largest in existence is now being built at Paris, 13 ft. in diameter and of 1000 hp. capacity. The typical wind tunnel has a stream of air 15 to 50 sq. ft. in cross-sectional area, flowing at a velocity of from 30 to 90 m.p.h.

THE QUESTION OF WIND-TUNNEL SPEED

13 The speed and size of a wind tunnel have a recognized importance in the interpretation of the results obtained. The usual coefficients of aeronautical engineering are determined by tests on a model aerofoil of from 3 to 6 in. chord at velocities of 45 to 130 ft. per sec. The data obtained must be applied in the full scale to areas a hundredfold greater in the case of wings, and to velocities ten times greater in the case of propellers. There may

be a discrepancy between model and full-scale coefficients, and this is spoken of as "scaling effect."

14 *Scaling Effect.* While scaling rules and empirical factors worked out in practice have enabled us to produce very fair results, there is decided room for improvement. Indeed, so meager is the available information on the scaling effect, as treated under the "law of dynamic similarity," that a growing tendency has appeared among aeronautical engineers to regard the classical coefficients K_x (coefficient of drag) and K_y (coefficient of lift) as inadequate.

15 It has been almost universally the practice to write

$$L = \rho/g K_y A V^2$$
$$D = \rho/g K_x A V^2$$

where L is the lift (vertical lifting force), D the drag (resistance),

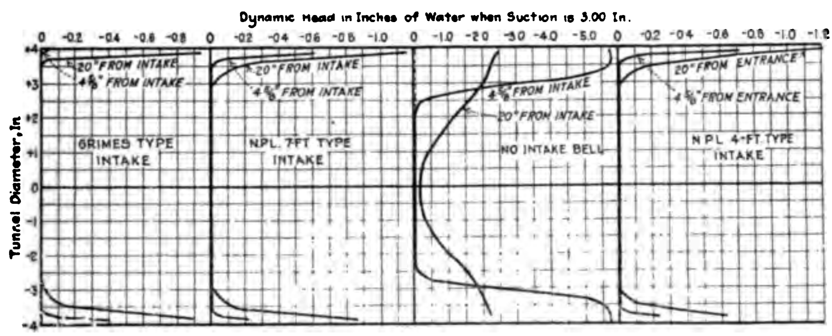


FIG. 3 CHART OF VELOCITY TRAVERSES FOR VARIOUS INTAKES

A the area of the supporting surface, V the velocity of advance, ρ the density of the air in weight units, g the acceleration due to gravity, and hence (ρ/g) the density of the air.

16 It is well known as the result of experience that the values of K_x and K_y vary somewhat with velocity and also with the size of the surface under consideration. If l represents one of the linear dimensions of the surface and γ the coefficient of kinematic viscosity, it is assumed, according to the law of dynamic similarity, that values of K_x and K_y are functions of Vl/γ. Since γ is usually considered constant, it is customary to compare the product Vl for experiments in a given medium.

17 The wind tunnel designed by the writers permits the attainment of a Vl product of 60 (V in ft. per sec., l in ft.) While this figure is twenty times less than the corresponding full-sized

wing values, it is only four to eight times less than full-sized propeller values.

18 As a matter of interest, tip speeds of some of the propellers in actual use are given below:

Airplane	Engine	M.p.h.
USD–9	Liberty–12	650
VE–7	Hispano-Suiza, 150 hp	545
Thomas-Morse	LeRhone, 80 hp	380
Verville Chasse	Hispano-Suiza, 300 hp	600
Roché XB–1–A	Hispano-Suiza, 300 hp	625
Curtiss JN–4	Curtis OX–5	420
DH–9	Rolls-Royce, 375 hp	430

19 It would be ideal if a wind tunnel could be built large enough for testing propellers and airplanes of full size and speed.

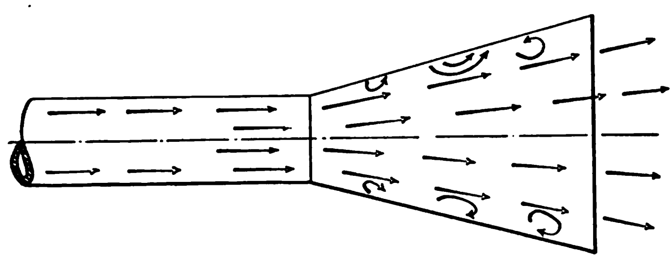


Fig. 4 Diagram Showing Eddies in Cone of Too Large Angle

Such apparatus would, of course, be enormous and of prohibitive cost. We must therefore be content with wind tunnels whose Vl product is smaller than the full-flight value. In designing a tunnel we may have either large size and low speed, or vice versa. In general a large wind tunnel requires a large, expensive hall, while a high-speed tunnel requires high horsepower.

20 Thus in the McCook wind tunnel we have attained a velocity equal to full-flight values, and the size of model is correspondingly small. The results obtained, however, demonstrate greater significance than were the same Vl product obtained with a larger model and smaller velocity. By means of the high velocity, discoveries of outstanding importance have been made, namely, the means of visualizing flight vortices, and the identification of changes in these vortices with simultaneous changes in the aerofoil coefficients.

DESCRIPTION OF THE McCOOK FIELD WIND TUNNEL

21 The tunnel has the form of a venturi tube 18⅜ ft. long, built of turned laminated wood. Its proportions and novel features have been the subject of exhaustive investigation and experiments on scale models.

22 *Intake Bell.* Thus the intake bell has a radius of curvature of 1⅝ times the throat diameter, such as best to take care of the

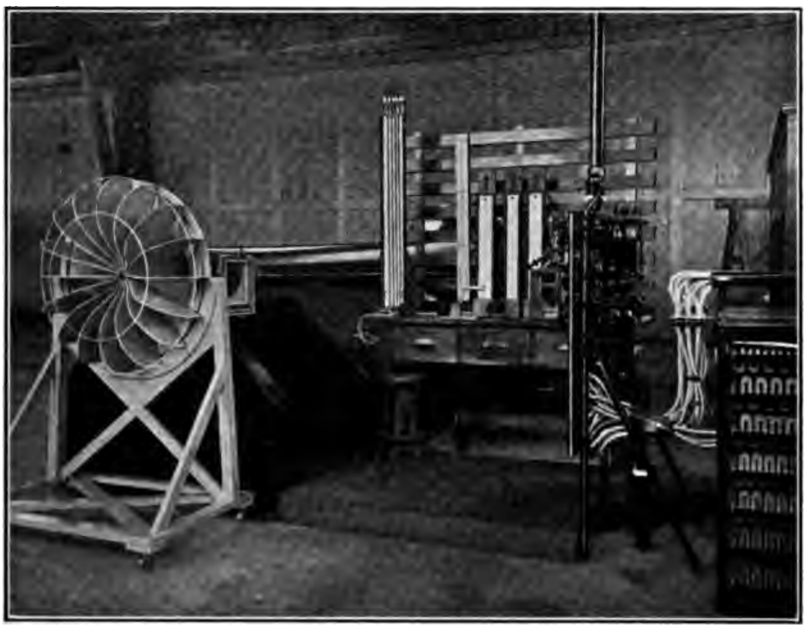


FIG. 5_McCOOK FIELD WIND TUNNEL, SHOWING OPERATING PANELS

vena contracta and insure best velocity distribution and parallelism in the air flow (see chart, Fig. 3.)

23 *Throat.* The throat is 14 in. in diameter and unusually short — 18 in. — with resulting economy of power. The short throat and the location of the model close to the intake are justified by the study of the chart in Fig. 3, and are made possible by the elimination of the usual "honeycomb" and by proper intake-bell design.

24 *Expanding Cone.* The cone leading from throat to fan has an angle of 7 deg., the maximum value established by Eiffel. We have found that in cones of larger divergence the air flow does

not fill the cone, a space of whirling and eddying air being left between the blast and the containing walls (see Fig. 4), with resultant loss of energy. The principle involved is that flowing air surrounded by solid walls loses less energy by surface friction than if surrounded by inert air.

25 *Straighteners.* To do away with velocity fluctuations a novel straightener is used — the conventional "honeycomb" being eliminated. The straightener is placed 4 ft. down the cone and has four radial blades 4 ft. long so located as to prevent inflow whirl by obstructing the formation of the pressure apex. An auxiliary straightener (Fig. 5) with flat radial vanes is located outside the

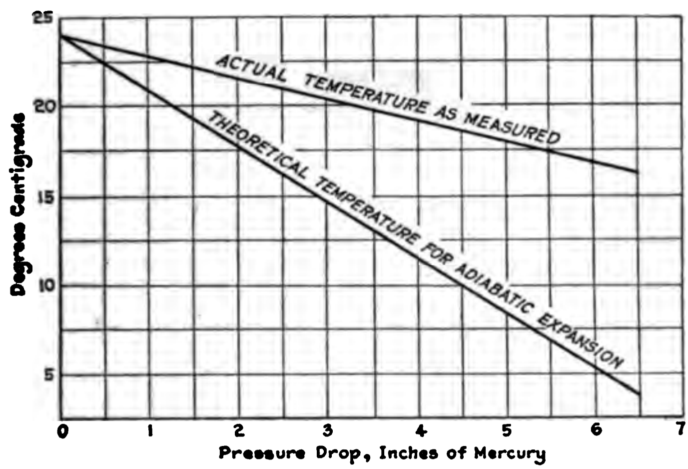


FIG. 6 CHART OF THEORETICAL AND MEASURED TEMPERATURES

intake; and together the two reduce velocity fluctuations from 15 per cent to 3 per cent, without appreciable power loss.

VELOCITY MEASUREMENT

26 Velocity is measured in terms of dynamic head minus static head. For measuring dynamic head there is an "impact" tube at the throat somewhat off the center of the tunnel axis. For measuring static head a perforated plate is used, set in flush with the walls. A differential manometer records the velocity head, the whole apparatus being analogous to the conventional pitot tube and Krell manometer.

27 Temperature of the air passing through the throat is calculated on the assumption that expansion is adiabatic from atmos-

pheric pressure to pressure corresponding to condensation, and is polytropic below the latter pressure. Correct knowledge of throat temperature is, of course, essential, and it is necessary to develop a special method of thermometry for reading it. Present methods are inapplicable to its direct measurement, for a thermometer introduced into the air stream occasions more or less adiabatic compression of the air striking it, with consequent rise of temperature at the point of impact. (See chart, Fig. 6.) The most advantageous position for the thermometer is with the bulb

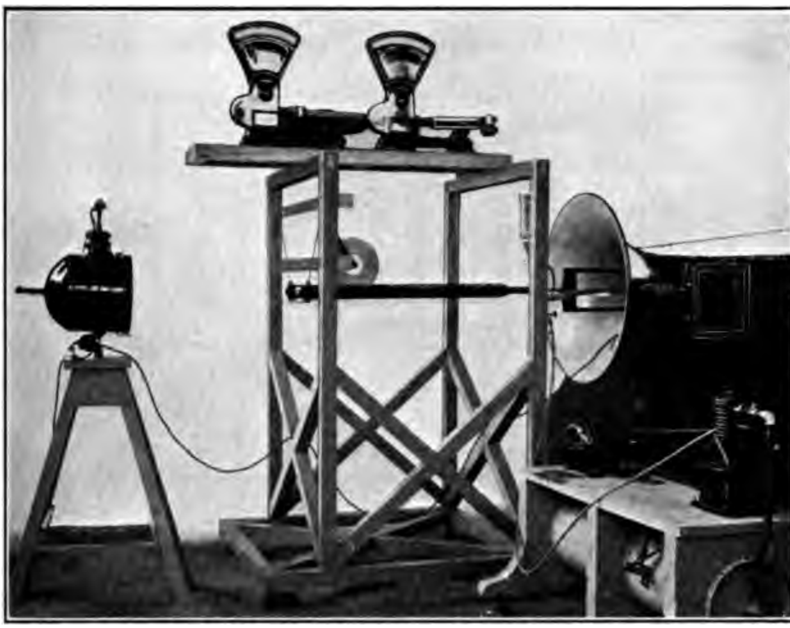


FIG. 7 McCOOK FIELD WIND TUNNEL, BALANCES AND INTAKE

down stream, where it is subject chiefly to skin friction rather than impact. Further investigations are being made of the matter.

BALANCES

28 The balances are of two types. The first one measures lift and drag on two separate instantaneous-reading Toledo scales and is mounted upon a portable carriage (see Fig. 7). The spindle for the model projects horizontally and axially from this carriage into the mouth of the wind tunnel, carrying the model at its free end. The spindle terminates in a thin flat bar, the latter clamping a graduated disk which is rigid with the model at the center of the

span. Three advantages of this type of balance are: (1) That instantaneous readings make it possible to synchronize balance and velocity observations and to practically eliminate the effect of velocity fluctuations. (2) The air forces can be qualitatively studied, as, for instance, in the case where a given setup has two values of K_y or K_x. The balance can then be seen to change from one reading to another. (3) The method of support affords a highly accurate means of skin-friction observation.

29 The conventional supporting spindle, as adopted at the National Physical Laboratory, cannot be utilized under the con-

Fig. 8 Tip Vortex, High-lift Régime

ditions of these tests. In order definitely to delimit the effect of the supporting member, further developments are proposed where this effect will virtually be eliminated from the tests. The effect of the center support on the lift coefficient is not considered serious. This conclusion is based on experiments run at other laboratories where the effect of the support has been determined; also on a comparison of the present series of experiments with tests made elsewhere on a larger model supported at the end, for the same Vl values. The effect of the center support on drag, however, is known to be large; the air-flow disturbance is visualized in Fig. 8, where it appears as a white tuft above the support.

30 *Power Plant.* The power plant consists of a Sprague dynamometer, capable of delivering 200 hp. for one-half hour at 250 volts and 1770 r.p.m without overheating. The 5-ft. fan (Fig. 9) is made with a solid-center disk 40 in. in diameter and has 24 blades 10 in. long. At the upstream side of the 40-in. disk a bell of equal diameter is fixed in the tunnel so that the air is led up to the annular discharge opening with a minimum of eddies. The operating efficiency of the whole tunnel is 75 per cent,-higher than has been usual for the determination of aerofoil coefficients in other wind tunnels. By efficiency is understood the ratio of kinetic energy of air stream

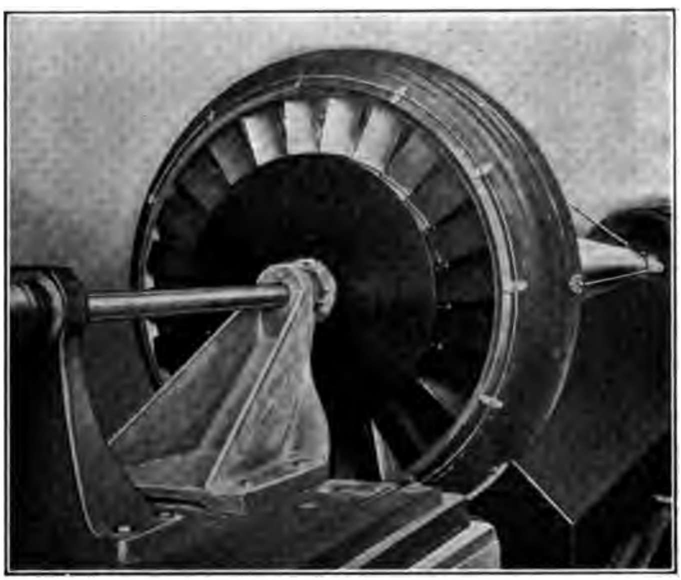


FIG. 9 MCCOOK FIELD WIND TUNNEL, FAN END

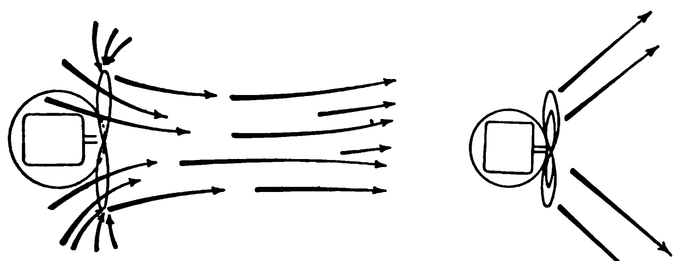
at throat of tunnel, minus the energy absorbed by the fan, all divided by the energy of the air stream at the throat.

31 Careful study of fan and cone design results not only in reduced losses but also in reduced noise. The question of noise has in the past been a serious objection to speeds greater than 70 miles per hour in wind tunnels. The roar of the fan is analogous to that of an airplane propeller, which usually makes more noise than does the unmuffled motor exhaust. For wind-tunnel use the combination of fan and cone adopted has brought about a considerable improvement, as indicated in the following tabulation:

From the operator's position:	Fan is noiseless at	50 m.p.h.
	Starts to roar at	60 m.p.h.
	Conversation easy at	125 m.p.h.
	Conversation slightly forced at	155 m.p.h.
	Possible 12 in. apart at	240 m.p.h.
	Possible 4 in. apart at	300 m.p.h.

32 *Fan.* The fan represents a very interesting union of airplane-propeller principles and blower-design principles, combining the high pressure usually associated with centrifugal blowers with the high efficiency of the airplane propeller. While essentially a propeller fan of aeronautical design, its 16-in. pressure head is such as would ordinarily be expected only from a centrifugal blower.

33 This application of airplane-propeller principles to blower



A B

FIG. 10-A AIR FLOW THROUGH TYPICAL ELECTRIC FAN
FIG. 10-B AIR FLOW THROUGH MODIFIED ELECTRIC FAN

design is significant to the ventilation engineer, who may by such means produce in wood, at comparatively small cost, blowing apparatus of high efficiency, high horsepower, high head, and reduced noise.

APPLICATION OF PROPELLER DESIGN TO COMMERCIAL USE

34 As a further illustration of the benefits to the ventilation engineer of airplane-propeller principles, the following may be cited as of interest:

35 It has been found possible, by impressing special properties, to extend the usefulness of one of our most common and universal devices — the electric cooling fan. In the ordinary electric cooling fan referred to, the air flow is analogous to that of an airplane propeller rotating at a fixed point on the ground, and is shown in the sketch, Fig. 10-A. The blast converges as it leaves the blades,

reaching a maximum velocity a short distance from the fan, and gradually dissipating its energy as the flow lines expand along a small-angle cone. Thus, in such a blast the energy is concentrated in a narrow stream. Numerous experimenters have conceived the idea of placing vanes in front of these fans to diffuse the air and make it spread outward, but no practical success has been secured.

36 The problem is one of many minor ones which have found their solution as a side issue to the war activities of the Air Service. By judicious application of aerodynamic principles, a cooling fan has been developed which delivers a fan-shaped expanding blast, while moving as much air as the older conventional type of fan. See sketch, Fig. 10-B. This fan directs its stream, not along a single axial line, but along a sector embracing 90 or even 180 deg. The advantages are obvious:

First, the energy of the blast is rapidly dissipated over a large area, causing a large, gentle circulation in place of a small, violent one.

Second, the blast spreads over an angle equal to that attained by the conventional oscillating type of fan, and accomplishes the same beneficial results without the unpleasant features associated with intermittent action.

Third, such a fan can produce a truly conical blast of which the fan-shaped expanding blast is only a special case; thus, the advantage of large cross-sectional area is secured from a small high-speed fan; and for purposes of overhead or ceiling use such a device is considerably cheaper than the conventional low-speed, large-diameter ceiling fan.

Fourth, the device involves only a simple, inexpensive addition to the conventional fan, may be readily applied to it, and causes no additional noise.

VISUALIZATION OF FLIGHT VORTICES BY THE WRITERS' METHOD

37 The method of visualizing air flow discovered by the writers together with C. P. Grimes offers a solution of the fundamental problems of aerodynamics. This problem is the quantitative empirical measurement of the phenomena of fluid dynamics as applied to flight and air flow.

38 The accepted theory upon which flight has its physical basis is purely rational. It has not yet been directly applicable to engineering design, because empirical measurement of flight vor-

tices has never been made. Therefore, the aeronautical engineer's use of aerodynamics is largely according to the cut-and-try method. He cannot, on the drafting board, depart from known shapes, speeds, or sizes without first building a model and determining the coefficients applicable to his new design.

39 To illustrate this point it is only necessary to refer to the simplest case, that of an airplane wing. We can measure the coefficient of force on a small model of this wing to an accuracy of 1 per cent, but we do not know definitely how the accuracy is changed by scaling up to full size, or to full speed. We cannot, without tests, predict the change of coefficient to be expected when the wing shape is altered, or the angle of attack, or the position with reference to other surfaces.

40 Aerodynamical theory will serve practical use when supported by empirical data. In the past, flight vortices have never been measured, nor even visualized to a usable extent. Analysis of air flow has been confined to the use of smoke or powder set loose in the air to indicate lines of flow; or of threads used as wind vanes. Or we have been driven to analogies derived from study of fluids of different viscosities and densities, such as water. Or, further, we have sought by measurement of static pressures in the air surrounding a body to deduce the lines of flow. But these methods have given small encouragement to the practical application of the vortex theory to engineering use.

41 The writers' method bids fair to supply the missing link between aerodynamical theory and design. It depends upon the fact that the moisture in the air condenses out as fog when the temperature is reduced to the dewpoint, provided there is a solid or liquid nucleus to start the condensation. In the McCook Field wind tunnel the temperature drop is brought about through expansion of the air during acceleration due to 100 in. of water suction. Relative humidity of the atmosphere can be artificially raised if too low. The necessary nucleus for condensation is provided by the model itself.

42 Flight vortices become readily visible by the writers' method and can be photographed with the aid of searchlights. Several efforts were made to take the pictures with a plate camera but these were not very successful. Finally a good moving picture was taken and some of the films enlarged. While these films showed up very well on the screen the detail was not very clear in enlargement, so that, in addition to the searchlights which were provided

with nitrogen-filled incandescent lamps, two carbon arc lights were set up in order to give a greater amount of blue light. The results obtained from the motion-picture camera with the carbon-arc lighting were fairly satisfactory, and a number of enlargements from the motion-picture films are reproduced in this paper. Fig. 1 is an enlargement of a moving-picture film looking downstream, and shows the left half-span of a small model aerofoil, with tip vortex trailing downstream from the tip and edge vortex-sheet below the

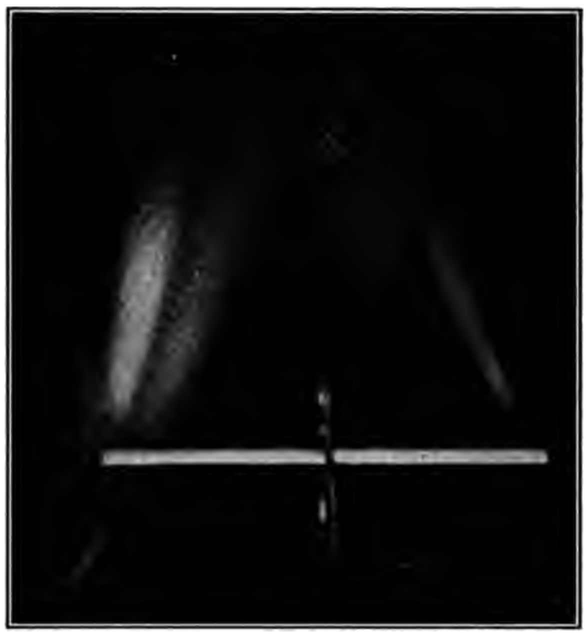


FIG. 11 PLAN VIEW OF FLIGHT VORTICES

model. It is somewhat inferior to visual observation. To the naked eye the tip vortices are in line with the wing tips and are clean-cut, perfect circles. They extend downstream a distance of several dozen chord lengths from the rear corner of each wing tip, enlarging in diameter as the distance increases, and converging slightly in the horizontal plane (see Fig. 11). In the vertical plane the tip-vortex axis takes a decided downward angle, intermediate between the horizontal and the line of the flat sheet of edge vortices. The edge vortex-sheet is made up of vortices with axes parallel to the rear edge of the model; they build up and run off at such frequency as to appear continuous.[1]

43 At slow speeds condensation is not obtainable, but the vortex phenomena can be corroborated by injecting a jet of visible steam into the air current. Such a jet, when of proper saturation, affords a better indication of flow direction than does smoke or thread.

44 Adequate analysis of the flight vortices is now being made with stroboscopic apparatus. The shape, size and direction of the tip vortices can be easily noted and seem fairly susceptible of measurement. The periodic run of the edge vortices is too quick for rec-

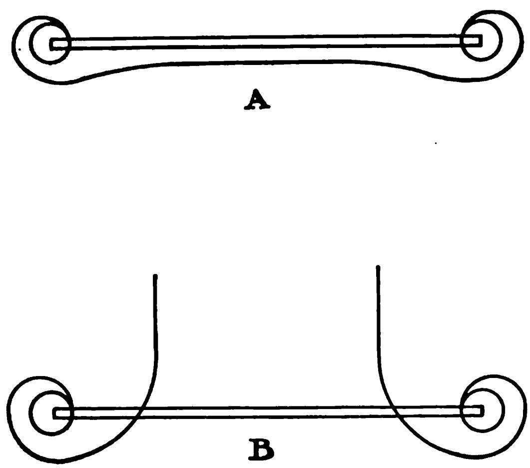


FIG. 12 DIAGRAM OF HIGH- AND LOW-LIFT VORTICES

ognition by the naked eye, or even for identification by the moving-picture camera; it requires stroboscopic analysis.

45 The observed vortices differ for different aerofoil setups and different speeds. For instance, the observed tip vortex at 18 deg. has less than one-half the diameter manifest at 8 deg., while the line of edge vortices is less noticeable at 18 deg.

46 Again, the character of the general vortex phenomenon undergoes remarkable change at the critical speed. In the high-lift

[1] For an excellent mathematical discussion of the shape and arrangement of the tip vortices and the trailing vortices, see report No. 28 of the National Advisory Committee for Aeronautics (U. S.). On page 44 the author, Dr. George de Bothézat, has shown in an interesting way that the axis of the tip vortices is intermediate between the direction of movement and the sheet of trailing edge vortices.

régime the general shape is like a trough whose floor (edge vortex-sheet) slopes downward from the trailing edge, and whose walls (tip vortices) are increasingly high as the distance downstream increases. The cross-section is roughly like a shallow U.

47 At higher speeds, however, in the low-lift régime, the observed phenomenon is suddenly altered. Following out the above homely analogy, we may imagine that the "walls" of the trough remain substantially as before. The "floor," however, splits longitudinally, curls upward, and extends the two limbs, now free, to a

FIG. 13 END VIEW OF HIGH-LIFT PHENOMENON SHOWN IN FIG. 12-A

point well above the level of the tip vortices. This is shown in enlargements of two motion-picture exposures which were intended to record the sequence of the change. These photographs, however, were not altogether satisfactory, and are therefore replaced by the two sketches of Fig. 12.

48 Fig. 12-A is a diagram of the end view of the high-lift phenomenon as it appears distinctly to the naked eye. Fig. 13 is a photograph showing the phenomenon. The right-hand side of the aerofoil in Fig. 8 approximates this condition, the left-hand side

having already gone over to low-lift flow. Fig. 12-B is a diagram of the low-lift phenomenon. Fig. 11 is a three-fourths view of the low-lift flow, and also represents other features mentioned above.

49 An interesting variation of the flight vortices is furnished by replacing the aerofoil by a flat disk normal to the wind. Here the phenomenon can be seen as a "stream-line" fog surface, converging towards a point half a dozen diameters downstream.

50 Fig. 8 illustrates the distance above the aerofoil to which the flight-vortex phenomena may extend, and shows their tendency to merge with other whirls attributable to the wind-tunnel walls. The extent of the phenomenon may be four or more chord lengths above the aerofoil; this further develops a discovery made by the writers in 1911, when it was shown experimentally that the air flow above a wing is disturbed to a distance of at least four chord lengths.[1]

VISUALIZATION OF UNOBSTRUCTED AIR FLOW

51 When the model is removed the vortices and eddies of flow through the unobstructed throat may be observed by looking into the intake or through the transparent shield of the observation section. The condensation is more pronounced behind the impact tube and thermometer bulb than elsewhere, since these are obstacles to the flow and therefore constitute nuclei for condensation. A projecting cotter pin $\frac{1}{16}$ in. high at the wall causes a perfect vortex, which shows up against the white, foggy background as a black circle.

52 The general appearance of the air flow, which may be considered typical of all air flow, is as follows: A cross-section at the throat shows a seething mass of fog specters, denser at the wall than at the center, though occasionally the entire disk fills up with fog to the point of opaqueness. The specters have in the cross-sectional plane a gentle movement like the flame of an alcohol stove, showing the constant readjustment of equilibrium. Vortices and S-shaped whirls continually form and, after moving about, lose themselves in the general confusion. In a diagonal view they take the appearance of long, foggy fibers, stretching down the tunnel like wooden moldings. The axes of whirl, of course, are longitudinal.

[1] See The Center of Pressure Travel on Airplane Surfaces and Birds' Wings, Mass. Inst. of Technology, 1911. Reference may also be made in this connection to the work of J. R. Pannell, dealing with experimental evidence as to the extent of circulation about an aerofoil.

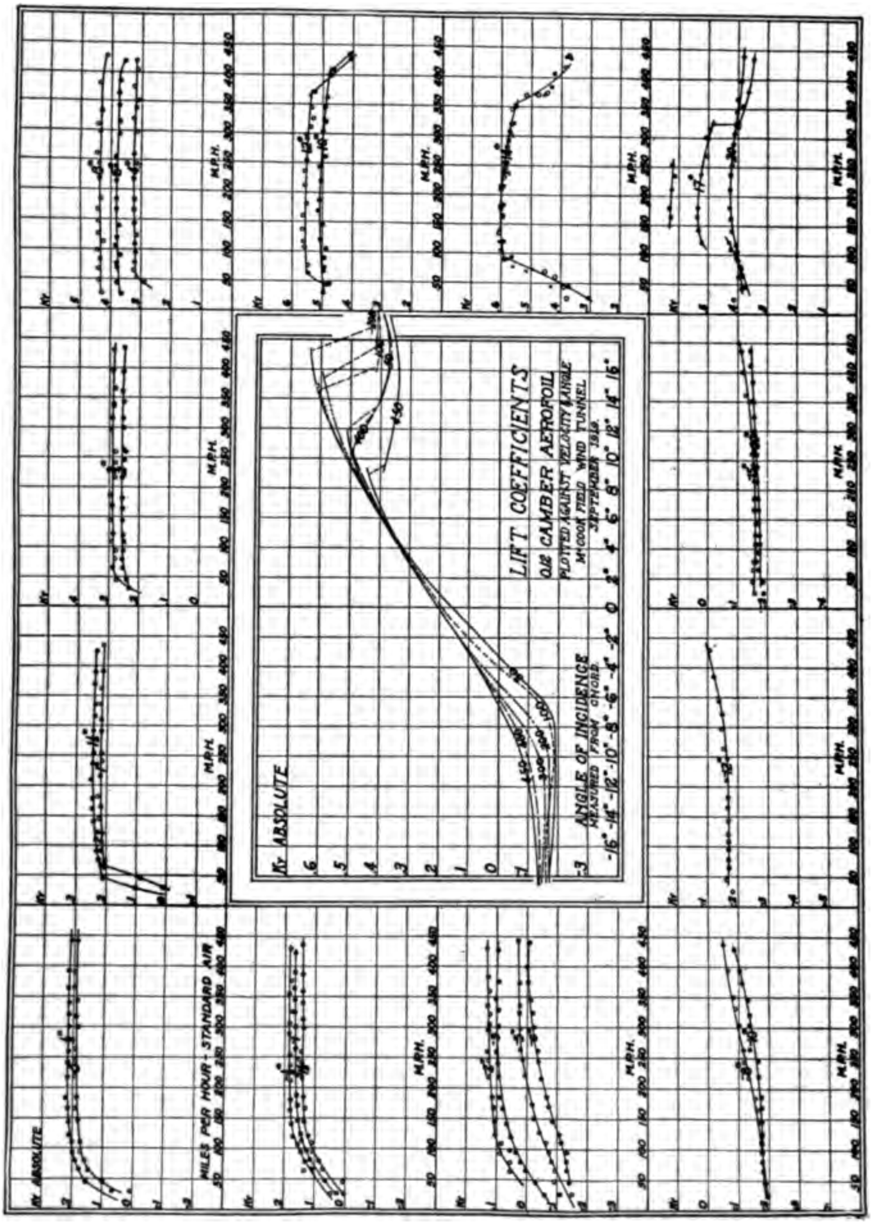


FIG. 14 CHART OF LIFT COEFFICIENTS, 0.12 CAMBER AEROFOIL

Under proper humidity and lighting conditions the whole becomes a beautiful iridescent sight, violet and purple hues predominating.

53 *Lift Coefficients.* Coefficients of lifting force and resistance have been determined for a series of model aerofoils 6 in. long, 1 in. in chord length, camber varying from 0.08 to 0.20. The cross-sectional shape is that upon which the Engineering Division has standardized for propeller use. A chart of lift coefficients for one of these aerofoils (0.12 camber) is presented, and shows the variation of lift coefficient over the entire range of speeds and angles tested. (See Fig. 14.)

54 The outstanding conclusion to be drawn from the tests is that we have more than one régime of air flow to deal with in aerofoil study, and that these régimes are separated by conditions of discontinuity. The characteristics usually associated in aeronautical engineering with a practical aerofoil do not apply outside the small range of cambers, speeds and angles utilized in flight. Beyond this range the flow about the aerofoil no longer produces the familiar results in terms of lift and drag but becomes analogous to the flow about a body of irregular shape. Efficient lift of an aerofoil is only a single case of several distinct aerodynamic phenomena resulting from air flow past a solid body. When the speed of air flowing past an aerofoil increases, there is first a régime of relatively low-lift effect, then at higher speeds an efficient lift effect such as applies in flight, then at still higher speeds a drop back to a second low-lift effect. As the angle of camber increases, the high-lift régime becomes discontinuous and is succeeded by the low-lift régime; the transition point is spoken of in conventional graphs as the critical, or stalling angle, or the "burble point."

55 All of the sections show, at certain angles, two speeds at which the flow is unstable and discontinuous. At the point of discontinuity occurring at the lower speed, increase of speed shows an increased lift coefficient and a decreased drag coefficient so that the lift-drag ratio (L/D) is enormously increased. At the point of discontinuity occurring at the higher speed, increase of speed shows a decreased lift coefficient and an increased drag coefficient, so that the lift-drag ratio is enormously decreased. Thus these sections have a definite speed range, at each angle, within which the flow is efficient and produces a high lift-drag ratio, and a fairly constant lift coefficient. It may be called the régime of high L/D, and includes the phenomena appertaining to practical flight. This speed range has been definitely measured for the higher angles, but ap-

parently it goes beyond the speed obtainable in the tunnel for the lower angles.

56 It will be noted that the K_y curves are drawn discontinuous to correspond with discontinuity in the type of air flow. In some cases the graphs show a third curve intermediate between the high-régime curve and the low-régime curve. This third intermediate line undoubtedly represents different types of flow on the two parts of the aerofoil. This is possible because of the center support which divides the aerofoil into two parts. At the point of discontinuity corresponding to the second critical speed the lift reading becomes unsteady and the flow phenomena become unstable and jump from one type to another until the new form is established. The flow on the left side of Fig. 8 has passed the transition point and belongs to the low-lift régime. The flow in Fig. 13 belongs to the high-lift régime.

57 The discovery of this second critical speed is one of the novel and significant features of the experiments. Simultaneous observation of the balance and the flight vortices made the discovery possible; affording proof that the two types of flight vortices can be identified with the two values of the lift coefficient, one belonging to a high L/D régime, the other to a low L/D régime.

58 Various experimenters have in the past given evidence of discontinuity of the flow past aerodynamic objects. As applied to aerofoils, Mr. Orville Wright conducted in 1918 a particularly interesting series of experiments, the results of which unfortunately have not been published, wherein new discoveries were brought out regarding the discontinuity of flow about propeller sections. He found that at certain angles thick sections manifested a dual value of the coefficients, the angle at which discontinuity occurred depending on whether the angle was increasing or decreasing at the time when the readings were taken. He interpreted the phenomenon as due to a change of air flow. He found two distinct values of the lift coefficient and of the drift coefficient throughout the range of instability; the high lift value always corresponding to a low drift value and vice versa. It was possible to make the values change back and forth between the two points at will by disturbing the air current.

59 The writers have been able to verify conclusively Mr. Wright's assumption that his results involved a change of air flow; wherever discontinuity has been encountered in our experiments it has always been found that the drag and lift became discontinuous

simultaneously, and that the high value of lift always coincided with the low value of drag.

PRACTICAL SIGNIFICANCE OF THE RESULTS

60 Reference has already been made to the prospect, opened by these experiments, of developing a usable physical theory of flight. The immediate practical results have also been important, especially as regards propeller design. For example, we have demonstrated the existence of a limiting tip speed.

61 When the critical speed at which the change in flow takes place is reached, there is in some cases a violent chattering of the model and support; and the nature of the vibrations readily suggests a connection with the fluttering sometimes observed in propellers. The speed encountered is equal to the tip speed of slow-speed propellers, but is considerably inferior to the tip speed of the very large fast-turning propellers used on the Liberty engine.

62 Many static tests carried out by the writers on propellers have shown an effect which appears to be related to the discoveries made during the series of aerofoil experiments. It is well known that the ratio of propeller thrust to propeller torque must be independent of speed of revolution if both thrust and torque are proportional to the square of the revolutions. It has been found in practice, however, that the ratio of thrust to torque decreases greatly with revolution speed when high tip speeds are reached, and that it sometimes shows a rapid decrease accompanied by violent fluttering. The analogy to the results of the aerofoil tests is obvious and it is felt that considerable study is desirable in order to connect these facts in a rational manner.

63 We have found by practical experience that if we do not go below a value of V/ND (velocity/r.p.m. of engine $\times$ diameter of propeller) of 0.65 we get a very fair propeller efficiency. As we have gradually increased the speed of our planes we have gone on increasing the r.p.m. of the engine and the diameter of our propellers so that the value of V/ND has remained about the same for the great majority of propellers in actual service.

64 We have always assumed that there was no limit to this development aside from the characteristics of the plane and engine. That is, we have made the assumption that we could double our propeller speed just as soon as we were able to double our plane speed and strengthen our engine enough to stand the stresses involved.

65 It now appears, however, as though there is a limit to pro-

peller speed aside from the value of V/ND or, to use more familiar terms, aside from pitch ratio. Unfortunately, even the speed obtainable in the McCook Field wind tunnel is not great enough to measure the limiting velocity for relatively thin sections when set at low angles, consequently we are only able to infer that it exists.

CONCEPTION OF LIMITING STRESS IN FLUID DYNAMICS

66 Mathematical studies of first importance, which are now classical, on the nature of the flow about an aerofoil have been developed by Helmholtz, Kirchhoff, Lord Rayleigh, Lanchester, Prandtl, Kutta, Karman, Greenhill, Lewis, and others. Dr. Georges de Bothézat has put forward some very interesting ideas about the effect of stresses in the fluid on the nature of the air flow, and by his theory it is proper to consider in fluid dynamics the same sort of stresses with which we are familiar in solid statics.

67 It is Dr. de Bothézat's conception that the type of flow which establishes itself is governed by the stresses set up in the air passing the aerofoil. The unit stresses increase as the velocity rises. It is easy to conceive that a given type of flow is possible only so long as the shearing stress developed in the fluid does not exceed a certain magnitude which depends on the value of the viscosity coefficient. When the stress reaches a certain critical value, rupture occurs in the air-flow structure; adjacent layers of air begin to slide past each other. Since there is no longer flow similarity, the aerofoil characteristics must change in the manner described earlier in this paper.

APPENDIX NO. 1

APPLICATION OF MODEL WING TESTS TO PROPELLER DESIGN

68 At first sight it may not be clear that aerofoil tests have advantages over model propeller tests as a preliminary to propeller design. Since, however, each different model propeller has distinctly different characteristics, we must have an almost infinite series of tests in order to apply the results. In the case of the aerofoils, it is possible to apply the results of a single series of tests to all propeller designs, provided we are willing to take the trouble to apply the proper mathematical analysis. It will be of interest to know that the Wright brothers made use of aerofoil tests in the design of propellers for their first aeroplane and that the method is still accepted as the most practical one and is almost identical with their original work.

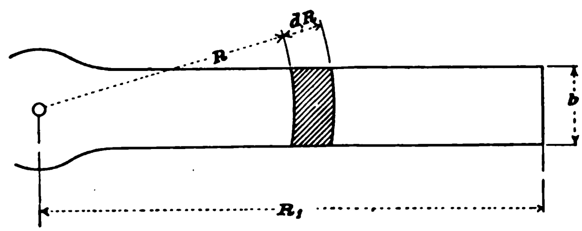


FIG. 15 DIAGRAM TO ILLUSTRATE METHOD OF CALCULATING THRUST

69 We have emphasized elsewhere in this paper the benefits which may accrue to the ventilation engineer from application to his problems of airplane-propeller design principles, and we therefore give the following example of the method mentioned above.

70 It is a special case of airplane propulsion, wherein the forward velocity is zero; and applies to a propeller rotating at a fixed point, as is the case in ventilation. The torque and thrust (i.e., the fan horsepower and integrated pressure head) are shown by the following simple calculation to be easily determined if aerofoil coefficients are first secured in the wind tunnel.

71 Consider an elemental strip of length dR and of width b. (See Fig. 15.) It is necessary to study the thrust and power absorbed by this strip and then by integration to obtain the thrust and power absorbed for the whole propeller.

THRUST

72 In order to make an intelligent study of the problem of static thrust without becoming involved in complicated mathematics we may confine ourselves to a series of propellers with comparatively low blade angles and having a uniform blade width from tip to boss, and a uniform lift coefficient K_y from tip

to boss. It will be noted that no value is chosen for the blade angle and these angles will have to be worked out later on.

73 A further assumption, which is an approximation only, is that the lift-drag ratio γ varies according to the following formula:

$$\gamma = \frac{14R}{R_1} + 8 \quad \ldots\ldots\ldots\ldots\ldots \quad [1]$$

where R_1 is the radius at the tip of the blade and R is the radius at any point under consideration. Strictly speaking, the equation for thrust will be:

$$T = L\cos\phi - D\sin\phi \quad \ldots\ldots\ldots\ldots \quad [2]$$

However, as D is small compared to L and $\sin\phi$ is small compared to $\cos\phi$, the product $D\sin\phi$ will be very small compared to the product $L\cos\phi$, and the term $D\sin\phi$ may be ignored. Also since ϕ is small we may take $\cos\phi = 1$. Then we shall assume that $T = L$. But

$$dT = \rho/gK_y b\, dRW^2 \quad \ldots\ldots\ldots\ldots \quad [3]$$

where b = blade width at the point considered, bdR the area of an elemental strip, and W the resultant velocity at the radius.

$$W^2 = V^2 + (\tfrac{1}{2}V_1)^2 = (2\pi NR)^2 + (\tfrac{1}{2}V_1)^2 \quad \ldots\ldots\ldots \quad [4]$$

Hence

$$dT = \rho/gK_y bdR[(2\pi NR)^2 + (\tfrac{1}{2}V_1)^2] \quad \ldots\ldots\ldots \quad [5]$$

74 It now becomes necessary to find the value of V_1, which is the slip-stream velocity. It should be noted here that the usual assumption has been made that the slip-stream velocity before the disk is one-half the slip-stream velocity after the disk.

75 In order to estimate the slip-stream velocity we may make use of the fundamental equation for impact $F = MV_1$. In this case $T = MV_1$. Evidently the mass of air handled per second will be the volume per second multiplied by the mass density of air ρ/g. But the volume is equal to the cross sectional area A times the velocity $\frac{1}{2}V_1$. Hence the formula becomes:

$$T = \rho/g\, \tfrac{1}{2}V_1^2 A$$

or

$$dT = \rho/g\, \tfrac{1}{2}V_1^2 dA$$

but

$$dA = 2\pi R\, dR$$

hence

$$dT = \rho/g\, V_1^2 \pi R\, dR \quad \ldots\ldots\ldots\ldots \quad [6]$$

76 As a first approximation in order to compute V_1 only, we may ignore the second term of Equation [5] and assume

$$dT = \rho/gK_y b(2\pi NR)^2 dR \quad \ldots\ldots\ldots\ldots \quad [7]$$

Hence

$$\rho/gK_y b(2\pi NR)^2\, dR = \rho/gV_1^2 \pi R\, dR$$

$$V_1 = 2N\sqrt{K_y b\pi R} \quad \ldots\ldots\ldots\ldots \quad [8]$$

Substituting in Equation [5] we have

$$dT = \rho/g K_y b[(2\pi NR)^2 + (N^2 K_y b\pi R)]dR \quad \ldots\ldots\ldots \quad [9]$$

Integrating and substituting limits, we get for each blade

$$T = (2\pi NR_1)^2 K_y b \; \rho/g(\tfrac{1}{3}R_1 + K_y b/8\pi) \quad \ldots\ldots\ldots \quad [10]$$

77 If we assume the blade width $b = R_1/6$ which is very good practice, and assume $K_y = 0.3$ for standard air at 760 mm. and 15 deg. cent., Equation [10] will reduce to:

$$T \text{ (per blade)} = 0.00157N^2R^4 \quad \ldots\ldots\ldots \quad [11]$$

$$T \text{ (for two blades)} = 0.00314N^2R^4 \quad \ldots\ldots\ldots \quad [12]$$

or

$$T = 0.0001961N^2D^4 = 1.961N^2D^4 \times 10^{-4} \quad \ldots\ldots\ldots \quad [13]$$

78 This is the old familiar formula which forms the basis of the Eiffel chart and later methods of interpreting experimental results. In this case, however, we have evaluated the thrust coefficient from aerofoil tests, thus obviating the need of experimental model propeller tests.

TORQUE AND POWER ABSORBED

79 In the above discussion we have been using the velocity

$$W = \sqrt{V^2 + \tfrac{1}{4}V_1^2}$$

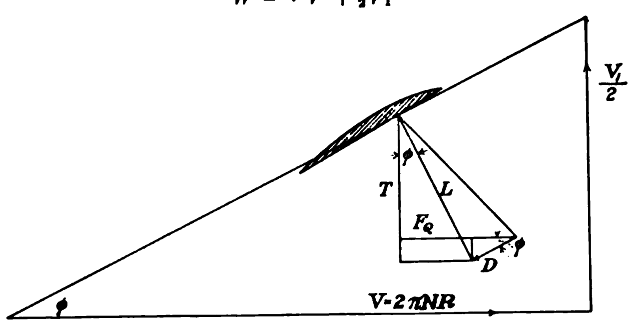


FIG. 16 DIAGRAM TO ILLUSTRATE METHOD OF CALCULATING TORQUE AND POWER ABSORBED

It will be of interest to note the effect on the precision of this method if the term $\tfrac{1}{4}V_1^2$ were ignored. In the final form of Equation [10] we have the factor $\tfrac{1}{3}R_1 + K_y b/8\pi$, where $\tfrac{1}{3}R_1$ results from V, and $K_y b/8\pi$ results from $\tfrac{1}{2}V_1$. If we substitute the values assumed for K_y, R_1, and b the expression reduces to the following:

$$\frac{1}{3}R_1 + \left(0.3 \times \frac{R_1}{6} \times \frac{1}{8\pi}\right) = R_1(0.333 + 0.001987)$$

the second term is only 0.6 per cent of the first term; therefore, in order to simplify the discussion of the torque and power absorbed we may assume $W = V$.

80 Referring now to Fig. 16. If F_Q represents the torque force then:

$$F_Q = L \sin\phi + D\cos\phi \quad \ldots\ldots\ldots \quad [14]$$

If P represents the energy absorbed per unit of time

$$P = (L \sin \phi + D \cos \phi)\, 2\pi NR$$

or

$$dP/dR = (L \sin \phi + D \cos \phi) 2\pi NR \quad \ldots\ldots\ldots \quad [15]$$

$$L = \rho/g K_y b (2\pi NR)^2$$

$$D = L/\gamma = \rho/\gamma g K_y b (2\pi NR)^2 \quad \ldots\ldots\ldots\ldots \quad [16]$$

81 It now becomes necessary to introduce the assumption of Equation [1]:

$$\gamma = \frac{14R}{R_1} + 8$$

$$D = \frac{R_1}{14R + 8R_1}\left[\frac{\rho}{g} K_y b (2\pi NR)^2\right]$$

Then

$$\frac{dP}{dR} = \frac{\rho}{g} K_y b\, (2\pi NR)^3 \left(\sin \phi + \frac{R_1 \cos \phi}{14R + 8R_1}\right) \quad \ldots\ldots \quad [17]$$

82 For small angles $\sin \phi$ may be taken equal to $\tan \phi$, therefore

$$\sin \phi = \tan \phi = \frac{\frac{1}{2} V_1}{V} = \frac{N\sqrt{K_y b \pi R}}{2\pi NR} = \frac{1}{2}\sqrt{\frac{K_y b}{\pi R}} \quad \ldots\ldots \quad [18]$$

$$\cos \phi = \sqrt{1 - \sin^2 \phi} = \sqrt{\quad - \frac{K_y b}{4\pi R}} \quad \ldots\ldots\ldots\ldots \quad [19]$$

However, it may be shown that the error in assuming $\cos \phi = 1$ for the range of small angles under consideration is less than 1 per cent. Taking $\cos \phi = 1$, we have

$$dP/dR = \rho/g K_y b (2\pi NR)^3 (\tfrac{1}{2}\sqrt{K_y b/\pi R} + R_1/\overline{14R + 8R_1})$$

by integration we obtain the following

$$P = \frac{\rho}{g} K_y b (2\pi N)^3 \Bigg|_0^{R_1} \left(\frac{1}{2}\sqrt{\frac{K_y b}{\pi}} \times \frac{R^{3.5}}{3.5}\right) +$$

$$\frac{R_1}{14}\left[\frac{R^3}{3} + 0.571 R_1 (R_1 R - R^2 - \overline{0.571 R_1^2} \log \overline{R + 0.571 R_1})\right]$$

Substituting $K_y = 0.3$ and $b = R_1/6$ this reduces to the form

$$P = 9.17 N^3 R^5 \times 10^{-4}$$
$$= 2.86 N^3 D^5 \times 10^{-5}$$

This is again the familiar form of power equation and we have been able to evaluate the coefficient for power absorbed.

83 From the above example it will be seen that the performance of certain types of static propellers, or fans, may be calculated to a very good accuracy if only we have the proper aerofoil coefficients from wind-tunnel tests. The ventilation engineer will be quick to appreciate the advantages to be had from adapting this method to the prediction of horsepower and pressure heads of his propeller fans.

DISCUSSION

Edward P. Warner. The work done by the authors in the visualization of vortices and of air flow in general has amply justified the construction of the high-speed wind tunnel, and is one of the most striking and important aerodynamic developments of the last five years. Aside from the work on air flow, however, a large tunnel operating at moderate speed is better than a small one giving a very high speed. The lower speed is better because the same value of VL can be secured with the expenditure of less power, because the supporting spindle can be made smaller in proportion to the size of the model, with a resulting reduction of relative interference, and because the relative accuracy of construction can be greater for a large model than for a small one.

If the speed of flight, or of a model test, is so high that compressibility is not negligible, the coefficient of resistance becomes a function of V/C, where C is an index of compressibility of the fluid medium, as well as of VL/γ. It therefore is not fair to expect that the coefficients obtained in a test at very high speed (over 300 mi. per hr.) will be closely similar to those found at moderate speed on a larger model giving the same value of VL.

It is impossible to simulate very satisfactorily the conditions on propeller blade elements. In order that the effect of compressibility in the model test may be the same as on the propeller, the speed of test must be equal to the peripheral field, and this condition is substantially realized in the McCook Field tunnel. The further condition that VL should approach as nearly as possible the full-scale value is far from being fulfilled, and the high-speed tunnel at McCook is inferior in this respect to numerous others in this country and abroad.

Walter C. Durfee. The kind of water vapor condensation or fog which was used by the authors can be easily reproduced by sucking strongly on a bottle which has been recently washed inside with water. A fog can be seen to form inside the bottle with great suddenness. It can be made to appear and disappear and flicker very rapidly according to the suction. The photographs shown by the authors indicate that such a fog can be produced in much less than a ten-thousandth of a second, that is to say in less time than it took the air to travel a distance perceptible on their photographs at the rate of about 500 m.p.h. Although a process of evaporation is not necessarily the reverse of condensation, owing to readjustments of the size of the particles after their formation, it seems probable that the fogs used by the authors were able to vanish very suddenly. Because of these considerations, the results require careful judgment in their interpretation. It is entirely possible for a fog to appear stationary in the midst of an atmosphere which is in rapid motion. The fog simply grows at one end and vanishes at the other in such a way as to maintain its position in spite of the wind. The fog then simply marks the location of air which has been shortly beforehand at a low temperature. Ordinarily, as in the case of the experiment with the glass bottle, it is not necessary to supply any nuclei for condensation in addition to those already present in the atmosphere, so that it seems probable that the nuclei furnished by the model wing in this experiment were not of chemical or electrical nature, but were certain microscopic regions of low pressure existing in the centers or axes of various small eddies. The S-shape swirls mentioned in Par. 52 probably were pairs of eddies having a similar direction of rotation. Such pairs naturally rotate around each other so as to keep approximately the same distance apart, and appear as a single unit.

Regarding the great value of studies which the authors are now making, and propose to make, with their stroboscope analysis of vortex action, the writer believes that the statement in Par. 56 deserves considerable attention. If it is possible for one end of a wing to fail while the other continues to lift, it must be expected that accidents might happen from this cause, as one would expect an airplane to turn over sideways under such circumstances. The statement that this phenomenon is possible because of the center support dividing the aerofoil in two parts, would not offer much consolation even if no other causes existed, because most airplane wings are similarly divided into two parts. Consequently it is a problem of practical importance to discover the reasons for the breaking down of the so-called high-lift type of flow, and the means of maintaining this type of flow if any means are possible.

In connection with the above study, and other studies, the writer believes that some changes in the terminology would be useful. The term "trailing vortex" might well be abandoned in the future writings, because in Par. 42 and its footnote the word "trailing" is used with reference to the tip vortex, which alone has the appearance of trailing in the sense of hanging on behind. In the footnote the word "trailing" is evidently used with reference to the eddies produced by the trailing edge, and under another definition of the word "trailing" the word could equally be applied to eddies produced by surface friction which are likely to have the appearance of trailing in the sense of following or pursuing. The words "diameter of vortex" and "distance to which vortex phenomenon may extend" might well be abandoned because the vortex does not have a diameter although there may be a diameter to the portion of it which is rendered visible. Probably also it would be better to say "sheet of vortices" rather than "vortex sheet," as this latter term has a special dictionary meaning. A further improvement in terminology might be worked out so as to emphasize the continuity of the vortices or eddies. The rear edge vortices, for example, will probably be found to join on as branches of the tip vortex or group of vortices. The tip vortex itself will be found to have a relation to the circulation which exists normally around the wing. When this conception of continuity of vortex is recognized, it will probably contribute to the success of any study which is connected with the second critical speed mentioned by the authors, or with the building up and breaking down of the conditions of flow which are considered to be desirable in flight. This will probably be more especially true if the tip vortex and the circulation about the wing are considered as composed of numerous smaller elements grouped together.

Regarding the internal stress and the breaking down of the air flow mentioned in Par. 67, it seems quite certain that the building up of the desired vortex system must be due to some of the special properties of the air, since it is commonly believed that a vortex would not be generated in a "perfect" fluid. On the other hand, the tendency for such vortices to break down calls to mind an interesting experiment published by Lord Kelvin. As the writer remembers this case, a volume of water was set in rotation in a glass globe in such a manner that the water climbed up on the inside and formed the usual parabolic depression in the center. This mass was stirred vertically in such a manner as not to interfere with the rotation in any way. The turbulance in the water caused a striking rearrangement of the motion which began to more nearly resemble a perfect vortex so that the central depression in the surface of the water took on the shape of a deep hole. In the case of air it might be imagined that a similar rearrangment of the tip vortex group, due to turbulance or other causes, might produce a very high vacuum, more or less dangerous to the stability of the flow system.

Regarding the design of the wind channel and the outside straightener, Par. 25, the writer thinks that the authors might furnish some interesting information concerning the need for these straighteners. If they were used to counteract the rotation of the air in the room, due to the action of the fan, one would think that they might have been placed at the other end of the apparatus. If they were used to counteract the effect of the rotation of the earth, one would think that this might have been done by placing the tunnel due east and west. If they were used on account of some twist transmitted from the fan through the tunnel by internal stress of the air, say as those mentioned in Par. 67, one would be interested to know why the inside straighteners were not sufficient. If the straighteners were used to take care of eddies caused by the furniture of the room, this would be an interesting feature.

M. C. STUART. In connection with the design of air propellers by the methods depending upon the law of similarity, attention is invited to the paper on Air Propeller Performance and Design by the Specific Speed Method, by M. C. Stuart, Journal of the American Society of Naval Engineers, Vol. 30, No. 2, May, 1918.

The specific speed method of treating the subject of geometrically similar air propellers is similar to the specific speed method of treatment used for many years by engineers in the design of hydraulic turbines, and later in the design of centrifugal pumps and fans.

Briefly, the specific speed method places the propeller upon the basis of diameter and rotational speed required to produce a unit of velocity of forward movement and a unit power. Then, by means of the specific speed formula, the speed and diameter for any other velocity and power may be directly computed. In view of the fact that methods similar to this have been used so successfully in connection with hydraulic turbines and other rotating machinery, it is believed that a study of this method in connection with the design of air propellers may be made with profit by those interested in the subject.

MORGAN BROOKS. The work done by the authors is not only interesting but will be of great value in the design of airplanes and propellers. The definiteness with which discontinuity is fixed by the experiments is striking. We have not had access to very recent experiments and the speed of discontinuity has not been well fixed, but when it is fixed in this way it is more conclusive than when arrived at by the point by point method.

It is interesting to compare these moving pictures with the work of the oscillograph which, in electrical work, has shown irregularities not found by point-by-point tests. The method of the moving picture will supersede the slower method of individual tests.

Has the critical speed anything to do with the different action of the flow of air on the upper and lower surfaces of the wing? The laws of pressure and of suction must necessarily be so different that discontinuity might result at some velocity which may be analyzed by this method.

There is some suggestion toward the end of the paper in regard to the influence of this discontinuity on very high flights. It will probably have greater effect upon the tip of the propeller. We are now reaching speeds of six- or seven-hundred feet per second at the tip of the propeller. Discontinuity must come at the velocity of sound, eleven-hundred feet per second at sea level and other speeds for rarer air. Hence propellers reaching seven- or eight-hundred feet of actual movement in air may be approaching this point of discontinuity and require a different design.

The Authors replied to the discussion as follows:

1 The fog seems to persist as far as the fan — 16 ft. Light conditions have not yet been arranged to determine whether fog exists down stream from the fan.

2 The vortex formations in the open throat are affected by local conditions and by the absence of the intake straightener. The flow, while of smoothness that can scarcely be excelled, is still unsteady, and complete cycles are somewhat embarrassed by local conditions.

3 In regard to unsymmetrical forces to right and left of fuselage in an actual airplane, it may be said that the changes in flow occurring at the "burble" point or stalling angle of conventional airplane wings is probably not so violent as in the propeller sections.

4 The straighteners are used to off-set the whirl-tendency formed at entrance due to air moving radially; here a miniature cyclonic whirl sets up due to radial inward flow toward regions of low pressure. It is not necessary to explain the phenomenon by the rotation of the earth. Spin has been observed in a direction contrary to fan rotation.

5 The moving pictures[1] and photographs are vastly inferior to the actual view of the phenomenon in the wind tunnel. The effect

[1] The moving pictures referred to were shown at the session at which this paper was presented. — Ed.

of discontinuity has already changed design to some extent and it is believed that we are now in the realm of fluttering on the propellers. The tips of propellers are analagous. The speed of the wing is far below the speed being used in the tests and it is doubtful if this phenomenon will affect an airplane greatly.

6 The discussion presented by Mr. Warner correctly summarizes the purpose of the design of the McCook Field wind tunnel. The aim has been not to duplicate existing data, but to initiate original investigations. The conventional wind tunnel fails us in the researches which we have laid out and which apply primarily to propeller tip phenomena. As is well known, every wind tunnel is a compromise between high speed and large size, and falls short of full-scale conditions. No case is known wherein the *VL* product for full-scale propellers is so nearly approximated as at McCook Field, (*VL* reaches one-half full-scale value), and we therefore assume that Mr. Warner has in mind airplane wings rather than propellers when he says the McCook Field tunnel is inferior in respect of *VL* product attainable.

7 Regarding compressibility in this tunnel: As shown by the charts, density questions are eliminated by referring both K_y and V to "standard air" conditions; and, in effect, K_y is plotted against the energy content of the air. The graphs, when plotted to absolute velocity as abscissæ, on the assumption of true adiabatic expansion, are less convenient than as shown. They are of similar character, but somewhat stretched out along the direction of the velocity abscissa.

8 We have not yet published our data on coefficient of resistance.

9 As concerns the coefficient of lift, the authors regard the law of dynamic similarity as unimpeachable only within certain ranges. As our paper has been at pains to show, the experiments recorded cast serious doubt on the universality of the law of dynamical similarity if it be assumed that the viscosity does not change much. The experiments definitely show that discontinuity of flow depends more upon velocity than on the product *VL*. The data has been published in simple familiar form such that the coefficient may, by interested readers, be readily transformed for comparison with any desired system. It is not profitable to compare the McCook Field researches with others in terms of the product *VL*, since it is in large measure this product which we are studying.

10 It is desired to give full and appreciative credit to Mr. C. P. Grimes, who was in direct charge of the wind tunnel and with whom the writers worked jointly prior to his departure from Govern-

ment Service. His ability has been manifest not only at the time of his service with the Government, but also in later use of the apparatus which he devised. The construction of the wind tunnel was carried on jointly with Mr. Grimes. A particularly meritorious feature of Mr. Grimes' work is the cone straightener; we found after his departure that the straightener system devised by him increased the efficiency 10 per cent. The balance is also attributable to Mr. Grimes, and represents a departure from conventional design which promises well for further development.

No. 1740

ANALYTICAL THEORY OF AEROPLANES IN RECTILINEAR FLIGHT AND CALCULATION OF THE MAXIMUM CRUISING RADIUS

By A. Rateau,[1] Paris, France
Honorary Member of the Society

Theory of Horizontal Rectilinear Flight of Aeroplanes. Description of underlying basic principles, together with their application to the solution of certain important problems.

Characteristic Relations Connecting the Main Factors Governing the Operation of the Propellers. In these the author expresses practically all functions in terms of slip or functions thereof.

Characteristic Functions and Coefficients of an Aeroplane. These are expressed in terms of weight of the aeroplane, incidence and four specific coefficients which have to be determined by measurement and computation.

Minimum Consumption of Fuel and Oil per Kilometer of Flight. The determinations of consumption which are made are dependent on the altitude of flight; tables are given showing the results of computations for what the author calls the "Aeroplane-Type."

Cruising Radius of Aeroplanes. Formulæ are given for determining the cruising radius and it is shown that with the present state of aeronautical engineering the cruising radius in still air is theoretically limited to 7000 *km.* (4350 *miles*) *and that even this can be achieved only by flight at elevations ranging from* 16,000 *to* 25,000 *ft.*

THE purpose of this paper is to give a brief outline of the fundamental principles from which the writer has succeeded in developing the analytic theory of the rectilinear flight of aeroplanes. An exposition of these principles together with examples of their application was presented to the Academy of Sciences in June and July 1919, and February 1920. More detailed information may be found in the Annals of the Academy as well as in a book which will shortly be published.

2 It is also desired to undertake, by way of illustration and

[1] Mem. Institut de France, Chairman, Board of Directors, Rateau Battu and Smoot Company.

Presented at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers.

application of the theory, the determination of the altitude of flight at which the consumption of fuel and oil is a minimum, and similarly the maximum cruising radius which aeroplanes as now constructed can attain.

3 It is customary to study problems relative to aeroplanes by geometric methods involving the construction of characteristic curves. Such methods, however, are not followed in the present instance, not because they are considered inadequate, as in fact the writer has successfully applied them in the study of hydraulic turbines, but because they are not sufficiently precise and are very difficult of application. For this reason the analytic method, which permits the immediate calculation of the element under investigation, is preferred.

4 The theory to be presented is based on certain characteristic equations which it is first necessary to establish, and it is believed that the simple forms in which they are given are for the most part new.

5 The problem of the aeroplane is highly complex. In order to treat it correctly it is necessary to take into consideration not only the aeroplane and its motor but the propeller as well, the slip variations of which have a very considerable significance.

6 The following conditions are assumed:

a That the motor operates at constant compression, without supercharging by turbo-compressor, and that the carburetor is provided with an altitude regulator maintaining the mixture constant;

b That the propeller is of constant pitch, not subject to deformation, and that it is keyed on the shaft of the motor.

7 The theory, as a matter of fact, is applicable also to the case of the geared propeller, but a direct propeller drive is here assumed in order to avoid complicating the formulæ by taking the gear reduction into consideration.

8 The various characteristic formulæ of the motor, the propeller and the aeroplane involve the specific weight of the air in which the aeroplane moves, and it is the specific weight of air upon which depends principally the power of the motor, the tractive effort of the propeller, and the lifting power and the resistance to flying of the aeroplane. In order, therefore, to study the influence of altitude in flight, it will be necessary to begin by determining the relation between this altitude Z and the specific weight ω of the air.

CHARACTERISTIC FORMULA OF STANDARD ATMOSPHERE

9 The atmosphere is changing incessantly. The specific weight of air varies continuously, more so than the pressure. Calculations cannot be effected except by referring to a fictitious, mean atmosphere, or as it is often called, "standard" atmosphere.

10 By analogy with Halley's formula, which expresses the relation between altitude and the logarithm of the ratio of the baro-

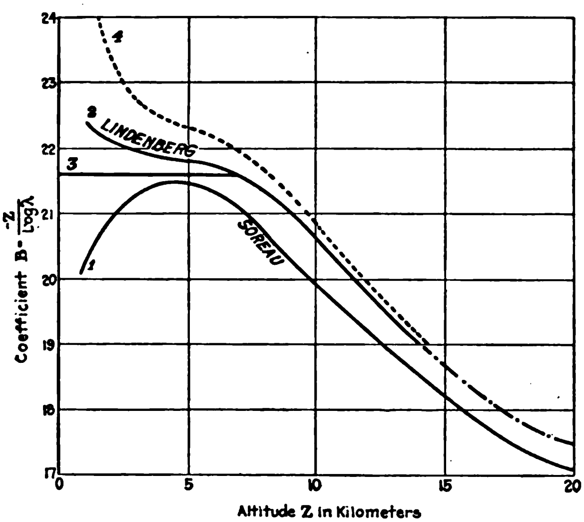


FIG. 1 VARIATION OF ATMOSPHERIC COEFFICIENT B WITH ALTITUDE

metric pressure at this altitude and that at sea level, the following expression may be written:

$$Z = -B \log \frac{\omega}{\omega_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

ω and ω_0 being the specific weights, in kilograms per cubic meter, at altitude Z, in kilometers, and at sea level, respectively, and the logarithm being taken in the usual system, of base 10.

11 Fig. 1 shows the variations of coefficient B according to the most recent observations with sounding balloons, while Curve 1 indicates results of the analysis made by R. Soreau.[1] It will be seen that B reaches a maximum of 21.45 for a maximum value of Z = 4.4 km. nearly, and drops to 17 for Z = 20 km.

[1] Comptes rendus des Séances de l'Académie des Sciences, December 1, 1919.

12 Curve 2 corresponds to the figures obtained in experiments at the Observatory of Lindenberg, Germany, in the course of ten years of study, after making the necessary correction for the altitude, which is 116 meters. Curve 4 gives the same values without making such correction.

13 The curves 1 and 2 do not coincide. Below 4 km. their slopes differ considerably, but above 5 km., they are substantially parallel. With these results so little in agreement and the lack of any further observations to throw more light on the matter, the best that can be done is to adopt $B = 21.6$, represented by the horizontal line 3, from $Z = 0$ to $Z = 6$ km., and to decrease B for higher values of Z but always maintaining its value within the interval of the two curves.

14 From $Z = 6.5$ km. to $Z = 15$ km. it is possible to substitute, without material error in place of Soreau's curve the simple function

$$B = 22.75 + 5.5 \log \frac{\omega}{\omega_0}$$

which connects Z to $\log \dfrac{\omega}{\omega_0}$ by an expression of the second degree.

15 This will suffice for some time, because already last year's altitudes of 7 km. have been passed and it is not presumable that consideration will be given soon to altitudes greatly beyond 12 km., at which altitude it is now desired to effect long flights at great speeds, enclosing the pilot and the passengers in a suitable cabin where a pressure of the order of two-thirds of an atmosphere absolute will be maintained.

CHARACTERISTIC FORMULÆ OF MOTOR

16 What is of importance to consider in a motor is not the power, which varies greatly with the speed of rotation, but the couple it gives. Carburetion being properly regulated, this couple changes so slightly with the speed of rotation n that its variation may be disregarded, at least in a preliminary approximation. Corrections will be made later if necessary.

17 The couple, Γ, on the other hand, is influenced enormously by the specific weight of the air supplied. These are not directly proportional to each other by reason of various passive resistances. As they are practically constant, however, it is possible to write

$$\Gamma = A\ (\omega - \upsilon) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

A and v being two specific constants of the motor in question, the latter varying slightly from motor to motor and approaching the value of 0.12.

18 It may well be said that Equation [2] is far from being exact, for besides the fact that A increases slightly when the speed n decreases, it is also greatly influenced by the carburetion, v is also modified by a great number of circumstances, particularly by the character of the lubrication; but it is not possible to do better with our present lack of knowledge of these factors. The equation shows, for example, that for $\omega = v = 0.12$, corresponding to $Z = 17.5$ km., the motor will not give a useful couple.

19 On the other hand, the consumption of fuel and oil of the motor varies with n and with ω. In regard to the fuel, if the carburetor is well regulated the consumption is proportional to the total weight of the air taken in and consequently to n and to ω. The oil consumption depends a great deal on the nature of the oil and on the temperature of the air supplied to the motor, increasing with the latter. It also increases with the speed n of rotation and more rapidly than this speed. But it is almost independent of the specific weight ω of the air supplied, although it has a tendency to decrease when ω decreases, probably because at the same time the temperature diminishes.

20 It is impossible to take exact account of all these variations, but we do not go far wrong when we assert that the oil consumption is proportional to n and independent of ω.

21 The total consumption of fuel and oil is therefore proportional to $\omega + o$, o being a constant which for non-rotating motors lubricated with castor oil has the value of 0.08, this value becoming about 0.1 for engines lubricated with mineral oil, and being much larger for rotary motors.

22 As the couple is proportional to $\omega - v$, it is seen that the consumption of fuel and oil per horsepower is proportional to $\dfrac{\omega + o}{\omega - v}$.

If E_0 represents the consumption per horsepower when operating on the ground, then at any altitude the consumption E of fuel and oil per horsepower is

$$E = E_0 \frac{\omega_0 - v}{\omega_0 + o} \times \frac{\omega + o}{\omega - v} \quad \text{. . . . . [3]}$$

23 For example, if $\omega_0 = 1.22$, $v = 0.11$, $o = 0.09$, at $Z = 5.5$

km. where $\omega = 0.5\ \omega_0$, then $E = 1.19\ E_0$, which means that there is an increase of consumption per horsepower of nearly 20 per cent.

CHARACTERISTIC FORMULÆ OF THE PROPELLER

24 The following discussion on the propeller is an abstract of a theory founded on the hypotheses presented by the writer in 1900 before the International Congress of Architecture and Naval Construction. The demonstrations will appear in a book which is now on the press.

25 The chief variable is the *true slip*, σ, that is, the relative loss of the actual forward movement of the propeller with reference to that which it would have if it rotated in air without exerting any tractive effort.

26 Let A be the actual advance of the propeller per revolution and H the advance which would give zero thrust. Then

$$\sigma = 1 - \frac{A}{H} \quad \ldots\ldots\ldots\ldots \quad [4]$$

H is the *effective pitch* of the propeller in question, which is greater than the geometric pitch of the active face which we denote by the term "nominal pitch" and which is usually the only one taken into consideration. The difference is of the order of 20 per cent with the usual shapes of cylindrical sections of propeller blades.

27 The effective pitch H is the only one that should be considered. Unfortunately, its definition is of a physical character and it is impossible to measure it directly on the propeller. It may be determined, however, by experiment or by comparison with similar propellers.

28 The nominal pitch is of no importance whatsoever. It is the least interesting of all and has only one advantage and that is that it can be easily measured. In order to clearly present the properties of propellers it must absolutely be replaced by the effective pitch, which, moreover, is a constant characteristic of the propeller no matter what the speed of rotation may be, as has been proved in actual practice.

29 Theory shows that the tractive force F of the propeller in kilograms and its counteracting couple Γ in kilogram meters are expressed by the relations

$$\left.\begin{aligned} F &= b\omega n^2 f(\sigma) \\ \Gamma &= \frac{bH}{2\pi}\,\omega n^2 \varphi(\sigma) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots \quad [5]$$

where $f(\sigma)$ and $\varphi(\sigma)$ are the functions of slip and of certain coefficients of the propeller which will be mentioned later on, and where b, which I shall call the *factor of thrust*, has for its expression in the case of aeroplane propellers with narrow blades

$$b = \frac{2k\pi}{g} HM \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [6]$$

where π is the ratio of circumference to diameter, g the gravitational constant in metric units $(= 9.81)$, k a numerical coefficient of the order of 2, M the moment, referred to center of propeller, of the projection of the active surface of the blades on a plane perpendicular to the axis. The units in the case of H and M are the meter and the square meter, respectively.

30 The function $f(\sigma)$ is of the form $\sigma(1 - e'\sigma)$ where e' is a coefficient depending on the ratio p of the pitch H to the peripheral diameter D of the propeller and decreases in a certain proportion to the decrease of the ratio above referred to. When p is less than unity (and this is the case with practically all aircraft propellers) e' is less than 0.08. Since, on the other hand, the slips σ as a rule vary only from 0.2 to 0.4, one can see why the corrective term $e'\sigma$ varies only by about 0.016 in actual practice and hence may be entirely neglected when subtracted from unity, which is what the author proposes to do. At the same time whenever necessary the complete formula permits carrying out calculations with the necessary degree of precision, when desired.

31 As regards the expression $\varphi(\sigma)$, which is of basic importance, it has the form

$$\varphi(\sigma) = a + \sigma - e\sigma^2 \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [7]$$

where e always has a value in the neighborhood of 0.5 and a of 0.03 and vary from one propeller to another according to the shape and roughness of the surface. It is only this a that is effected by losses in the propeller due to shocks, friction and turbulence of the fluid, and it is connected with the coefficient ϵ (loss of relative velocities across the propeller) by the relation

$$a = \epsilon\left(\frac{p}{\pi r} + \frac{\pi r}{p}\right)^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; [7a]$$

where r is the ratio of a certain average radius to the radius of the periphery.

32 If we assume that $e = 0.5$, we have

$$\varphi(\sigma) = a + \sigma - \frac{\sigma^2}{2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7b]$$

and hence

$$\psi = \frac{\varphi(\sigma)}{f(\sigma)} = 1 - \frac{\sigma}{2} + \frac{a}{\sigma} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

This function ψ of the slip will constantly occur in our aeroplane formulæ.

33 The efficiency of the propeller ρ, if we denote by v the velocity of advance, is

$$\rho = \frac{Fv}{2\pi n\Gamma}$$

If we replace v by $nH(1-\sigma)$, and F and Γ by their expressions taken from Equation [5], we shall have

$$\rho = \frac{f(\sigma)\,(1 - \sigma)}{\varphi(\sigma)} = \frac{1 - \sigma}{\psi} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

34 Attention is called to the fact that in this expression of the efficiency of the propeller in which ψ may be used in the form given by Equation [8] only the slip σ and the coefficient of losses a are used.

35 All these novel and fundamental formulæ are excellently confirmed by experiments as precise as one could wish for in a subject of such complexity as the present one.

CHARACTERISTIC FUNCTIONS AND COEFFICIENTS OF THE AEROPLANE

36 For a given aeroplane the essential variables are the total weight P in kilograms and the incidence α, which is the angle in degrees between the wings of the plane and its trajectory. Further on I shall give a more exact definition of incidence.

37 In a rectilinear horizontal flight the weight P is balanced by the lifting thrust of the air

$$P = Y\omega v^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [10]$$

and likewise we have the expression for the resistance to forward motion R

$$R = X\omega v^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [11]$$

In these expressions the coefficients Y and X for a given aeroplane are merely functions of the incidence α.

38 With the exception of the cases of discontinuity and fracture which are revealed in the instance of wings of ordinary shapes in the characteristic curves with an incidence lying always in the neighborhood of 1 deg. (a discontinuity to which I shall call special attention elsewhere and which corresponds to a change in régime of the flow of air around the wings), experience proves that the curve of Y plotted as a function of α is within useful limits practically a straight line, while the curve of X is practically a parabola.

39 We have therefore to write

$$\left.\begin{array}{l} Y = Y_0(1 + \eta\alpha) \\ X = X_0(1 + \xi\alpha^2) \end{array}\right\} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [12]$$

Y_0, X_0, η and ξ are four specific coefficients of an aeroplane. Strictly one should consider two groups of these four coefficients, one applicable to small incidences, inferior to the incidence at which discontinuity occurs, and the other at incidences above this limit. This distinction, however, is disregarded in the present paper because the small incidences in the neighborhood of zero, such as those, between -1 deg. and $+1$ deg., are met with only in flight at low altitudes in aeroplanes having a very high ceiling. I shall restrict myself, therefore, to indicating that in order to be strictly precise one has to make such a distinction.

40 Of course, Equations [10], [11] and [12] apply to the aeroplane as a whole where sustentation is produced mainly by the wings though also to a small extent by the fuselage and the rear rudders. It is reduced by the inverse thrust of the elevators and this reduction is the greater the greater the incidence.

In the same manner the resistance R comprises not only the resistance of the wings and rudder but also the parasite resistance of the fuselage, radiators, struts, pilot, etc., the relative importance of which is particularly great at small incidences.

41 Judging from Equation [12], the origin of incidence lies not at the *average chord* of the wings but along the direction which reduces to a minimum the resistance to the forward motion of the aeroplane as a whole. This direction may materially differ (by as much as 2 deg. or more) from the direction of the average chord and it is important to bear this in mind in order to apply the formulæ properly.

42 As an example, I shall consider a fictitious aeroplane, which

I shall designate by the name of Aeroplane-Type, having as characteristic coefficients

$$Y_0 = 0.5897 \qquad \eta = 0.3$$
$$X_0 = 0.1204 \qquad \xi = 0.017$$

These figures correspond very nearly to the Bréguet army plane having 50 sq. m. of wing surface. They do not, however, apply exactly to that machine, because of our neglect to take into consideration the true characteristics of that type of apparatus.

OPTIMA ANGLES

43 In accordance with Equations [10] and [11] we have at once

$$\frac{R}{P} = \frac{X}{Y} \qquad [13]$$

44 For any weight P the incidence α_m which gives the minimum of resistance to forward motion is that which corresponds to the minimum of X/Y: this minimum is called the "fineness" of aeroplane.

45 In accordance with Equation [12], one must try to obtain the minimum of $(1 + \xi\alpha^2)/(1 + \eta\alpha)$, or, what is the same thing, the minimum of its logarithm. Equated to zero the logarithmic derivative gives an equation of the second degree, namely,

$$\alpha^2 + \frac{2\alpha}{\eta} - \frac{1}{\xi} = 0 \qquad [14]$$

the useful root of which is

$$\alpha_m = \frac{1}{\eta}\left(-1 + \sqrt{1 + \frac{\eta^2}{\xi}}\right) \qquad [15]$$

46 This shows that this incidence — the "optimum" incidence — depends only on the coefficients η and ξ. As regards the minimum of X/Y, or rather the maximum of Y/X, its value is determined by

$$\left(\frac{Y}{X}\right)_m = \frac{Y_0}{2\,X_0}\left(1 + \sqrt{1 + \frac{\eta^2}{\xi}}\right) \qquad [16]$$

47 In particular, with the data given above for the Aeroplane-Type, we find from these equations that

$$\alpha_m = 5.03 \text{ deg. and } \left(\frac{Y}{X}\right)_m = 8.59$$

In the same manner a special optimum incidence may be determined for every variable which it is desired to keep as maximum or minimum.

48 Let us take another example. Let us find the minimum of tractive power that it is necessary to maintain with given values of P and ω.

49 The tractive power is $Rv = X\omega v^3$. Comparing it with Equation [10] in order to eliminate v we find that

$$Rv = \frac{X}{Y^{\frac{3}{2}}}\sqrt{\frac{P^3}{\omega}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [17]$$

Hence we must reduce to a minimum the value of

$$\frac{X}{Y^{\frac{3}{2}}} = \frac{1 + \xi\,\alpha^2}{(1 + \eta\alpha)^{\frac{3}{2}}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [18]$$

50 If we take the logarithmic derivative we have an equation which is analogous to Equation [14] but materially different.

$$\alpha^2 + \frac{4\alpha}{\eta} - \frac{3}{\xi} = 0$$

The useful root of this equation is

$$\alpha = \frac{2}{\eta}\left(-1 + \sqrt{1 + \frac{3\,\eta^2}{4\,\xi}}\right) \quad . \; . \; . \; . \; . \; . \; . \quad [19]$$

which, with the data given for the typical aeroplane, brings us to $\alpha = 8.2$ deg., which is very much higher than the optimum value found above.

51 I would here call attention to the fact that in the same aeroplane equipped with a 300-hp. motor running at 1600 r.p.m. and a propeller such as is specified in my paper presented to the Academy of Sciences on July 28, 1919 ($H = 2.279$ m., $b = 2$, $a = 1/30$) we find that

For incidence in ceiling flight.................................. $\alpha = 5.98$ deg.
For maximum velocity of climb in rising from the ground......... $\alpha = 4.28$ deg.
For maximum slope of climb.................................. $\alpha = 7.03$ deg.

PRINCIPAL EQUATIONS FOR AN AEROPLANE IN RECTILINEAR HORIZONTAL FLIGHT

52 Two principal relations govern the rectilinear horizontal flight: one establishes a simple correspondence between the slip of

the propeller and the incidence, this being done through X, and the other indicates the specific weight ω of the air in which the aeroplane can fly at a given incidence.

53 Since the tractive force of the propeller, $F = b\omega n^2\sigma$, balances the resistance of the propeller, $R = X\omega v^2$, we have

$$bn^2\sigma = Xv^2$$

but v/n is the advance $H(1 - \sigma)$ of the propeller per revolution. After making the proper substitution we obtain

$$\frac{1}{\sigma} + \sigma - 2 = \frac{b}{H^2X} \quad \text{[20]}$$

This equation shows that the slip depends only on X, that is to say, on the incidence, and that it increases simultaneously with X.

54 If, instead of $F = b\omega n^2\sigma$, we take the more complete form where σ is multiplied by $1 - e'\sigma$, the equation becomes somewhat more complicated, but still remains of the second degree, namely,

$$\frac{(1 - \sigma^2)}{\sigma(1 - e'\sigma)} = \frac{b}{H^2X} \quad \text{[21]}$$

55 On the other hand, the power delivered by the motor, which is $2\pi nA\ (\omega - \upsilon)$, according to Equation [2], balances the power absorbed by the propeller, which latter is equal to the resistance $X\omega v^2$ of the aeroplane multiplied by its velocity of forward motion $v = nH(1 - \sigma)$ and divided by the efficiency ρ of the propeller. We have, therefore,

$$2\pi A\ (\omega - \upsilon) = \frac{X\omega v^2}{\rho} H(1 - \sigma)$$

Replacing now ωv^2 by P/Y in accordance with Equation [10] the expression becomes

$$\omega - \upsilon = \frac{1 - \sigma}{\rho} \frac{X}{Y} \frac{H}{2\pi A} P$$

But $(1 - \sigma)/\rho$ is the function ψ as defined by Equation [8], hence

$$\omega - \upsilon = \psi \frac{X}{Y} \frac{H}{2\pi A} P \quad \text{[22]}$$

56 This expression, which explicitly or indirectly involves the ten specific coefficients of the aeroplane, the propeller and the motor,

shows very clearly the influence of each of them. It determines ω, and by Equation [1] the altitude Z of flight, at any given incidence. In the second member it is only $\psi \frac{X}{Y}$ that depends on this incidence, the relation being established for X and Y by Equation [12], and for ψ by Equations [8] and [20].

57 Unfortunately, these two latter equations as well as the one for X are of the second degree, so that it is not possible to express ψ explicitly as a direct function of α; but the computation is always possible, at least by successive approximations through the intermediary equations.

58 In order to simplify the expression for Equation [22], we may replace the function $\psi \frac{X}{Y}$ by Φ, thus:

$$\Phi = \psi \frac{X}{Y} = \left(1 - \frac{\sigma}{2} + \frac{a}{\sigma}\right) \frac{X}{Y} \quad . \; . \; . \; . \; . \; . \; . \quad [23]$$

59 The ceiling of an aeroplane is determined by the minimum value of this function Φ and not by the minimum value of X/Y; it is influenced by the variation in the slip of the propeller. For our Aeroplane-Type the incidence at the ceiling is given by $\alpha_p = 5.98$ deg., or roughly 6 deg., corresponding to $X = 0.194$, $\sigma = 0.269$, $\psi = 0.989$ and $\Phi = 0.1163$.

FLIGHT ALTITUDE FOR MINIMUM CONSUMPTION OF FUEL

60 By way of exemplifying the application of the general equations which have just been established, I shall try to determine at first the altitude at which one should fly in order to secure the minimum consumption of fuel and oil per kilometer of distance covered and then the maximum distance which can be reasonably expected to be covered under the present actual conditions of the technique of aeroplane construction.

61 Let Π be the useful power of the engine in kilogram-meters per second at any kind of horizontal rectilinear flight régime. We have then

$$\Pi = \frac{Rv}{\rho} = \frac{X\omega v^3}{\rho} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [24]$$

Dividing Equation [24] by Equation [10], member for member, we get

$$\frac{\Pi}{P} = \frac{Xv}{Y\rho} \quad \ldots\ldots\ldots\ldots \quad [25]$$

from which v can be computed. The distance dL covered by the aeroplane in the time dt is $v\,dt$; hence

$$dL = \rho \frac{Y}{X} \frac{\Pi}{P} dt \quad \ldots\ldots\ldots\ldots \quad [26]$$

62 On the other hand, during time dt the consumption of gasoline and oil which decreases the weight of the aeroplane by dP is

$$dP = -\frac{\Pi E}{75 \cdot 3600} dt \quad \ldots\ldots\ldots\ldots \quad [27]$$

where E represents the fuel and oil consumption per horsepower per hour. Eliminating dt from Equations [26] and [27] and evaluating the distances in kilometers instead of in meters, gives

$$dL = -270 \frac{\rho}{E} \frac{Y}{X} \frac{dP}{P} \quad \ldots\ldots\ldots \quad [28]$$

from which the consumption of fuel and oil per kilometer, which we designate by K, is

$$K = -\frac{dP}{dL} = \frac{EP}{270\rho} \frac{X}{Y} \quad \ldots\ldots\ldots \quad [29]$$

If we replace E by its value from Equation [3] and $X/\rho Y$ (according to Equations [23] and [9]) by $\Phi/(1-\sigma)$, then

$$K = \frac{E_0 P}{270} \frac{\omega_0 - \upsilon}{\omega_0 + o} \frac{\omega + o}{\omega - \upsilon} \frac{\Phi}{1 - \sigma} \quad \ldots\ldots \quad [30]$$

63 In this expression we can eliminate ω and substitute for it the function Φ. If we compare flight at ground level (where Φ becomes Φ_0) to flight where the specific weight of air is ω we have, according to Equation [22],

$$\frac{\omega - \upsilon}{\omega_0 - \upsilon} = \frac{\Phi}{\Phi_0} \quad \ldots\ldots\ldots\ldots \quad [31]$$

Equation [30] therefore takes the simpler form

$$K = \frac{E_0 P}{270} \frac{\omega + o}{\omega_0 + o} \frac{\Phi_0}{1 - \sigma} \quad \ldots\ldots\ldots \quad [32]$$

Replacing ω now by its value in Equation [31] we obtain as a final expression

$$K = \frac{E_0 P}{270} \frac{\Phi + \frac{\upsilon + o}{\omega_0 + o}(\Phi_0 - \Phi)}{1 - \sigma} \quad \ldots\ldots \quad [33]$$

64 This formula shows that the consumption per kilometer at any altitude whatsoever is proportional to the consumption per horsepower-hour of the motor on the ground, to the total weight of the aeroplane, and to a given function of Φ and σ which is basically a function of the incidence α.

65 To show its practical application let us consider the case of the Aeroplane-Type of which the specific coefficients have been given above and let us further assume that $P = 1700$ kg., $E_0 = 0.270$, $\omega_0 = 1.22$ and $\frac{\upsilon + o}{\omega_0 + o} = 0.15$. To each incidence α there correspond X and Y determined from Equation [12], then a slip σ which can be computed from Equation [20] (making $b = 2$ and $H = 2.279$), a ψ which is obtained from Equation [8] (in which a is estimated to be 1/30), and Φ determined from Equation [23]. The specific weights of air ω are computed by means of Equation [22] and the corresponding altitudes Z by Equation [1], with $B = 21.6$. Table 1 gives the results of such computations for values of the incidence α varying by 1 deg.

TABLE 1 AEROPLANE CHARACTERISTICS FOR FLIGHTS AT DIFFERENT ANGLES OF INCIDENCE

α deg.	σ	Φ	$\frac{0.85\,\Phi + 0.15\,\Phi_0}{1 - \sigma}$	K Kg. per km.	ω Kg per cu. m.	Z m.
0	0.200	0.2178	0.2723	0.4628	1.220	0
1	0.202	0.1699	0.2220	0.3774	0.979	2068
2	0.209	0.1438	0.1959	0.3332	0.847	3423
3	0.219	0.1291	0.1825	0.3103	0.773	4281
4	0.233	0.1211	0.1770	0.3008	0.773	4784
5	0.250	0.1173	0.1766	0.3002	0.713	5032
6	0.269	0.1163	0.1772	0.3059	0.708	5106

66 The curve in Fig. 2 shows in a clearer manner than the table the variations of consumption. As abscissæ are plotted distances in meters of flight considered to take place at the ceiling, and

as ordinates the increase of consumption in percentages with respect to the minimum equal to 0.2997 kg. per hp-hr. for $\alpha = 4.6$ deg.

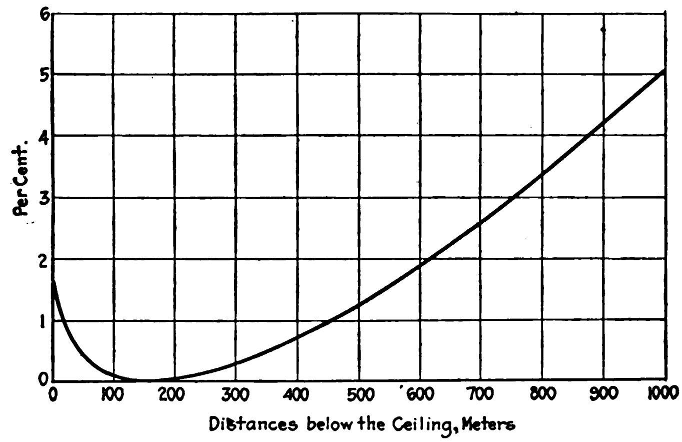


FIG. 2 VARIATION OF K (CONSUMPTION OF OIL PER KILOMETER) WITH ALTITUDE FOR THE AEROPLANE-TYPE

67 Fig. 2 shows that the minimum fuel consumption is attained at an altitude only 151 m. below the ceiling. At the ceiling the consumption is 2 per cent higher than the minimum; at 450 m. below the ceiling 1 per cent; at 650 m. 2 per cent; and at 1000 m. 5.1 per cent. Such are the limits of altitude within which the pilot must keep in order not to exceed 1, 2 or 5 per cent of fuel consumption in excess of the possible minimum.

CRUISING RADIUS

68 Aeroplanes have already been made to traverse in a nonstop flight distances in excess of 3000 km.; for example, across the Atlantic between Newfoundland and Ireland. What cruising radius, then, is obtainable in calm air with the means at present available?

69 While it is apparently possible to go beyond 5000 km. and even 6000 km. it is doubtful if it would be possible to obtain 7000 km. It is evident that one must start with the greatest possible load of fuel and always fly at altitudes where the consumption is a minimum. We have just seen that this altitude is about 150 m. below the ceiling and that the incidence corresponding thereto is about 0.5 deg. below the optimum incidence. Under these condi-

tions the aeroplane loses a good deal of its velocity, and flying is unstable and exhausting for the pilot. It is sufficient, however, to keep at an incidence of 1.5 deg. below the optimum incidence, which will cause the plane to fly at about 400 m. below the ceiling; under these conditions the consumption of fuel and oil is not materially different from the minimum, and it is useless to change the incidence as no appreciable gain is secured thereby.

70 Let us assume now — and this will be our basic condition — that the pilot maintains the incidence constant throughout his entire trip, except, of course, when he has to change it in order to circumvent some obstacle. Since α is determined the same holds good according to Equation [12] for X/Y, which, besides, will have a value very close to the minimum; according to Equation [20] the slip σ is also constant, and hence ψ is constant according to Equation [8] and so is ρ according to Equation [9].

71 All the values in Equation [22] are therefore constant with the exception of ω and P. Taking the logarithmic differential we have

$$\frac{d\omega}{\omega - v} = \frac{dP}{P} \quad \ldots\ldots\ldots\ldots \quad [34]$$

72 This value of dP/P and the value of E in Equation [3] make it possible to express dL given by Equation [28] as a function of the sole variable present ω, so that the integration is quite easy. On making these substitutions we get

$$dL = -270 \frac{\rho Y}{E_0 X} \frac{\omega_0 + o}{\omega_0 - v} \frac{d\omega}{\omega + o} \quad \ldots\ldots \quad [35]$$

and the integration from the starting point gives

$$L = \frac{270}{M} \frac{\rho}{E_0} \frac{Y}{X} \frac{\omega_0 + o}{\omega_0 - v} \log \frac{\omega_0 + o}{\omega + o} \quad \ldots\ldots \quad [36]$$

where M is the modulus 0.4343 of base 10, permitting the use of common logarithms.

73 If the consumption of oil, represented by o, were not constant but proportional to the consumption of gasoline, we would have to incorporate o into v, and o would not appear in the logarithm. As a matter of fact, however, up to altitudes in the neighborhood of 8 km. log (ω_0/ω) is, according to Equation [1], proportional to the altitude. This being the case, the inclination of the trajectory $(= Z/L)$

is *strictly* constant and moreover it is very slight, since, as we will see, it is of the order of 1.4 m. per km. Essentially, the flight is therefore of a rectilinear horizontal character. The presence of o in the logarithmic expression causes L to increase somewhat less rapidly than the altitude. In other words, the inclination of the trajectory of the aeroplane, because of the character of the consumption of oil being more constant than that of the fuel, increases somewhat as the aeroplane advances and rises. But a contrary effect is created by the decrease of coefficient B of Equation [1] above $Z = 6$ km.

74 According to Equation [22], $\omega - v$ remains proportional to P, and we have

$$\frac{\omega - v}{\omega_0 - v} = \frac{P}{P_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [37]$$

75 The specific weight ω of the air in which the plane flies decreases somewhat less rapidly than the total weight P and therefore, according to Equation [10], where Y is constant, the velocity v along the trajectory decreases gradually but to a small extent, the less so, as v is squared in this equation. The same applies to the speed of rotation of the motor as the slip of the propeller does not vary.

76 We may now establish a relation between L and P by substituting for ω in Equation [36] its value taken from Equation [37]. In this way we obtain

$$L = \frac{270}{M}\frac{\rho}{E_0}\frac{Y}{X}\frac{\omega_0 + o}{\omega_0 - v}\log\frac{P_0}{P + \dfrac{v + o}{\omega_0 + o}(P_0 - P)} \quad . \quad . \quad . \quad [38]$$

All that is necessary to obtain the cruising radius L in kilometers then is to give to P its value P_1, which is the net weight at the end of the trip when all the fuel and oil have been consumed.

77 At the end of the trip, however, the plane is at an altitude Z corresponding, according to Equation [1], to the specific weight ω computed from Equation [37] with P having the value P_1. When at that elevation the plane still has the ability to volplane down under its optimum incidence and to cover a small additional distance equal to Y/X times its altitude. The total distance is then expressed by L_m.

78 In an earlier part of the paper it was stated that the relative consumption of oil decreases a small amount simultaneously

with a decrease in the specific weight ω of the air drawn in by the motor. This being so, one could come still closer to precise values by dividing o into two parts: one o', the main part, independent of ω; and the other o'', which is proportional to ω. This latter does not enter into the part of the equation under the symbol of the logarithm, the correction consisting in replacing o by o' in the ratio $\frac{v+o}{\omega+o}$. This correction is small, and results, for example, when o' is equal to $\frac{2}{3}\,o$ in diminishing the ratio by about 13 per cent, while the logarithm is increased only by a quite insignificant amount.

APPLICATION

79 The optimum ratio $\left(\frac{Y}{X}\right)_m$, inverse of the fineness, goes as high as 9 in the case of well-made planes, and one may even discount the possibility of attaining a ratio of 10. Here, however, we are in the region a little below the ceiling and, moreover, the fuel reservoirs are quite large. It is therefore difficult to admit that in actual practice Y/X may materially exceed 8 or, let us say, 8.5.

80 The propeller operating at constant slip will be naturally laid out to be sufficiently large in order to obtain a slip exactly corresponding to its maximum efficiency. Now, this maximum efficiency with good propellers goes as high as 75 per cent, perhaps 78 per cent, or even 80 per cent with geared propellers, let us say $\rho = 0.75$.

81 The fuel and oil consumption E_0 on the ground for good engines, with perfect carburetion, is of the order of 0.270 kg. per hp-hr.

82 Let us assume further $\omega_0 = 1.22$, $v = 0.11$ and $o = 0.085$, which corresponds to a consumption of oil 7 per cent by weight of the fuel in running on the ground, and let us assume that the oil consumption is independent of ω.

83 It is the ratio P_0/P_1 (total weights at starting and arriving) that has most influence on the cruising radius. It is easy to build aeroplanes that will lift a weight of fuel and oil equal to their own weight, including therein the reservoirs, the pilot and all of his requisites. There is no doubt that planes can be built that will lift 1.5 times their own weight; planes that could lift twice their own weight are less certain, and it would be quite difficult to go beyond that.

84 Bearing these various hypotheses in mind, Equation [38] gives the cruising radii L in the second column of Table 2, and Equation [37] the specific weight at the end of the trip. The altitudes reached Z are given in the third column as calculated from Equation [1] in which B equals 21.6 for altitudes below 6 km. and $22.75 + 5.5 \log (\omega/\omega_0)$ for the two other altitudes. The final volplaning increases L by about 8.5 Z, and column 4 gives the total cruising radius L_m. Finally, the fifth column gives the average angle of climb θ in thousandths.

85 We have assumed that $\rho = 0.75$ and $Y/X = 8.5$. If we had taken $\rho = 0.80$, which no doubt could be obtained with a larger propeller driven by a reducing gear, and $Y/X = 9$, a value which is possible of attainment, all the other values in Table 2 could be in-

TABLE 2 CRUISING RADII FOR VARIOUS LIFTING RATIOS OF PLANES

$\frac{P_t}{P_1}$	L kilometers	Z kilometers	L_m kilometers	θ
2	4148	5.692	4196	1.37 per 1000
2.5	5346	7.153	5407	1.34
3	6267	8.314	6338	1.32

creased by about 13 per cent, so that maximum L_m would become 7162 km. instead of 6338.

86 It would appear, therefore, that 7000 km., in round number, is the limit of cruising radius that one can hope to attain with the actual conditions of aeronautical engineering now existing. To exceed this limit it will have to become possible to construct aeroplanes of higher fineness or aeroplanes capable of leaving the ground with a load of fuel and oil more than twice as great as their own load; and, furthermore, without materially increasing the weight of the plane, to provide conditions that will permit the pilots to remain for dozens of hours at altitudes in excess of 5000 meters, perhaps as high as 8000 meters.

DISCUSSION

Edward P. Warner. The analysis which the author gives for conditions of minimum fuel consumption is based upon the assumption of a wide-open throttle at all altitudes. It is possible materially to improve on the figures given, for all altitudes up to the ceiling, by throttling the engine and flying at reduced speed. The work done in traveling a given distance is equal to the product of the distance and the resistance to be overcome. Since the resistance of any particular airplane is a function only of the angle of attack and of the load carried, being quite independent of altitudes, the distance traveled for a given expenditure of useful work is the same for all altitudes and the angle of attack for maximum range is the same for all altitudes and is equal to the angle for which the lift-drag ratio for the machine as a whole is a maximum (usually between 6 and 8 deg.). The condition of absolute maximum range is that in which the consumption per horsepower per hour is a minimum. Economy decreases with increasing altitude, but it also decreases when the engine is throttled to fly at economical speed near sea level. If, however, the airplane has a number of power units, the power can be reduced by shutting down one or more entirely, keeping the remainder running at full load and max mum economy. If this be done, keeping the same values as those suggested by the author, for Y/X and P, and taking P_0/P_1 as 2, the maximum theoretical range can be increased from 4196 km. to 5930 km., flying at sea-level at an angle of 6 deg.

It would be interesting to know something of the developments on which are based the author's statements relative to the weight of fuel which can be carried. A useful load of 40 per cent of the total load is considered exceptionally high, yet the figures given in the paper as easy to realize mean a useful load, including tanks and pilot, of at least 57 per cent of the total load. With such load-carrying capacity a straight-line, non-stop flight from New York to Paris becomes the merest bagatelle.

E. N. Fales. As supplement to the author's paper it will be interesting to examine the actual performance of an American-built airplane recently tested at the Air Service Experimental Station, McCook Field, Dayton, Ohio. The airplane selected for discussion is a Thomas-Morse, MB3, with 300-hp., 8-cylinder Hispano Suiza engine. Like all planes, it loses speed as altitude increases, due to the decrease of engine power in thin atmosphere. The author is one of the pioneers in developing a "supercharger" or compressor designed to furnish air at normal density to the engine carburetor at all altitudes. It will be appropriate to compute the improvement to be expected in the performance of this airplane, assuming a supercharger added of such perfection that the engine continues to deliver its 300 hp. at all altitudes.

Much progress has been made since the war in altitude flying. Major Schroeder recently made at McCook field a world's record of 38,180 ft., aided by a Moss supercharger. The adjustable pitch propeller, which is a necessary complement of a supercharger has also been perfected at McCook field by Messrs. Hart and Caldwell. Therefore the attainment of comparatively constant propulsive power in the near future is not impossible.

Our computations will show speeds of 200 to 300 m.p.h. Lest such value seem too amazing to the mechanical engineer, it is well to note that airplane speed is governed by factors which do not enter into earth-borne speeds; the variation of air density and of angle of attack in mechanical flight produce effects which have few parallels in other forms of locomotion. It is therefore fair to expect airplane speed records to augment faster than earth-borne speed records; depending largely upon the development of a propulsive power which does not, like the gasoline engine, decrease in the lighter air of high altitudes.

It must be remembered, however, that the benefits of high-altitude transportation are not available for short flights. For example, should we wish to utilize the maximum cruising radius, according to the author's analysis in Fig. 2, we must approach the ceiling. In practice, an airplane without supercharger requires a full hour to reach the ceiling, and it is apparent that a trip of an hour or two derives less advantage from height than does a 10-hour trip.

While future cruising speeds of 300 m.p.h. are shown below to be possible, it does not necessarily follow that we can get from St. Louis to Indianapolis, 240 miles, in 50 min.

When the mechanism of altitude flight shall have been perfected, the benefits will be most apparent for trips exceeding 500 miles. Utilization of differential velocities in winds at various heights, and more direct course, will aid in securing speeds fourfold greater than railway speeds.

The calculations below show that New York may be reached from St. Louis in 4 hours (see Fig. 3). This example is fairly typical of what the future holds forth, since no excessive fuel capacity is demanded for such a trip. However, the great altitude affords comparative safety in case of engine failure, — the plane always having a reserve radius of 50 miles without power, due to its height.

The calculations are according to the approximate method of

Lt.-Col. V. E. Clark;[1] the data are from actual flight tests made February 20, 1920, at McCook Field. Fig. 3 shows the improvement in speed and height to be expected on the airplane by maintaining its propulsive power constant, that is, by supercharging the engine and using an adjustable pitch propeller. It must be assumed that engine cooling is properly maintained and the pilot properly supplied with oxygen and heat.

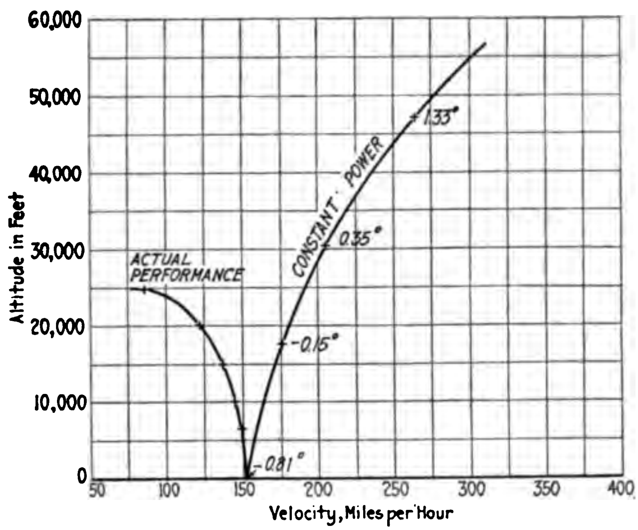


Fig. 3 SPEEDS WITH AND WITHOUT SUPERCHANGES

CALCULATIONS

By the Clark method, the velocity necessary for support of the airplane is:

$$V = C_1\sqrt{\frac{S}{\delta}} \quad \ldots\ldots\ldots\ldots \quad [1]$$

V = vel. in m.p.h.
S = pounds weight supported per sq. ft. wing surface
δ = relative density air
C_1 = Constant depending on angle of incidence of wings

$$C_1 = \frac{1}{\sqrt{Ky}}$$

[1] See Technical Orders No. 10, Air Service, Dayton, O.

The power required is

$$P_h = \frac{C_3 \times \delta \times WV^3}{Se} \quad \ldots\ldots\ldots\ldots \quad [2]$$

P_h = Hp.
C_3 = 375 × coef. of res. of plane
W = weight in pounds
V = m.p.h.
e = propeller eff.

Therefore velocity attainable for a given power, P_h

$$V = \sqrt[3]{\frac{P_h S e}{C_3 W \delta}} \quad \ldots\ldots\ldots\ldots \quad [3]$$

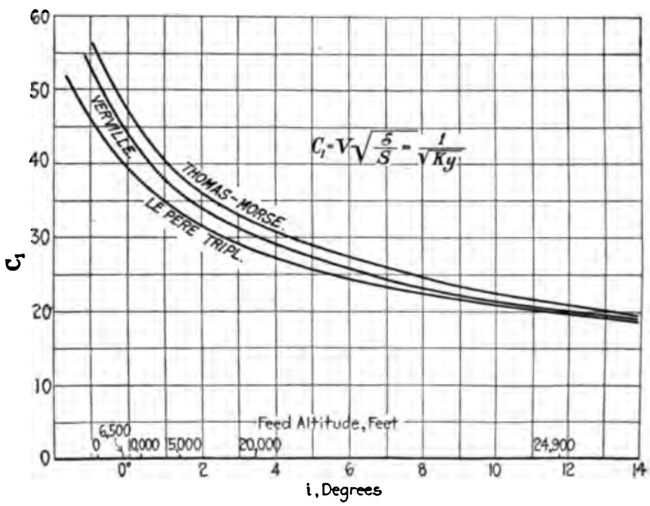


Fig. 4 VALUES OF C_1

The altitude attainable is given in terms of relative density

$$\delta = C_4 S\left(\frac{W}{P_h e}\right)^2 \quad \ldots\ldots\ldots\ldots \quad [4]$$

where $\qquad C_4 = C_3^2 C_1^6$

Assume that

$$P_h = P_o \times \delta \left(\frac{1}{\delta^{.78}}\right)$$

P_h = Hp. at altitude of relative density δ
P_o = Hp. at sea level.

Then by supercharging the plane we increase the altitude according to the change in density:

$$\frac{\delta}{\delta\left(1+\frac{2}{\delta^{.78}}\right)}$$

In Table 1 is given the calculation of the coefficients. C_1 (see Fig. 4) is deduced from wind-tunnel tests, being a factor dependent on the lift coefficient of the wings.

(C_2) (C_3) (C_4) (See Fig. 5) are from flight test at McCook Field.

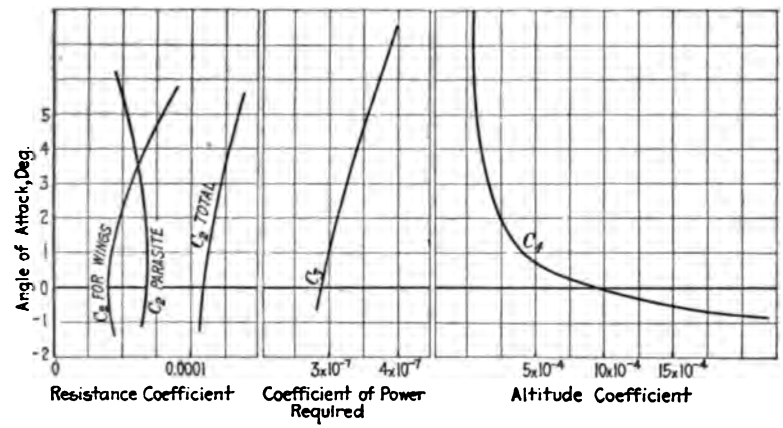


FIG. 5 VALUES OF C_2, C_3 AND C_4

TABLE 1 PERFORMANCE COEFFICIENTS FOR THOMAS MORSE PURSUIT PLANE M-B-3 WITH 300-HP. HISPANO SUIZA ENGINE

(See Performance Test Report No. 2/20/20)

Alt. (miles)	δ	$\delta\left(1+\frac{2}{\delta^{.78}}\right)$	$\delta\left(\frac{1}{\delta^{.78}}\right)$	Vel., m.p.h.	R.p.m.	$\frac{V}{ND}$	Prop. Eff. e	C_1	$C_2 = 375C_3$	C_3	C_4	Angle of attack, degrees (i)
0	1.0	1.0	1.0	152	1835	0.90	0.81	54.6	$1.04(10^{-4})$	$2.77(10^{-7})$	$20.1\ (10^{-4})$	−0.81
6,500	0.825	0.535	0.800	148	1820	0.88	0.81	48.3	$1.09(10^{-4})$	$2.91(10^{-7})$	$10.7\ (10^{-4})$	−0.15
10,000	0.740	0.350	0.682	144	1790	0.87	0.80	44.5	$1.12(10^{-4})$	$3.00(10^{-7})$	$6.88(10^{-4})$	+0.35
15,000	0.628	0.164	0.515	136	1730	0.85	0.79	38.7	$1.15(10^{-4})$	$3.08(10^{-7})$	$3.14(10^{-4})$	1.33
20,000	0.530	0.068	0.350	122	1640	0.80	0.77	32.0	$1.25(10^{-4})$	$3.35(10^{-7})$	$1.24(10^{-4})$	3.5

S = Weight per sq. ft. = 7.74 lb.
Weight per hp. = 6.48 lb.

Weight = 1945 lb.
Area = 250.5 sq. ft.

In Table 2 is given the calculation of the probable speed and altitude if the airplane has constant power.

Fig. 4 is a chart of C_1 plotted against angle of attack (or altitude).

Fig. 5 gives C_2 (Resistance coefficient)

C_3 (Coefficient of power required)

C_4 (Coefficient for altitude)

TABLE 2 PERFORMANCE OF THOMAS MORSE M-B-3 ASSUMING ENGINE POWER CONSTANT AT ALL ALTITUDES. DAYTON, OHIO 5/21/20

Angle of attack, deg. (i)	C_1	C_4	$\delta = 533\ C_4$	Altitude ft.	V
−0.81	54.6	20.1 (10^{-4})	1.00	0	152
−0.15	48.3	10.72 (10^{-4})	0.572	17,800	178
+0.35	44.5	6.88 (10^{-4})	0.367	30,700	205
1.33	38.7	3.14 (10^{-4})	0.167	47,000	264
3.5	32.0	1.24 (10^{-4})	0.066	63,000	347

$$\delta = C_4 S \left(\frac{W}{P_o e}\right)^2 = 533 C_4$$

$$V = C_1 \sqrt{\frac{S}{\delta}} = \frac{2.79\ C_1}{\sqrt{\delta}}$$

$e = 0.78$ (Assumed constant)

J. G. Coffin [1] wrote that the author went into greater detail than could be applied in practice, and referred to his report [2] on the transatlantic flight of the NC's in which he covered the question of airplane ranges. The aim in his report has been to introduce the least number of complications possible in the matter and derive a practical method for giving an estimate of possible ranges. Since 1918, not only the French but the Italian and English have made reports on this subject, all coming to substantially the same conclusions. The English report in particular is very elaborate. The author's contribution is that of going more deeply into the detail of the matter from a theoretical standpoint. Such work, of course, is always of value.

Walter C. Durfee. The most that the writer feels able to contribute to the discussion is to point out that the calculation of

[1] Director of Research, Curtiss Aeroplane & Motor Corporation, Garden City, Long Island, N. Y.

[2] Report No. 69 of the National Advisory Committee for Aeronautics.

maximum cruising radius as presented by the author seems to depend, among other things, on the conservative assumption that it is desirable to operate all of the engines, with all of their cylinders, all of the time, until the fuel is exhausted.

Although the high altitudes which the author mentions may have some advantages in connection with speed, it seems very probable that a somewhat greater cruising radius could be attained at lower levels of the atmosphere by a suitable manipulation of engines, which, if not now considered desirable, might be arranged in the future.

The author states that a good airplane can be flown with a most economical tractive force, which amounts to about 1/9 of the total weight, at every level of the atmosphere and at every loading of the machine. When this tractive force is used, the efficiency of the propeller remains constant, with a constant slip and a constant number of revolutions per mile. As the airplane becomes lighter by expenditure of fuel, this tractive force, which is 1/9 of the total weight, becomes less. Consequently, if the carburetion of the fuel continues to be perfect, the engine will furnish an excessive tractive force which will drive the airplane at a wasteful or uneconomical speed. In order to reduce the tractive force in proportion to the lessened weight of the airplane, the engine is allowed to operate at higher levels of the atmosphere where it will be supplied with less amounts of air and fuel. The result of this readjustment of height throughout the flight is that the airplane rises at a grade of about seven feet to the mile, until at the end of three or four thousand miles it attains a great height where special provision must be made for the protection of the passengers.

Since the sole object of this rise to high elevations, so far as the cruising radius is concerned, is to cut down the tractive force while maintaining a perfect carburetion of fuel, it would seem that the same thing could be arranged by reducing the tractive force in some other manner, as by cutting out some of the engines in case there are several suitably arranged for this purpose. Another plan involving, however, questions of propeller design, would be to fly the plane constantly at its most economical speed and allow the surplus tractive force of the engine to be stored as potential energy of elevation which could be used up from time to time by shutting off the engines and allowing the machine to glide. Under these circumstances no energy would be lost on account of the friction and the sustention of the airplane, except for the economical amount previously indicated. The cruising radius would be somewhat greater as compared with the high altitude radius. This is because the flight might be accomplished at a low level where the internal losses of the motor would be a smaller proportion of the total expenditure of energy, as is indicated by Eq. 2. The first plan outlined above simply amounts to making the engine constant A in Eq. 22 a variable so as to alter the calculation for Z in Par. 65. The second plan somewhat resembles a series of flights of the kind outlined by the author, but at lower altitude.

The author's estimate of maximum cruising radius can be said to be conservative. Some greater distance, corresponding to improved engine efficiency, might be covered by traveling closer to the earth's surface in a suitable manner.

MORGAN BROOKS. The author adds the amount of one percent to the cruising radius by planing down after all the gasoline has given out, claiming that if the machine is 8 km. high he can plane down 40 km. in the maximum flight he gives. He goes not allow for the extra gasoline required to reach that great height. The statement is made that the rise is made so gradually that it is equivalent to a horizontal flight. Nevertheless, the rise is there, and it must take more gasoline to rise 8 km. than to fly on the level.

No. 1741

AERONAUTIC INSTRUMENTS

GENERAL PRINCIPLES OF CONSTRUCTION, TESTING, AND USE

By Mayo D. Hersey, Washington, D. C.
Member of the Society

This paper, which is a contribution from the Aeronautic Instruments Section of the National Bureau of Standards, affords a brief sketch of the general principles of such instruments. The discussion takes up successively matters relating to the construction, the testing, and the practical use of instruments for aircraft. In each case principles rather than details are emphasized. In a few instances particular instruments are described or the results of particular investigations stated, in order to illustrate the principle involved. The paper, however, does not pretend to serve as a final or comprehensive report on the subject.

That the general field of aeronautic instruments is a less familiar one to the engineer at large than it logically ought to be, is probably due to the confidential character of the work which has gone forward during the war. Now that attention is again directed to civilian aeronautics the subject of instruments takes on great importance for two chief reasons: first, because the efficiency and safety of air navigation, especially in clouds, depends on the reliability of the instruments which are used; and second, because the whole future development of aeronautic engineering rests upon the measurement of aircraft performance, which requires suitable instruments of precision.

THE life-cycle of any aeronautic instrument starts with the recognition of some distinct physical measurement that has to be made in flight; then, broadly speaking, it passes through the three main stages of construction, testing, and use; and ends by revealing the need for some particular improvement or new development, at which point the cycle begins over again.

2 As a convenient way of arranging the available information, we will consider in succession these different stages or processes, emphasizing only the general principles involved, but illustrating the latter by their application to particular instruments such as the aneroid barometer or tachometer.

Presented at the Spring Meeting, St. Louis, Mo., May, 1920, of The American Society of Mechanical Engineers.

1 CONSTRUCTION OF INSTRUMENTS

3 In classifying instruments with regard to the purpose for which they are needed a fundamental distinction may be drawn between service instruments and experimental instruments. The former class, seen in Figs. 1, 2, and 3, includes only the usual instruments forming a permanent part of the equipment; the latter class includes special instruments, usually with self-recording attachments, for use in experiments on aircraft and especially for conducting performance tests on airplanes. Service instruments have to be light and rugged, fool-proof, and reliable under all conditions of flight, and must not occupy too much space in the cockpit. Experimental instruments must be extremely accurate and the self-recording feature is desirable, but there is great latitude in attaining these ends.

4 A further distinction may be drawn between instruments for balloon and airship use, and those for airplanes. Historically, instruments for lighter-than-air craft were the first to be developed; witness for example the origin of the aneroid barometer for aeronautic use in France in 1798. But owing to the more extensive use of airplanes during the war, the scientific elaboration of airplane instruments stands now far in advance of airship instruments. The chief characteristics of airship instruments are the relatively lower range of speed and altitude over which the mechanism has to function accurately, and the correspondingly weaker forces available to operate the movement. In addition, there are some extra instruments needed on airships, for measuring conditions relating to the air and hydrogen supply.

5 Finally, all aircraft instruments, whether for service or experimental use, and whether for lighter-than-air or heavier-than-air craft, may be classified with respect to the nature of the measurement made. From this viewpoint the most important groups are, first, altitude instruments; second, speed indicators; third, direction indicators; fourth, tachometers; fifth, gasoline indicators; sixth, thermometers and pressure gages for the power plant; seventh, time-pieces; eighth, oxygen apparatus; ninth, additional instruments for lighter-than-air craft.

ALTITUDE INSTRUMENTS

6 Taking up these nine groups in order, altitude instruments broadly considered may be regarded as covering the following species:

a Altimeters
b Barographs
c Thermographs and aerographs
d Statoscopes
e Rate-of-climb indicators
f Night altitude indicators.

7 The *altimeter* is in principle a common aneroid barometer with a moveable altitude scale instead of a fixed-pressure scale. For airplane use, however, the aneroid has to be made uncommonly well; otherwise the instrument will have serious errors caused by the peculiar conditions of flight, and the pointer may vibrate so much that it can hardly be read. The writer had his first opportunity to go up in the air and to observe the performance of aneroids under flight conditions at the Annapolis meeting of this Society in 1913. A barograph and a 3-in. surveying aneroid were taken along for that purpose. Altimeters made their first appearance during the subsequent year and the other airplane instruments rapidly followed.

8 A typical altimeter is seen in the upper left-hand part of the instrument board picture, Fig. 1. The elements of its construction are probably familiar, comprising a stiff steel spring coupled to a thin corrugated metal vacuum box, together with a delicate transmission mechanism which multiplies the movement of the box. A dilation or contraction of the box only 0.002 or 0.003 in. causes a motion of 1 in. at the tip of the pointer. Various minor adjustments are provided, and usually a bimetallic bar forms part of the lever system, serving in some measure to compensate for the influence of temperature.

9 *Barographs* usually contain a tier of vacuum boxes, one on top of another, from two to five in number, with a pair of steel springs inside each box. The multiplying factor of the transmission mechanism is decidedly less than in altimeters, so that the pen arm has correspondingly greater power for tracing a firm ink line on the revolving chart.

10 *Thermographs* are intended for recording the temperature of the atmosphere at different altitudes. They have been built

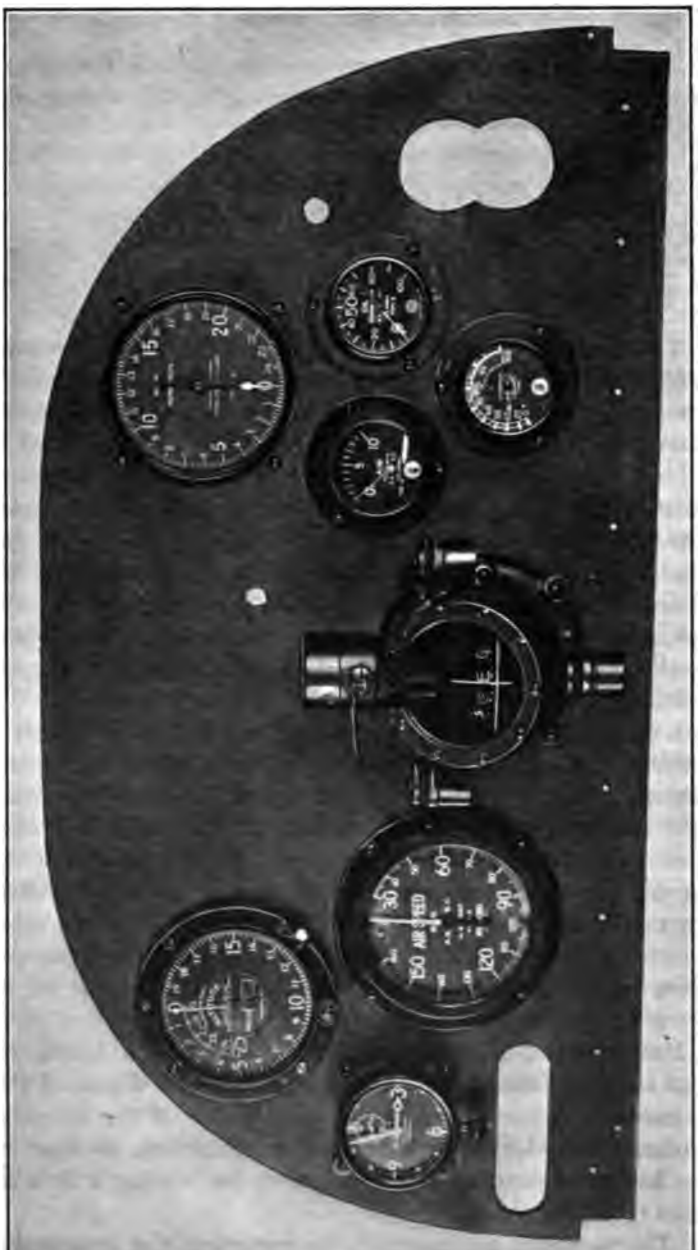


FIG. 1 TYPICAL AIRPLANE INSTRUMENT BOARD, SHOWING ALTIMETER, TIMEPIECE, AIR-SPEED INDICATOR, COMPASS, TACHOMETER, PRESSURE GAGES, AND RADIATOR THERMOMETER

both with bimetallic elements and with delicate Bourdon tubes containing mercury or ether. For performance testing of airplanes direct-reading strut thermometers are more commonly used. *Aerographs* are combination recording instruments comprising hair hygrometer for humidity determination along with the barometric and thermometric elements.

11 *Statoscopes* serve to tell whether a balloon or airship is slightly ascending or descending. They are remarkably sensitive, too much so to be of much use on airplanes. The statoscope in its simplest form is a thermally-insulated vessel of air, connected to the external atmosphere by a glass tube containing a drop of red liquid which moves freely one way or the other in response to any minute change of external pressure. In another form, the drop of indicating liquid is replaced by a sensitive metallic diaphragm and lever system.

12 *Rate-of-climb indicators* give a direct reading of the vertical speed in feet per minute. The most usual types are based upon the pressure difference set up between the two ends of a capillary tube connecting a thermally-insulated air container with the external atmosphere. What is actually observed is the manometer indication of that differential pressure. At first liquid manometers were used, but later a purely mechanical type involving sensitive diaphragms was developed by the Bureau of Standards.

13 The *night altitude indicator* is an optical projection instrument of the range-finder type, particularly valuable for seaplanes flying low over the water.

SPEED INDICATORS

14 Turning to the second general group, speed indicators may be subdivided somewhat as follows:

a Air-speed indicators, pressure type
b Pressure heads for airspeed indicators
c Air-speed indicators, anemometer type
d Ground-speed indicators.

15 The *pressure type of air-speed indicator* shows the impact pressure of the air due to the motion of the aircraft and so enables the pilot to keep above his stalling speed and below the danger point due to excessive speed. It does not show the true speed relative to the atmosphere, except near sea-level unless a subsequent

calculation is made to allow for the change in the air density with altitude. Such instruments consist of two parts: the indicator proper which is mounted on the instrument board in the cockpit (See Fig. 1), and a nozzle or pressure-head mounted out on some strut where it will catch the full force of the undisturbed atmosphere going by. The indicator part measures the differential pressure generated by the pressure head and transmitted to it through some length of tubing. Liquid manometers were used for the indicator early in the war, but later were superseded by various diaphragm constructions. Not only corrugated metal diaphragms, but even rubber and doped silk have been found satisfactory. The pressure difference acting on the diaphragms of an air-speed indicator is far less than in the aneroid barometer, but somewhat greater than in the rate-of-climb indicator. As a general principle it is of course easier to develop a satisfactory mechanism when there is a large force available to actuate it, than when the force is small, on account of the difficulties of eliminating friction.

16 The pitot tube and venturi tube have been tried out in a great variety of forms as *pressure heads for air-speed indicators.* A combination of the two gives the greatest differential pressure and was adopted in this country in preference to the pitot tube, in order to accelerate production of indicator mechanisms during the war. Neither this combination nor the venturi by itself claims any scientific superiority over the familiar pitot tube. Figure 2 shows the particular style of pitot-venturi adopted as standard by the Army. It generates a differential pressure of about 31.5 in. of water at 100 miles per hour near sea level, which in numerical magnitude is about 6.4 times the pressure given by a pitot tube.

17 *Air-speed indicators, anemometer type* (or true air-speed type), are usually based on some rotating vane or Robinson cup principle and connected, either directly or through electric transmission, with a tachometer movement. They are more useful than the pressure type for navigation, but even so require a previous knowledge or conjecture as to the wind velocity in order to determine the ground speed.

18 *Absolute, or ground-speed indicators* have been successfully made in the visual type which gives its result by combining an observation of the angular velocity of the line of sight with a knowledge of the altitude. Some form of instrument is needed, however, which does not depend on the visibility of the ground.

DIRECTION INDICATORS

19 The third general group, direction indicators, may be subdivided as follows:

a Compasses and turn indicators
b Inclinometers and banking indicators
c Angle-of-attack indicators

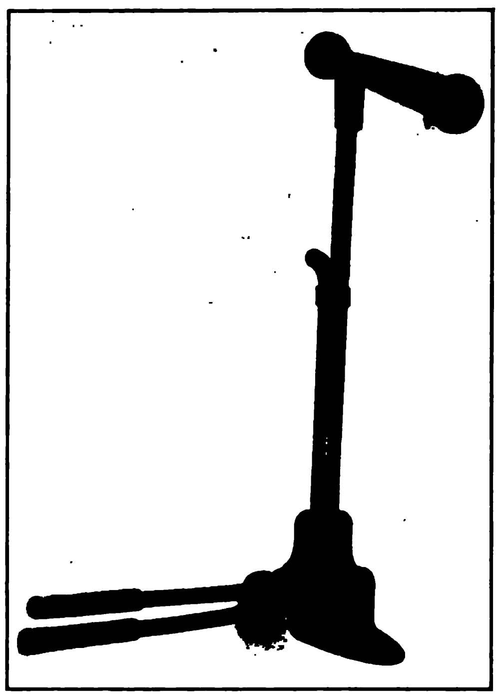


FIG. 2 PITOT-VENTURI PRESSURE-HEAD FOR AIR-SPEED INDICATOR

d Side-slip or yaw indicators
e Drift indicators
f Course-finding instruments.

20 *Compasses* may conceivably be either magnetic or gyroscopic, but the latter have not yet been successfully developed for airplanes. Magnetic compasses in common use are of the short-period type. A long-period type was favored by many practical pilots because of its virtues as a turn indicator, but is no longer

produced. Both liquid-damped and air-damped compasses exist, and of both the flat card and vertical card constructions. ***Turn indicators*** have been made in both aerodynamic and gyroscopic forms. The former type measures the differential pressure between two static heads placed one at each wing tip. It is therefore a purely atmospheric instrument which might read zero if the aircraft were resting in a large portion of the atmosphere itself undergoing rotation. The gyroscopic forms show absolute rotation: one type is a very sensitive indicator showing whether the aircraft is or is not turning, and whether to the right or left, while the other type is somewhat less sensitive but actually measures the rate of turning.

21 *Inclinometers* are to be distinguished from banking indicators in that they measure the inclination of the aircraft to the ground, while *banking indicators* show only the departure of the aircraft from the proper banking position. Except for unaccelerated flight, inclinometers can only be correct if they operate on some gyroscopic principle. Gyroscopic inclinometers, whether of the simple spinning-top variety or some other, are usually of the long-period-pendulum type. They function just as a simple pendulum might if it were several thousand feet long, and so are free from the effects of transient accelerations, but not from persistent acceleration in the same direction, as with continuous flight in a circle. The most common inclinometers are liquid-filled tubes in the form of a large, triangular closed circuit, and are used for fore and aft observations. When the aircraft climbs upward, the liquid seeks its level, and the liquid rises in that part of the tube which is nearest the pilot, so that the movement of the meniscus can be read on a scale. The commonest form of banking indicator is a modification of the familiar spirit-level, bent into an arc of about 20 deg. each side of the zero. Mechanical pendulum types also are in use.

22 *Angle-of-attack indicators* show the angle between the wing chord (say) and the direction of motion. Both weathervane and differential-pressure types have been used, the latter like the side-slip indicator below.

23 *Side-slip or yaw indicators* show whether the aircraft is heading exactly in the direction of motion (relative to the atmosphere) or not. If the banking indicator does not show zero it can be inferred that side-slipping or skidding must exist, but this instrument is direct acting. It operates by the differential pressure

set up between symmetrically placed openings on the two sides of a sphere. This sphere is attached to some strut so as to head directly forward. Evidently if the force of the air comes sideways, due to yaw, more pressure will be set up in one orifice than in the other. The two openings are connected to the two sides of a diaphragm movement with dial and pointer.

24 *Drift indicators* show the speed of drift compared to the speed of advance and involve the rotation of some reference line by the observer until it is parallel to the apparent trajectory of objects going by on the ground.

25 *Course-finding instruments* as made at present are not direct-acting physical instruments but merely mechanical devices for determining the course and distance made good, by reference to data already available. Aircraft sextants are the same as marine sextants in principle but require an artificial horizon.

TACHOMETERS

26 Tachometers constitute the fourth general group. All the various types serve for the sole purpose of indicating the revolutions per unit time of the propeller shaft, with the exception of one instrument which indicates also the angular acceleration of the propeller shaft, that is, its rate of speeding up or of slowing down. The diverse types of construction may be subdivided as follows:

a Chronometric
b Centrifugal
c Magnetic
d Electric
e Air-viscosity and air-pump
f Liquid.

27 The *chronometric* or clock-work type, which is the most accurate but also the most complicated, actually counts the number of revolutions that take place in a fixed time interval.

28 The *centrifugal* type, now the most commonly used on aircraft, works on the principle of the change in position of a revolving mass due to centrifugal force acting against the restoring force of a spring.

29 The *magnetic* type operates by the rotation of a permanent magnet near an electrically conducting disk which is mounted on the pointer spindle and controlled by a spring. By virtue of induced currents, the magnet exerts a force tending to drag the disk around,

thus deflecting the spring by an amount depending on the speed of rotation of the magnet.

30 The *electric* type consists of a magneto, geared to the engine shaft, which transmits the resulting voltage to a millivolt meter in the cockpit which is graduated to read directly in revolutions per minute.

31 The *air-viscosity* type is similar to a torsion viscosimeter. Any such viscosimeter measures the product of viscosity by speed of rotation. Here the viscosity of the fluid, air, is so nearly a constant that the deflection of the spring and pointer is almost directly proportional to the speed of rotation. The *air-pump* type is based on the force of impact of the air against a movable vane resisted by a spring; the pump is geared to the engine shaft so that the deflection depends on the speed of rotation.

32 The *liquid type* involves the observation of the maximum height of a liquid column or deflection of Bourdon gage which can be maintained by a centrifugal pump geared to the engine shaft. By comparing the height of this column with that of a similar one heavily damped an indication of angular acceleration is also obtained. Several of the above physically different types of tachometers are each found in a variety of mechanical modifications.

GASOLINE INDICATORS

33 Gasoline indicators, the fifth group, comprise broadly speaking depth gages and flow meters. *Depth gages* show the available gasoline supply or volume and are of two types, those based on a simple float principle and those which measure the hydrostatic pressure near the bottom of the tank. The latter form permits the indicator part to be mounted in the cockpit. *Flow meters* show the rate of consumption of gasoline at any instant. They have been based on the venturi principle and also on the principle of utilizing the force of impact of the stream of liquid against a movable element resisted by a spring or by gravity.

THERMOMETERS AND PRESSURE GAGES

34 The sixth group comprises thermometers and pressure gages for the radiator, the lubrication system, and the gasoline tank, respectively. These thermometers are of the long-distance transmission type with liquid-filled or vapor-filled bulbs. The

indicator movement, installed in the cockpit, consists essentially of a Bourdon tube element, as do also the air and oil pressure gages.

TIME PIECES

35 The seventh group, time pieces, presents no great novelty aside from the reversing stop-watches used in connection with certain bomb sights; but they have to be rugged enough to withstand airplane service and provided with suitable dials.

FIG. 3 AVIATOR'S OXYGEN REGULATOR

OXYGEN APPARATUS

36 Oxygen apparatus for the aviator, group eight, may be either of the compressed oxygen, liquid, or chemical type. Only the first has been put into regular use in this country. These outfits are provided with automatic regulators which control the flow of oxygen from the storage tank to the aviator's breathing attachment. The regulators are designed to deliver an amount of oxygen which increases at the higher altitudes in accordance with a prescribed formula. The mechanism is operated by diaphragms. An example of such a regulator is shown in Fig. 3.

BALLOON AND AIRSHIP EQUIPMENT

37 The ninth and last group comprises balloon and airship equipment not already included among the foregoing. Special pressure gages are made for indicating the amount of water ballast carried. Delicate manometers are made for measuring the air pressure in the balloonetts and the hydrogen pressure in the envelope; liquid types have been used, but sensitive diaphragms are better. Electric resistance thermometers are used to indicate the temperature of the gas in the envelope. Hydrogen detectors based on the action of osmotic pressure are valuable for locating leaks.

GOVERNMENT INSTRUMENT COLLECTION

38 Examples of practically all the foregoing types and some others are to be found in the government collection of aeronautic instruments, which is maintained by the Bureau of Standards at Washington. This collection is now one of the most complete in the world. It contains instruments of French, British, Italian, Russian, Swiss, Danish, Dutch, German, and Austrian manufacture as well as American.

DESIGN, DEVELOPMENT, AND PRODUCTION

39 The design of aeronautic instruments is at present almost wholly on a cut-and-try basis. The Bureau of Standards has made some progress, however, in placing instrument design on a rational basis. Consideration has been given to such general problems as: the stiffness of elastic systems where two bodies such as a spring and diaphragm are coupled together; the effects of temperature and elastic lag on coupled systems; bimetallic bars for temperature compensation; the balance of moving parts to resist angular acceleration and vibration; the securing of a uniform scale by suitable design of transmission mechanism; and the treatment of aerodynamic problems and the general action of damping fluids by the method of dimensions described before this Society by Dr. Buckingham at the Buffalo meeting in 1915 in his paper on Model Experiments. There is hardly space here even to outline those results, which it is hoped may form the substance of later communications.

40 Experimental development has to follow the preliminary design of an instrument before it can be put into quantity production. It is here that the physical laboratory, in immediate con-

junction with an instrument shop suitably equipped with watchmaker's tools in addition to the ordinary machine tools, plays an essential part. Among the recent development projects of the Aeronautic Instruments Section at the Bureau of Standards may be mentioned the following, several of which were undertaken at the request of the Air Service or the National Advisory Committee for Aeronautics:

a The conversion of barographs from a low-altitude to a high-altitude range by the addition of a suitable external spring, together with the development of recording processes to eliminate the use of ink.

b The conversion of altimeters from an altitude scale to a pressure scale by the modification of the mechanism and engraving of special dials, so as to adapt them for aircraft performance testing. In connection with this work improvements were also made in the method of mounting the main spring of commercial altimeters so as to diminish the observed differences between readings with increasing and decreasing altitudes, or hysteresis as it is commonly called.

c The development of a precision altimeter with a large dial giving a ¼-in. movement of the pointer for each 100 ft. of altitude. In order to justify such high precision of reading, that is, such high sensitivity or openness of scale, a special design had to be made for the elastic system to diminish elastic lag. This was accomplished by making the steel main spring so many times stiffer than the diaphragm that the resultant action of the instrument is practically independent of the quality of the material in the diaphragm.

d The development of a direct-reading, rate-of-climb indicator based on the capillary-tube-leak principle, but in which the use of a liquid has been entirely dispensed with through the use of sensitive diaphragms.

e Development of a reduced-scale working model of a proposed gyro-stabilizer on the long-period pendulum principle, possessing mechanical features of extreme simplicity.

f The application of the moving picture camera, by means of an automatic timing mechanism, for the purpose of securing a record of instrument readings during the performance test of an aircraft in flight. The whole

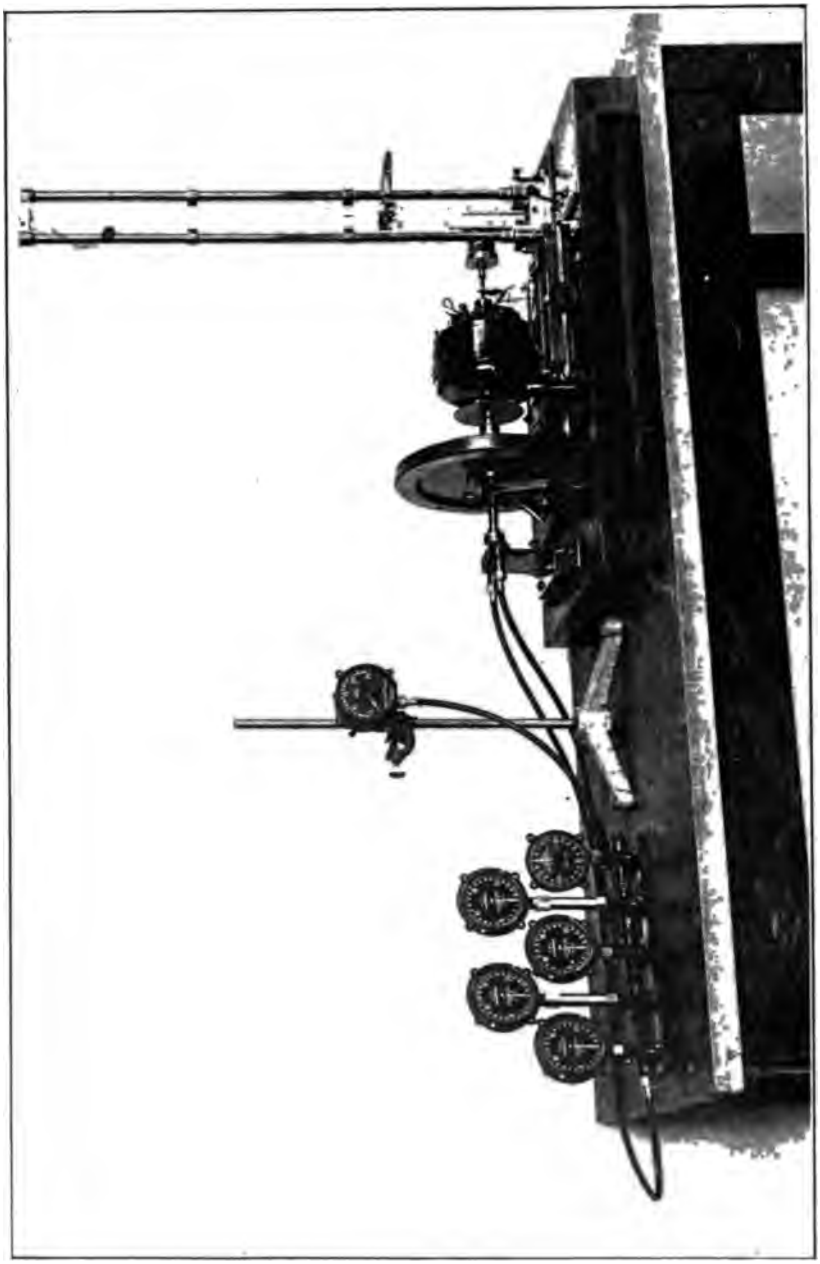


Fig. 4 Testing Outfit for Airplane Tachometers

outfit, instruments, illumination, camera and timing-system, is self-contained and forms a unit which takes the place of the human observer in the airplane.

g A working model of a type of dynamical ground speed indicator which would become available if there existed a suitable gyro-stabilizer to hold it horizontal. This model consists of a large steel ball free to roll back and forth in a glass tube filled with a viscous liquid. It can be shown mathematically that if the fluid resistance is directly proportional to the speed of the ball through the tube, while the tube is held in a horizontal fore-and-aft position, then the displacement of the ball at any instant from its initial position in the tube is directly proportional to the absolute ground speed of the aircraft.

41 The development work on any given model of an instrument must be dropped and the design rigidly fixed before quantity production can be efficiently carried on. In this country the quality of products made in quantity has to be artificially held up to standard by means of specifications and inspection. This situation is not so commonly met with abroad, especially in France, where every individual involved, from the mechanic in the shop up to the head of the concern, takes a professional artistic pride in the quality of his output. This difference is not the fault of the American manufacturer, and there are some advantages in our system, yet the fact remains that in Europe, instrument production has been successfully carried on without specifications. This spectacle of successful instrument production without specifications formed one of the most astonishing impressions which the writer gathered from his trip to Europe during the war.

2 TESTING OF INSTRUMENTS

42 The testing of any aeronautic instrument consists in general of a direct comparison with some suitable standard, the instrument being operated from point to point over its full scale: the whole procedure should then be repeated under different temperatures or other important conditions varying in flight. The most important of these conditions are:

a Extreme low temperatures
b Change in pressure or density of air
c Acceleration and inclination
d Vibration
e Time elapsed during the flight.

43 A typical installation of testing apparatus, in this instance for the testing of tachometers, is shown in Fig. 4. The results of such a test are ordinarily given out in the form of tables or curves showing the correction of the instrument from point to point of its scale. By the correction is meant the amount that has to be added algebraically to the reading of the instrument in order to give the true value of the quantity measured. Typical correction plots or curves of this sort are seen in the first two charts shown on Fig. 5.

44 There are various degrees of refinement possible in the testing procedure, and instrument tests may be classified as follows:

a Factory inspection tests
b Short test for service instruments
c The Bureau of Standards general test
d More thorough tests on sample instruments
e Special tests on experimental instruments
f Reproduction of the flight-history.

45 *Factory inspection tests* can be reduced to a comparatively simple routine after the first few instruments of each pattern have been inspected. In some cases it is permissible to determine simply the corrections at one or two points on the scale, and to omit some parts of the test entirely, except on a prescribed number of samples selected at random from each lot. The purpose of such tests is to control the quality of the output so as to maintain the standard of performance agreed upon in the specifications.

46 The *short test for service instruments* is an attempt to secure, in a minimum time, such data regarding the maximum calibration correction or temperature effect as may be needed for actual reference in connection with the installation or use of instruments on aircraft.

47 The *Bureau of Standards general test* is the one which is furnished on instruments submitted to the Bureau without special instructions. It is a comprehensive standard form of test affording a complete calibration curve together with, at least, numerical values for the more common errors due to temperature and other well known conditions. The data from this general test are sufficient for calculating the probable performance of the instrument under any given set of flight conditions. However, when those conditions are actually known, more exact results can be obtained by carrying out the flight-history tests described under (*f*) below.

48 The Bureau makes *more thorough tests on sample instruments* of new type whose mechanism is not familiar. Such additional observations usually include some form of accelerated life-test, such as a five-day run to determine the endurance of the instrument under vibration. For this purpose adjustable vibrating stands have been made which reproduce the condition of vibration of an airplane instrument board. The steadiness of the pointer and the accuracy of the instrument with and without vibration are also observed in connection with such a test.

49 The *special tests on experimental instruments*, depend, of course, on the nature of the work for which the instrument is to be used, and on the flight conditions which will prevail. The readings in such a test may have to be taken to a higher degree of precision than usual, and may require specially refined standards.

50 For instruments such as aneroid barometers, which involve serious errors due to elastic lag or temperature lag, there can be no more reliable method of determining the corrections necessary for the interpretation of the readings than to make a *flight-history test* by actually reproducing in the laboratory the identical variation of pressure, temperature, etc., which was experienced from time to time during the flight. This is particularly important in connection with the use of barographs for competitive altitude flights. It is now well known that barographs subject to any appreciable amount of elastic lag will read a higher altitude the greater the time elapsed during the flight.

51 It is a short step from some of these more complete testing methods to the general subject of the experimental investigation of sources of error in instruments. Such investigations have formed one of the most interesting features of the work of the Bureau of Standards and are illustrated by Figs. 5 and 6, which are based on experiments with high-grade aneroids.

52 In Fig. 5 the first chart (*a*) shows the ordinary calibration of an aneroid changing the pressure at the rate of an inch of mercury in five minutes. (At sea level one inch of mercury corresponds to about one thousand feet of altitude. At twenty thousand feet one inch corresponds to about two thousand feet of altitude). The calibration curve would be distinctly different at some different speed, due to elastic lag in the corrugated metal vacuum boxes. For the same reason, as shown in the second chart (*b*), the discrepancy between the readings going up and coming down is much greater when the aneroid is held at the maximum altitude

for a considerable time interval. This drift, or change of reading at constant pressure, is shown in the third chart (*c*) as a function of the time elapsed. Such curves may be taken as a criterion of the elastic

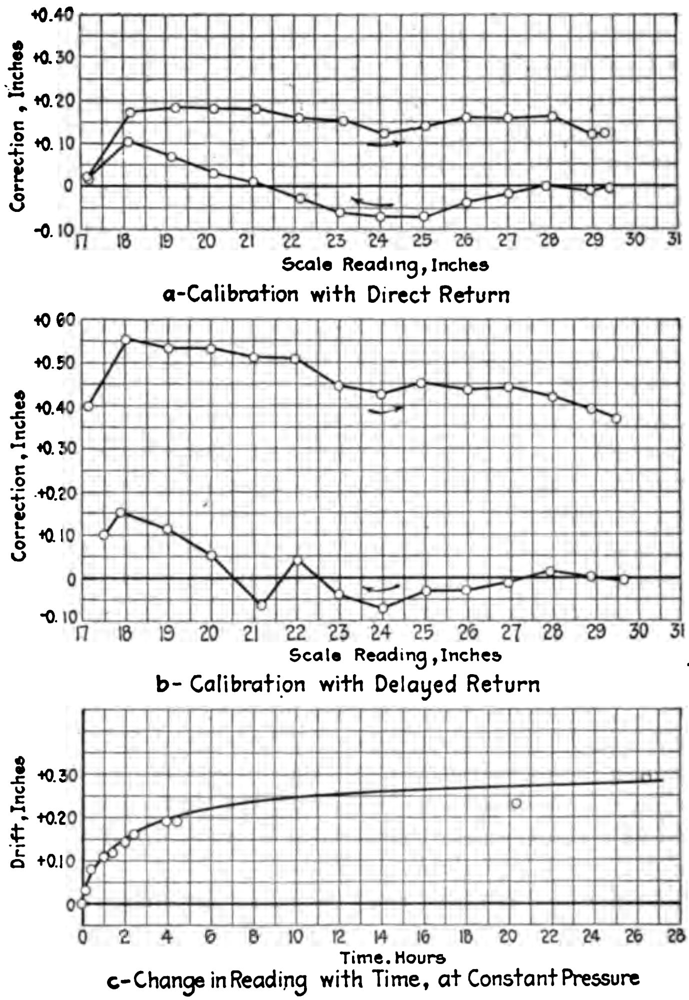
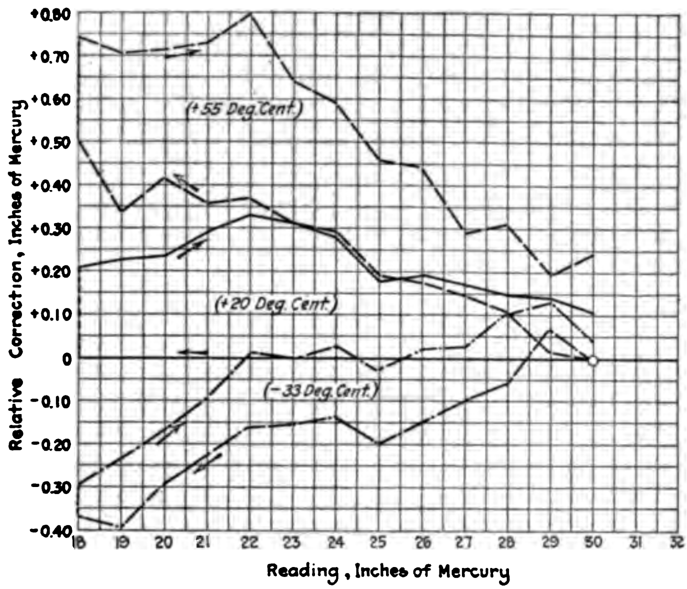


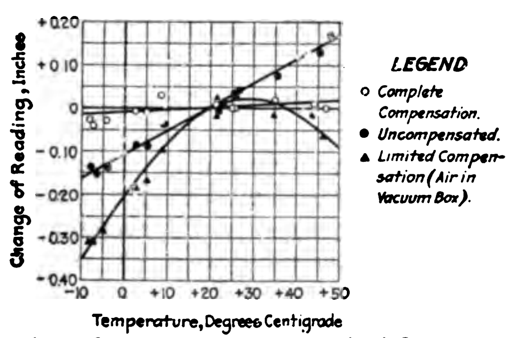
FIG. 5 CHARTS SHOWING EFFECT OF ELASTIC LAG ON ALTIMETERS

quality of the material in the instrument; and several hundred such have been determined in the course of the Bureau's investigations. The curves in Fig. 6-A show the increase in the width of the hysteresis loops with rise of temperature, and it also shows the decrease in the sensitivity of the movement when an instrument is chilled. This last effect is largely due to the change of elasticity with temperature, and cannot be fully compensated for by means



Calibration of Same Aneroid at Three Temperatures.

A



Calibration for Temperature at Constant Pressure.

B

FIG. 6 CHARTS SHOWING EFFECT OF TEMPERATURE ON ALTIMETERS

of the ordinary bimetallic lever. The remaining curves, Fig. 6-B, show, on the other hand, what can be done in the way of temperature compensation. One represents a typical uncompensated aneroid. The graph shows the change of reading with temperature at a constant pressure corresponding to sea level conditions. Another shows a partial degree of compensation obtained by admitting air to the vacuum box. The third shows satisfactory compensation by means of a bimetallic bar so far as sea-level observations are concerned; but this same aneroid may have a serious temperature error when the movement has shifted into the new position which it will occupy at twenty thousand feet.

53 Still further characteristics of aneroid barometers have been studied at the Bureau in addition to those shown in Figs. 5 and 6 and similar investigations have been extended to all of the other aeronautic instruments.

3 USE OF INSTRUMENTS

54 Many practical airplane pilots disclaim the use of instruments, and certainly it is desirable that the aviator should be trained to become just as independent of any mechanical aid as possible. Yet it would hardly do to equip planes without all of the usual service instruments, for those instruments which one pilot thinks he can dispense with are precisely the ones the next pilot would insist upon having. And if the instruments are to be installed at all, even though they may not be read frequently, it is necessary for them to be reliable, and so the whole subject has to be studied and put on a solid scientific foundation.

55 Some of the times when airplane instruments are really of the most importance are the following:

- *a* For various military and naval operations such as bomb dropping
- *b* Long distance navigation
- *c* Flying at night and in clouds
- *d* Preparing to land on a perfectly smooth body of water where the height is deceptive
- *e* Flying in formation with instructions to hold to a prescribed air speed and altitude
- *f* Photographic and survey work.

56 The need for instruments is still more evident in airship work. Here it is necessary to control the position of the ship, especially near the ground, with comparatively greater precision; nor is it possible for the pilot to act so quickly in response to his unaided senses as it might be in an airplane.

57 The following precautions may be suggested for getting the best results in the practical use of service instruments:

a Select for installation in the first place only those instruments which can be certified by the Bureau of Standards as suitable for the intended use. Such approval will ordinarily depend more on the performance of each individual instrument as evidenced by tests, than it will upon the make or design.

b In any event see that the instrument is tested immediately prior to installation to avoid the occurrence of large errors due to secular changes or to mechanical injury. If the errors thus found are not considered negligible, a small correction card can be furnished with each instrument.

c Where accuracy is required in flight observations the aviator must refer to these corrections. He also must remember where appreciable lag errors are stated to exist, whether due to elastic lag or temperature lag or other causes, that some time will be required for the instrument to completely respond to any given change, and that the results will be different with increasing and decreasing scale readings. Instruments are usually adjusted to be correct for increasing readings. If such instruments show an appreciable lag during the test, then in service they will tend to read too high when the pointer is falling.

d At least once in six months it is desirable that instruments should be overhauled, during which time they may perhaps be replaced by a reserve supply of fresh instruments properly tested and adjusted. Instruments which are visibly damaged or out of adjustment may be sent to some central station such as the Bureau of Standards for salvaging. Even if they are not visibly out of adjustment near the zero point of the scale, they may reveal surprising errors further up on the scale if properly tested, and this should be done periodically.

58 The problem of salvaging has been studied by the Bureau of Standards both in connection with the writer's official visit to the various airdromes and salvage stations abroad during the war and more recently in connection with the salvaging of instruments

for the Air Mail Service. The statistical results of such work are of the greatest value as a basis for future improvements in instrument construction, and should provide the raw material for important scientific study. The actual salvage proposition consists of dismantling a certain number of damaged instruments and putting them together again so as to form a smaller number of good instruments. The study of expedients for readjusting the mechanism forms an integral part of such work. This can only be done in the testing laboratory. Now unless it is early discarded in favor of some new type, it will be the destiny of every instrument that is manufactured to come back for salvage or readjustment eventually. Hence the facility with which such readjustment can be made, is a criterion not to be overlooked in connection with the purchasing of instruments. Such readjustments have to be made with due regard to securing a uniform scale and a satisfactory degree of temperature compensation. In some cases it is simpler to prepare new dials graduated empirically to fit the altered performance of the movement.

59 So much for service instruments. The use of experimental instruments, on the other hand, is more obvious, since they do not form a part of the permanent equipment and are called for only when actually needed for special observations. The procedure for such observations is too elaborate to go into here except by way of illustration. Consider, for example, the use of the barograph for high-altitude determinations. The following is an outline of the procedure for determining

a The most probable altitude
b The official altitude for competitive purposes in accordance with the rules of the International Aeronautic Federation.

60 Whichever method of computation is to be used, only such barographs should be selected for the flight as have passed, or could pass, the performance requirements for a Bureau of Standards certificate. Particular attention should be given to the reliability of the clockwork, which is preferably such as to give a rapid movement to the drum: for example, one revolution per hour. The pressure scale should also be as open as possible, which means an extra tall drum. The whole movement must be rigid and strong. The pen must be adjusted without undue friction, but so that the ink will flow freely and trace a delicate but definite line. Charts

graduated in pressure units are convenient, and must be of good quality.

61 Unless the barograph is practically free from temperature errors, a standardized thermometer should be attached to the barograph case and several readings taken during the ascent. And unless the barograph is also free from temperature lag, such a thermometer should be placed inside the case, with its bulb close to the vacuum boxes.

62 For the *most probable altitude* determination, two barographs may be taken, and the results averaged after all corrections have been applied. In this method, the pen may be adjusted before leaving the ground to read the true pressure as determined by a standardized mercurial barometer at the field. Readings of a good strut thermometer or thermograph should be obtained at least every 2000 feet for the determination of atmospheric temperature distribution.

63 To determine the instrument correction at the starting point and at the ceiling by laboratory test after the flight, the flight-history method of testing should be used; that is the barographs should be put in a receptacle within which both temperature and pressure can be made to follow sensibly the same curves with regard to time that were experienced during the flight. By comparison with a standardized and fully corrected mercurial barometer, the true pressures corresponding to the base and ceiling readings can thus be found.

64 The altitudes corresponding to these two pressures are next to be found on the 50 deg. fahr. altitude-pressure table issued by the Bureau of Standards. By subtracting the base altitude from the ceiling altitude thus found, a figure results which represents the relative altitude above the ground on the assumption of an isothermal atmosphere at 50 deg. fahr. This fictitious altitude is then multiplied by a correction factor given in the Bureau of Standards tables, to correct the result for the true mean temperature of the air column. This mean temperature has to be computed in the proper manner. The derivation of the most general altitude formula from fundamental physical principles shows that the particular kind of a mean temperature which is needed here, is the arithmetic mean of a series of temperatures equally spaced with regard to the logarithm of the pressure; or what amounts to the same thing, equally spaced with regard to the isothermal 50 deg. fahr. altitude. It would not be correct to take the mean temperature

for equally spaced intervals of pressure. To the altitude thus obtained, there has to be added, finally, the official altitude of the base station above sea level.

65 For the determination of *competitive altitudes* by the International rule, it is only necessary to carry up one barograph, and no strut thermometer is required.

66 Purely instrumental corrections may be obtained as before, but the method of computation is different. This method of computation is very much simpler than the other, in that it does not require any record of the atmospheric temperature during the flight. It consists of a direct substitution of the indicated ceiling pressure into the International formula. This formula is a definite relation between pressure and altitude, involving only numerical constants. It was based upon sounding balloon observations taken in Europe.

67 In the use of this method, it is understood that the barograph pen should be adjusted, at the start of the flight, to read whatever pressure corresponds to the known altitude of the station by reference to the International formula.

CONCLUSIONS

68 In concluding this paper, a recapitulation may well be made of some of the outstanding problems of instrument development, the solution of which is much to be desired:

a A satisfactory gasolene depth gage is needed.

b Barographs and other recording instruments of better quality than any now existing are needed. Better clockwork, and a much more open scale, and more complete freedom from errors due to friction, temperature, elastic lag, and lack of balance are desirable.

c A wholly satisfactory form of compass for airplanes is not as yet available. Besides the northerly turning error, so-called, to which all single-pivot compasses are necessarily subject, there are troubles due to the action of the damping fluid.

d Air-speed indicators suitable for the low speeds of airship flight are required.

e A gyro-stabilizer is needed, different from existing types in that it must be able to stabilize a body which is not rigid, but contains a freely moving mass. Such a stabilizer is necessary before the full possibilities of any dynamical type of ground-speed indicator can be realized.

69 The material in this paper is of course not due to any one individual, but is based upon the work of the entire Aeronautic Instruments Section of the Bureau of Standards. Later, the detailed official reports prepared by the various members of the staff concerned, should be available through the publications of the Bureau and the National Advisory Committee for Aeronautics.

DISCUSSION

S. Herbert Anderson.[1] The author has given a comprehensive presentation of the status of aeronautic instruments. As he has pointed out, this is a subject of much importance in aeronautic engineering, but one concerning which most engineers have little information. This is not surprising when one recalls that three years ago there was scarcely a single reliable aeronautic instrument of domestic production. And there was a good reason for this — the demand was extremely limited, both in variety and quantity.

When our country entered the war, the need for aeronautic equipment of all kinds brought forth many new industries, among which was the manufacturing of aeronautic instruments. At this juncture the Aeronautic Instrument Section of the Bureau of Standards gave indispensable service in designing, preparing specifications and testing these instruments. However anxious the manufacturers were to produce good instruments, they lacked men of the requisite training and experience to turn out a satisfactory product without careful supervision and inspection.

Because of the immediate demand for usable instruments, very little was done at that time in original design. Instruments in use by the British, French and Italian governments were copied, in some cases in exact detail, in others, with such modifications as were desirable to facilitate production.

There are perhaps two strictly American aeronautic instruments as a result of this war activity: First, the Venturi-Pitot airspeed meter, originally designed by Dr. Zahm of the Bureau of Construction and Repair of the Navy and improved by Major C. E. Mendenhall of the Army Air Service. In this an instrument has been

[1] University of Washington, Seattle, Wash.

developed that surpasses any of foreign manufacture in reliability and length of life. This, the author has pointed out, is due to the large pressure difference produced by the Venturi-Pitot pressure head, which makes possible the use of diaphragm pressure indicators of rugged construction. The second instrument of original American design is a gyroscopic turn indicator, the principle of which was suggested by Dr. A. H. Compton, and which was developed to a practical form by Major Mendenhall.

Need for New Designs of Instruments. Now that there is time for it, it is highly desirable that a systematic program of design of aeronautical instruments be established and carried out. The future of aeronautic engineering depends considerably upon this rather obscure phase of work. The activity of the Bureau of Standards, the Army and the Navy along this line should be encouraged and supported in every possible way. It was a cause of surprise and admiration to the writer, when he visited the Aeronautic Instrument Section of the Bureau of Standards some six months after the armistice was signed, to find the staff all just as busy as formerly, but occupied with a different character of work. Formerly instrument testing was taking a major part of the time and efforts of all; at this later time investigations of various character were in progress, all pointing to the improvement of aeronautic instruments in design and construction.

In his conclusion the author mentioned certain problems the solution of which is to be desired. The writer has had some experience in part of these problems in the flight testing of instruments.

Of the various service instruments used to indicate the performance of the *power plant*, all are quite satisfactory except the *gasoline level gage*. A reliable instrument of this type is much needed. All gages of the float variety have serious mechanical defects, and in addition necessitate the location of the indicator some considerable distance from the pilot, so that visibility is poor, and in some cases the indicator cannot be seen at all from the pilot's seat. For this reason the "hydrostatic" type of gage which permits the location of the indicator on the instrument board is to be preferred. But such an instrument requires at least one tube leading from the gasoline tank which may be a source of trouble and danger if it springs a leak. Various attempts have been made to avoid this difficulty, none of which are entirely satisfactory.

The *magnetic compass* for aircraft is not all that is to be desired. But in the writer's opinion the chief defect is the northerly turning error, which is due to the nature of the earth's magnetic field and cannot be eliminated by any improvement in construction. The compass can never be used on aircraft as it is on watercraft to indicate small deviations or turns from the course, but it is quite satisfactory for indicating the course the airplane or airship is following when flying in a straight line at uniform speed.

This inherent defect of the compass requires the use of some additional instrument for showing slight turns. Safe flying in clouds or fogs is impossible without it. Such an instrument is available in the gyroscopic turn indicator referred to above. This may be made sensitive enough to show the very slightest turn. An instrument that shows the rate of turn is hardly necessary, since the total amount of the turn can be told from the compass when the pilot has straightened out the course. If pilots are properly educated in the use of the turn indicator, and it is used to supplement the compass, aerial navigation in the clouds will offer no greater difficulties than ocean navigation in fogs.

A satisfactory *absolute ground-speed meter* would be the greatest addition to our present equipment of aeronautic instruments that one can think of. It would be of great service both in commercial and military aircraft, for at present the estimation of ground speed from the air speed and wind is hardly more than a guess. But in the development of such an instrument there are two important features to be considered: 1 Any integrating device for determining the ground speed from the acceleration is likely to be bulky and of complicated construction, difficult to keep in adjustment. This is not compatible with the desirability of being fool-proof. The instrument mentioned by the author, *viz.*, the ball rolling in a tube filled with a viscous liquid, is by far the simplest that has come to the writer's attention. It is a very clever idea that has much to commend it. 2 The successful operation of such an integrating device requires almost perfect stabilization, which again requires bulky and complicated mechanism. Consequently the development of a satisfactory ground-speed meter is one of the most difficult problems of aeronautic instruments.

Installation of Aeronautic Instruments. An engineering problem connected with aeronautic instruments is their proper installation. No matter how skillfully an instrument has been designed, and no matter how satisfactorily it has responded to the laboratory tests, it may be worthless if improperly installed. The installation of the engine or power-plant instruments is comparatively simple. But some of the navigating instruments require more attention. Two cases will be discussed.

1 In the case of the compass the magnetism of the ship causes *deviations* of the compass needle from the magnetic north. These must be reduced to a negligible amount by proper compensation, or the deviations must be determined for various positions of the ship and a table of corrections for the compass readings prepared. Generally both methods are used. The *semicircular* deviation, due to the permanent magnetism of the ship, is eliminated as much as possible by placing small permanent magnets of the proper strength in holes in the case above or below the compass bowl. The quadrantal deviation, due to induced magnetism from the earth's field in the soft iron of the ship, may be eliminated to a greater or less extent by placing pieces of soft iron, one on either side of the compass on a line through the pivot and at right angles to the longitudinal axis of the ship. The semicircular deviation is larger and may be more easily eliminated. However the magnetism of the ship frequently changes due to the vibration of the motor and to jarring on landing. Hence the compensations must be frequently made and the residue deviations determined. To insure reliability it should be done before every flight.

2 In the case of the air-speed meter of the pressure type, the location of the pressure head is an important matter. If possible it should be at a point where the air stream is undisturbed in its flow past the ship or any of its parts. Such locations are considerable distances above or below the wings or out in front. In airplane performance testing the pressure head is mounted on a spar extending six to eight feet in front of the leading edge of the wings. But of course such a position is impractical for a service instrument. The writer found that it was possible in the case of a biplane (and the same holds for a triplane) to find a position alongside of one of the interplane struts where the air speed is the same as that of the undisturbed flow. At a position near the lower wing the air speed is higher, and near the upper wing it is lower. This is due of course to the effect of the cambered surface of the wing on the flow of air over and under it. Since speed of flow varies with the type of airfoil, gap and stagger, it is necessary to determine this optimum position for the air-speed pressure head for each type of airplane. As might be expected the position varies with the speed of the ship. However it is possible to find a mean position for the speed range such that the maximum error is only about 1 per cent. This is much better

than placing the pressure head at random and in so doing involving an inaccuracy of from 5 to 25 per cent.

G. C. Westervelt.[1] The writer attaches particular interest to the opinion that the use of instruments should be more generally encouraged. A good many pilots appear to take the ground that the use of instruments is a reflection on their ability. There is absolutely no doubt that the use of instruments is necessary under conditions where there are no objects in view by which the airplane's altitude can be determined.

It is considered highly desirable that pilots ought to learn to use instruments, as thereby they can fly more accurately and obtain the best performance from their planes. The best means of recognizing correct altitude of a plane is through instruments rather than judgment.

It is quite probable that a great deal of the distrust of the pilots in the past has been due to the erratic performances of the instruments with which they have been supplied. Therefore, the work of such men as the author is of greatest importance.

Ford A. Carpenter.[2] The author has incorporated in the brief space of a few thousand words a lucid account of the underlying plans of design, and given the reader all of the details of everyday tests and much interesting data as to the use of instruments in flying.

The writer would like to have the author discuss more fully the equipment for a free balloon and for a dirigible. Also, it appears to the writer that an open-scale recording statoscope for spherical balloons might be worth while. For the latter also a good wind velocity recording apparatus would be appreciated by many pilots.

Alexander McAdie.[3] The paper is most timely and the author has packed a great deal into little space. His classification embraces, the writer believes, all known instruments, and is logical and systematic. We note, however, that the instruments are those of the machine, and little if any mention is made of what we may call instruments for the security and safety of the man.

[1] Commander CC, U.S.N., Manager, Naval Aircraft Factory, Philadelphia, Pa.

[2] Manager, Department of Meteorology and Aeronautics, Los Angeles Chamber of Commerce, Los Angeles, Cal.

[3] Director, Blue Hill Observatory, Readville, Mass.

Speaking broadly, in aviation, we deal with:

1 The machine, *i.e.* plane, ship or helicopter
2 The medium; the air; and all the problems of aerography
3 The man; or intelligence that operates and controls.

The author treats of the instruments used in the plane or ship. His treatment is at once comprehensive and concise and with the promise of the later detailed official reports prepared by the staff of the Bureau of Standards, mechanical engineers should congratulate themselves upon having at hand these general principles and tests of aeronautic instruments.

Under groups 2 and 3, that is, the *medium* and the *man*, the author understands that, at present, the design and testing of such apparatus and instruments are hardly in the province of the mechanical engineer and are rather for the physicist, physician and psychologist. So, we must wait awhile. For example under 2, it is up to the physicist (in this case the aerographer [1]) to give us an instrument for recording vapor pressures, mixing ratios, density, aerodynamic pressure apart from aerostatic pressure, condensation temperatures and cloud base heights, etc.; while under 3, it is up to the physician to devise such instruments as that most important of all instruments — the man himself — may need. For example at high levels the aviator needs help, perhaps as badly as the engine needs a compressor. He gets some in the shape of oxygen, and electrically heated clothing; but is this enough? There is the problem of evaporative cooling — and the need of proper action of the sweat glands, otherwise glandular fatigue and exhaustion result. At high levels we must keep him warm; but also we must keep him sufficiently cool. His perspiration or transpiration must be looked after not less than his respiration.

Coming now to the *machine* instruments which properly fall within the scope of the author's paper; it is not altogether satisfactory that the chief instrument of all, the altimeter, is a camouflaged aneroid barometer; and not well camouflaged at that. For inches of pressure we read feet in thousands; simply a movable altitude scale for the fixed pressure scale of yore, although in all

[1] It may be explained that the term aerography has come into use in the last three years. It means the science of the structure of the air. It supplants the older and awkward term meteorology. The latter has been defined as "weather for the automobilist" while aerography may be considered "weather for the aviator."

recent make high grade surveyor's aneroids, the scale was movable, and the zero datum adjustable.

And here is what seems to the writer to be the most important criticism he can make of the whole paper. The typical altimeter as described and figured (instrument board Fig. 1) is an aggravation. Why use inches and feet when meters are available? The instrument as shown reads to 25,000 ft.; and an airman reading it at that figure would be out 2000 ft. Eight per cent is a rather large error. Using meters will not of itself correct this error; but by using meters we have the larger and more convenient unit; we get in step with a more scientific notation and approach the absolute units, the dynamic meters, and units of force; to which eventually we must come.

But over and above all is the outstanding need of a temperature correction. It is significant that old Dr. John Jeffries who crossed from England to France through the air in 1785 with his mercurial barometer[1] could tell his maximum elevation (2012 meters, 6600 ft.), with greater accuracy than any flyer can today with an ordinary altimeter.

Thermographs and hygrographs are satisfactory. It may interest members of the Society to know that the atmosphere has been quite well explored and pressure, temperature and humidity conditions have been determined with much accuracy for heights far beyond airplane or airship limits. In fact by means of *ballons sondes*, and later, pilot balloons, elevations of nearly 40,000 meters (over 130,000 feet or 25 miles) had been attained, four or five years before the outbreak of the war. At present our planes are just able to reach the top of the *troposphere* or lower shell of our atmosphere — the region of convectional currents. We have not yet flown into the *stratosphere* or upper atmosphere. It is interesting to know that the lowest temperature in the air thus far obtained is 662 grads *i.e.* 130 deg. fahr., at an elevation of 17,000 meters, over the Equator. The balloon went considerably higher, but the temperature rose with increasing elevation.

And this brings us to the need of better temperature scales. We have been using successfully in our aerographic work at Blue Hill Observatory for several years a scale devised by the writer, in which zero is the absolute zero and 1000 the freezing point. There are no minus signs and no degree symbols. In the following table

[1] The writer has examined this old instrument. It is a high-grade instrument and still serviceable.

the heights, temperatures, pressures and densities are given in compact form. This notation is known as the 4 K system.

Km	Miles	T_k	Deg. Fahr.	Kb	$\rho = gm/m^3$
20	12.4	783	−75°F	55	87
19	11.8	787	− 73	63	102
18	11.2	783	− 75	74	121
17	10.6	772	− 80	87	144
16	9.9	772	− 80	102	169
15	9.3	772	− 80	120	198
14	8.7	776	− 78	142	233
13	8.1	783	− 75	167	268
12	7.5	790	− 70	195	314
11	6.8	802	− 66	228	365
10	6.2	816	− 59	266	415
9	5.6	838	− 47	309	470
8	5.0	864	− 35	358	528
7	4.3	890	− 22	413	592
6	3.7	920	− 8	475	662
5	3.1	945	5	543	733
4	2.5	967	16	618	815
3	1.9	989	27	703	905
2	1.2	1008	36	798	1011
1	0.6	1018	41	903	1134
0	0	1033	48	1017	1258

The table referred to above is based upon data given by Dines in his Characteristics of the Free Air supplemented by the work of other investigators. Km is the height in kilometers with equivalent miles, T_k is the temperature on the Kelvin kilograd scale; Kb, the aerostatic pressure in units of force or kilobars, and ρ, the density in grams per cubic meter. The table with some comments can be found in *Science*, March 19, 1920.

As the author states, some form of absolute or ground speed indicator is still needed, preferably one which could function when the ground is not visible. Visibility is too uncertain a condition to be relied upon. It seems to be impracticable for an aviator to determine while both machine and air are in motion, true velocity. But by sending to him from a central point a combined wireless and sound signal, a close approximation of his distance from the source can be made. The interval between the arrival of the two signals made simultaneously will give the time of travel of the sound, i.e. aerial wave; and with proper correction for temperature and density, the distance. There are, however, certain aberrations of the zones of aud bility especially during fog, which will to some degree invalidate the velocities.

The third general group, that of direction indicators promises well; and the installation of light weight gyroscopic compasses gives the airmen dependable directions. Let us ask, however, that on the cards, following recent navigational methods, directions be given in degrees. It would be a step forward to do as the French do, namely, decimalize the circle, and use grads instead of degrees.

It is interesting to note how many instruments have as an actuating principle, pressure difference on two sides of a membrane or metallic diaphragm.

We could wish that the author had added a few words on the proper placement of instruments. There is little doubt that bad "exposure" to use the technical term, influences all pressure instruments. Again, in the case of self-recording instruments, some information as to the degree of vibration experienced would be appreciated. It has been our experience that pens as a rule write decidedly better on a vibrating base than when restrained.

Elmer A. Sperry.[1] During 1913 and 1914 we commenced to appreciate that a magnetic compass on an aeroplane was in serious trouble. The so-called *spinning* of the magnetic compass card when navigation is undertaken through fog creates an extremely serious situation. Although many minds have concentrated on remedying it, the best we can do is to give it a certain known amount of damping. This phenomenon arises from the necessity of out-of-balance support of the magnetic compass needle, due to the element known as "dip;" the accelerations laying hold of this out-of-balance factor, the two finally synchronizing, reacting, and building up without limit.

Referring again to the magnetic compass for smaller aircraft, its unaided performance is so utterly hopeless in fog, cloud and night flights, and its function is so extremely important, that we have developed an important compass safeguard in the form of a gyroscopic turn indicator, operating by simply the venturi side of the little device shown in Fig. 2 of the author's paper. This has come into such general use that I am surprised to find that it is not mentioned in this paper. The way this safeguards the magnetic compass is as follows:

The magnetic compass is found to be of quite high reliability on straight flight, but on straight flight only. In the turn indicator we have an instrument that indicates straight flight with a high

[1] Sperry Gyroscope Co., Brooklyn, N.Y.

degree of precision. Some turn indicators have been made to indicate a turn about a radius as great as five miles, distinguishing clearly between this and straight flight.

In safeguarding the magnetic compass by the use of the turn indicator, very excellent navigation has been secured and great circle flights have been made with the entire absence of land marks, as by its use it is found possible to hold the magnetic compass within reasonable limits of accuracy. The great advantage of dealing with true geographic meridians and the high degree of accuracy attained by the gyro compass make it very desirable for larger aircraft, either lighter or heavier than air.

The difficulty with the magnetic compass is so very great that we have gone far to solve the problem with the gyro-compass, but this will probably always be used for the larger aircraft. The gyro-compass is a precision instrument that deals not with magnetic but with true geographic functions solely and falls strictly within this class.

Two or three years ago the writer did considerable work with a small rate-of-climb indicator, which used a capillary, with which some quite wonderful results were obtained, but the small demand for this instrument did not encourage us to push it farther than building several samples, which gave quite a high degree of accuracy.

ALBERT F. HEGENBERGER.[1] The development of airplane instruments has been so extensive and recent that it would be quite difficult to treat the subject thoroughly in the limited time at this session. The author's presentation probably embraces as many phases and details of the subject as is possible in such a short treatise. Because of this and the lack of time for preparation these remarks are more or less supplementary and extemporaneous.

Until recently almost the sole demand in this country for airplane instruments has been made by the Air Service. The commercial demands for these instruments has not reached a point to warrant a manufacturer to undertake extensive research along these lines. The Air Service therefore must of necessity carry on a certain amount of development work, especially on problems peculiar to military aeronautics.

An example of this is the improvement of the air-speed indicator, against which many complaints have been received. Investigation of this problem requires:

[1] 2d Lieut., A.S.A. Air Service Representative, Dayton, Ohio.

1 Wind tunnel tests on various types of heads to study the magnitude of the errors due to dust, moisture, and obliqueness to the air stream
2 Study of installation of the tubing and connections
3 Measuring the forces involved
4 The efficiency of the various mechanical movements used in different types of instruments.

After this work has been completed a new instrument is designed, built, and tested. If satisfactory the instrument is ready to put in the hands of a manufacturer for production.

The Engineering Division, Air Service, went through a similar procedure in its search for a satisfactory gasoline level gage. After the experimental gage was successfully tested, bids were asked on quantities.

A new type of sensitive barograph is being developed by the Engineering Division. Although the working model has been functioning satisfactorily for several months, minor improvements and refinements are constantly being made, before a quarter-size model is built. This barograph as it now stands will record a change in altitude of about 30 feet.

The value of an instrument of this type for performance testing on aeronautics can readily be seen. Another important use would occur in making a preliminary survey of new territory.

A mechanical rate-of-climb meter was built about a year and a half ago by the Engineering Division which performed very well. Owing, however, to the rapid changing of personnel, work on this instrument ceased until recently when its further development was considered. This instrument used a sylphon for the sensitive diaphram element. A rate-of-climb meter which will indicate when the maximum climb is being obtained would be a great time-saver in performance testing and altitude flights.

Although ground-speed indicators of the visual type have been constructed which will function satisfactorily, nothing as yet has been produced which is light and compact enough to be acceptable. An instrument of this type would be of great value for bombing work. The further development of this instrument is on the program of the Engineering Division.

Improvements in aeronautical instruments in the last three years have resulted in producing instruments of the utmost practical use from what was formerly considered worthless. Certain instruments, however, are still far from satisfactory. This is particularly true of direction indicators. The northerly turning error of the magnetic compass is objectionable. Gyroscopic compasses, which have been tested so far, have been affected so greatly by bumps and similar accelerations as to be worthless. A turn indicator is a necessary adjunct to the compass when flying in clouds. The gyrostatic type should be improved to prevent oscillations of the hand which are very confusing. The static head type is free from this vibration but is affected by sideslips and skids.

A great many instrument problems would be solved if some means could be developed for indicating the true vertical and horizontal. Level flying which is essential to accurate photographic mapping and bombing, would then be much easier.

Chronometric and centrifugal tachometers give very reliable service. The whipping of the long flexible drive shafts however, has become a serious problem in the large multi-motored planes. To eliminate this trouble the Engineering Division is perfecting an electrical drive of the synchronous motor type.

While pilots will continue to be trained to fly by "feel", it is apparent that increasing reliance is being placed on instruments as they continue to improve. It is not at all improbable that in the near future, large planes and planes with enlosed cockpits, will be flown almost entirely by instruments.

J. A. Hoogewerff.[1] We wish particularly to invite attention to Par. 54. The author does not sufficiently stress the point that it is most necessary that the air pilot shall have a thorough knowledge of the instruments available for his use, and a respect for them based on that knowledge. There are marked inherent errors and inaccuracies in instruments which depend for their action on the use of a medium so variable in physical qualities as the atmosphere. The pilot should be sufficiently informed on the subject to appreciate this and to make due allowance for instrumental errors. But he must also realize that good instruments are far more accurate than human judgment and that to depend upon judgment alone may prove fatal. Constant experience with instruments is the best way to learn their virtues and their faults.

Thomas D. Cope.[2] In discussion of the recapitulation of outstanding problems given by the author, it may be said that a Mr.

[1] Rear-Admiral, U.S.N. Superintendent, U.S. Naval Observatory, Washington, D. C.

[2] University of Pennsylvania, Philadelphia, Pa.

Cole of the Air Service worked for some months at Langley Field on a gasoline depth gage. His experimental model seemed to give excellent service and reliable indications. Its new features consisted in a ball and socket endless chain over pulleys actuated by a float. The motion of the upper pulley was communicated through a ground joint to a rack and pinion and thence by a wire, enclosed in a flexible casing, to a dial and pointer on the instrument board.

THE AUTHOR. The various discussions which have been contributed certainly constitute, in the opinion of the author, a valuable source of information and suggestion for future reference by all workers in the field of aeronautic instruments, whether approaching the subject from a military, manufacturing, or scientific point of view.

Comparatively little comment needs to be made in reply because some of the questions raised by one contributor are answered by another, while many other points brought up in the discussion will be treated at length in the forthcoming government publications.

In addition to the American inventions cited by Professor Anderson, mention may well be made of Mr. T. C. Prouty's exceedingly compact form of oxygen regulator, produced by the Van Sicklen Company. The precision altimeter and rate-of-climb indicator developed by the Bureau of Standards are primarily the work of Mr. John B. Peterson and Mr. Atherton H. Mears, respectively. The new type of gyro-stabilizer was designed by Professor W. S. Franklin.

The turn indicator described by Mr. Sperry was briefly referred to in the paper toward the end of Par. 20.

Lieut. Hegenberger outlines some of the investigations needed on account of complaints received against the air speed indicator. A recent investigation of Venturi tubes by Dr. Franklin L. Hunt, Herbert N. Eaton and the present author has brought out one rather interesting point, namely the effect of viscosity, and consequent departure from the simple law of direct proportionality to the density and square of velocity. This research made use of a small "vacuum wind-tunnel," so as to realize by direct experiment the conditions encountered at successive altitudes. The results are of interest in connection with low speed and high altitude flight.

Rear Admiral Hoogewerff emphasizes the need for adequate instruction of pilots on the subject of instrument characteristics. Something was begun along this line by the Bureau of Standards during the war. Early in the summer of 1918 a group of officers were detailed from the Navy to secure specialized scientific instruction at the Bureau for a period of several days; later, another group, all of them navigation officers, were detailed to the Bureau for an intensive three weeks' course of lecture and laboratory work. A graduate course in Aeronautic Instruments of ten weeks' duration now forms a part of the requirement in Aeronautical Engineering at the Massachusetts Institute of Technology. Thus Admiral Hoogewerff's suggestions are already beginning to be realized, and he is undoubtedly right in urging the importance of instrument instruction. It remains only for such instruction to be more widely disseminated, so that it will become available to every actual pilot and not be limited to aeronautical engineers.

No. 1742

THE FLOW OF AIR THROUGH SMALL BRASS TUBES

By T. S. Taylor,[1] Pittsburgh, Pa.
Non-Member

A study of the flow of air through brass tubes ⅝ *in.* (1.58 *cm.*), ⅞ *in.* (2.22 *cm.*) *and* 1½ *in.* (3.81 *cm.*) *in diameter, respectively, has been made by means of small pitot tubes. Under the conditions of the experiment, it has been observed that the velocity distribution does not become constant in tubes of these dimensions until the air has passed through a length of about* 6.57 *ft.* (200 *cm.*) *The ratio of the average velocity to the maximum velocity at the center has been found to have a value of from* 0.82 *to* 0.85 *for all velocities for each of the tubes tested. Tests were made of the influence of dust and oil on the walls of the tubes and very interesting results obtained. A small quantity of dirt irregularly distributed greatly diminishes the total air flow for a given static pressure and also produces a marked change in the velocity distribution, the average velocity being made considerably less with respect to the maximum velocity.*

THE investigations discussed in the present paper were undertaken as a preliminary study of the general ventilation problem, dealing in particular with the flow of air through tubes of various sizes and shapes. At present, literature on ventilation contains very little information along this particular line, useful to the designing engineer. Quite a little attention has been given to the mathematical consideration of the flow of fluids through tubes, but so far no special experimental study has been made of the factors influencing the flow of air through small tubes.

2 In the first place, no very satisfactory device has as yet been made for measuring gas flow through such tubes as are dealt with in the present discussion. The tubes thus far tested in this investigation were of seamless brass having internal diameters of ⅝ in. (1.58 cm.), ⅞ in. (2.22 cm.), and 1½ in. (3.81 cm.) respectively.

[1] Mellon Institute, University of Pittsburgh, formerly of the Westinghouse Electric and Manufacturing Co., East Pittsburgh, in whose Research Laboratory the work herein described was done.

Presented at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers.

In order to determine the velocity distribution of the air across these tubes a hot-wire anemometer with thermocouple was developed. This was described in principle in a short paper presented before the American Physical Society at Baltimore in December 1918, and more fully at the Symposium on Pyrometry under the auspices of the American Institute of Mining and Metallurgical Engineers and the Pyrometry Committee of the National Research Council at Chicago, September 1919. This instrument was used in various forms such as: a plain cross as described in the papers cited; a small solenoid about $\frac{1}{16}$ in. (0.159 cm.) in diameter, having thermocouple embedded in a high-melting-point wax in the middle of the solenoid; and with heating wires in shape of a flat rectangle, the thermocouple being attached to the middle of the short side. The method of use was the same in all cases. This consisted in measuring the electrical energy required in the heating wire to maintain the temperature of the thermocouple a definite amount above the temperature of the air in which it was placed. The device proved to be a very satisfactory one for laboratory experiments. For shop use, however, it is not satisfactory as it requires accurate manipulation and is not "fool-proof." Attention was therefore turned to the usual pitot tube method. In the preliminary study it was only necessary to have a tube for measuring the total pressure, as the static pressure could be measured by means of a small tube in the side of the main tube through which the air was passing. The pitot tubes used were made from hypodermic needles about $\frac{5}{64}$ inch (0.198 cm.) in external diameter. The velocity pressure was measured in the usual way by means of inclined oil gages. The use of a special low-pressure differential gage was found unnecessary.

3 The air current was furnished by means of a No. 3 V blower made by the American Blower Co., and was driven by a Westinghouse d.c. motor which could be run so as to give any desired speed of revolution of the blower. To prevent vibration of blower and motor being transmitted to the tube, a short rubber hose was used for connection between blower outlet and the brass tube used in the experiment. From the readings of the inclined gage the velocity of the air was calculated by the usual formula

$$V = \sqrt{\frac{2\,g\,h\,d}{d'}}$$

where g is the acceleration due to gravity, h is the difference in level in the gage, d is the density of the medium in the gage, and d' is the density of the air. All velocities were calculated and reduced to standard conditions of 29.93 in. (760 mm.) pressure and air temperature of 77 deg. fahr. (25 deg. cent.).

4 A study was made of the velocity distribution at various positions along a 1½-in. (3.81 cm.) brass tube 8 ft. (243.6 cm.) long. This was done with a pitot tube about $\frac{1}{40}$ in. (0.0635 cm.) in external diameter inserted in the brass tube. The curves in Fig. 1 show the distribution of velocity for three different positions. The velocity at the center of the tube is seen to increase as the distance from the end nearest the blower increases. For position 1, 2.58 ft. (78.6 cm.) from the end of the tube nearest the blower, the maximum velocity is seen to be 6000 ft. (1828 meters) per min.; for position 2, 5.07 ft. (154.5 cm.) from end nearest the blower, it is 6200 ft. (1889 meters) per min., and for position 3, 7.57 ft. (230.7 cm.) from the end nearest the blower, the maximum velocity is 6270 ft. (1910 meters) per min. The average velocity for each of the cases is 5300 ft. (1615 meters) per min. This average velocity was obtained in the following manner. The cross section of the tube was divided into several small concentric zones and the area of each zone calculated. The area of each zone was then multiplied by its average velocity. The sum of the products for all zones was then divided by the area of the cross section of the tube which gave the average or mean velocity of air flow. In other words, this is equivalent to finding the volume described by revolving the curve for the velocity distribution about its axis and dividing this volume by the area of the tube. These results show clearly that the velocity distribution of air flow-

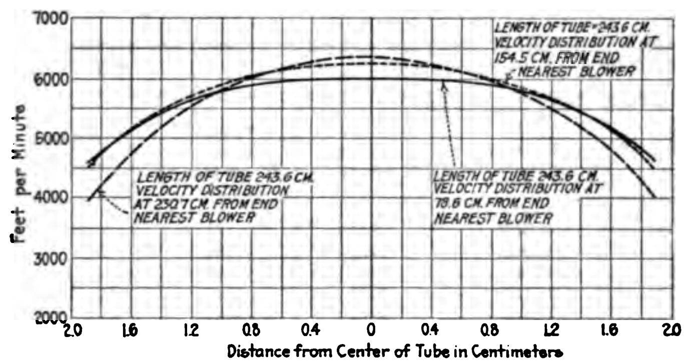


Fig. 1. Velocity Distribution at Three Points in 1½-in. (3.81 cm.) Tube

ing through tubes does not reach a steady state in the distance in which it is usually assumed that it does. It is customary to assume a steady distribution to exist in a length of tube equivalent to 10 times its diameter.

5 In order to test this last point a little more, a second tube 13.33 ft. (407 cm.) long and 1½ in. (3.81 cm.) in diameter was used. It was found that the velocity distribution had attained a definite condition and was the same for all points distant 7.88 ft. (200 cm.) from the blower end of the tube. Another rather interesting point was observed in regard to the ratio of the average velocity to the maximum velocity for various velocities. The results in Table 1 show that the average velocity is the same percentage of the maximum velocity for all velocities from 800 ft. (243.8 meters) per min. to

TABLE 1 SHOWING RATIO OF AVERAGE VELOCITIES TO MAXIMUM VELOCITY AT VARYING AIR VELOCITIES IN 1½-IN. BRASS TUBE

Blower Speed, r.p.m.	Static Pressure, in.	Maximum Velocity, ft. per min.	Average Velocity ft. per min.	Ratio $\frac{\text{Average Velocity}}{\text{Maximum Velocity}}$
435	0.0465	872	751	0.862
739	0.0913	1410	1186	0.853
983	0.146	1912	1610	0.842
1098	0.176	2144	1804	0.842
1490	0.322	2973	2480	0.834
1857	0.437	3784	3216	0.850
2710	0.980	5373	4460	0.830
3150	1.385	6330	5460	0.863
				General Mean. 0.847

6300 ft. (1919 meters) per min. These results were all taken in the long tube 407 cm. at a point 9.45 ft. (240 cm.) from the end adjacent the blower. The values of the static pressure were taken at the point along the tube when the velocity distribution was determined, and thus represent the pressure which is necessary to maintain the air flow through the remainder of the tube or 6.58 ft. (167 cm.) The fact that this ratio of the average velocity to the maximum is approximately the same for all velocity distributions seems quite interesting. Similar results were also obtained for tubes ⅞ in. (2.22 cm.) and ⅝ in. (1.58 cm.) diameters respectively. The general mean for the ratio of the average to the maximum velocity was found to be 0.830 for the ⅞ in. (2.22 cm.) tube and 0.870 for the ⅝ in. (1.58 cm.) tube. These results would indicate that there is little difference between these ratios for different sized tubes provided the surface conditions are identical. These results indicate that the velocity distribution, for steady state, across smooth brass tubes is practically the same for all tubes and all average velocities between 0 and 6000 ft. (1828 meters) per min. It is interesting to note that the velocity near the wall of the tube is so large as compared with the maximum velocity at the center. The velocity must therefore drop off very rapidly close to the wall of the tube. This drop occurs in such a short distance that it cannot be observed with a pitot tube $\frac{1}{40}$ in. (0.0635 cm.) in diameter.

6 Having determined the above data on the air flow through smooth tubes, the next step was to see what effect would be produced by the addition of dust to the walls of the tubes. Oil was poured through brass tubes of 1½ in., (3.81 cm.) ⅞ in. (2.22 cm.) and ⅝ in. (1.58 cm.) diameter respectively, each being 5 ft. 1⅛ in. (1.553 meters)

TABLE 2 COMPARISON OF VELOCITY RATIOS IN CLEAN AND DIRTY TUBES

Diameter of Tube, in.	Clean Tube. Ratio of Average to Maximum Velocity	Dirty Tube. Ratio of Average to Maximum Velocity	Approx. Thickness of Film of Dirt, in.	cm.
1½	0.832	0.749	0.006	0.0152
⅞	0.833	0.648	0.016	0.0406
⅝	0.825	0.795	0.008	0.0203

long. Coal dust was then sifted through the tube and only that was allowed to remain which was not readily jarred out when the end of the tube was struck against some object. The velocity across each tube was then measured near the outlet end for various air velocities. The results obtained are summarized in Table 2.

7 The influence of dirt collecting on the walls of tubes is quite marked upon the flow of air through them. The thickness of the dirt layer is only approximate. The dust and oil were wiped out of tubes after measurements were taken and the thickness of the layer determined from its mass and density. The dirt does not remain uniformly distributed over the tube but gathers in small clusters. This is doubtless the reason why the results in Table 2 are not comparable with each other for the various tubes. In making the measurements of the air flow through the dirty tubes, the air was first caused to flow through the tube at its maximum velocity in order that whatever disturbance would be produced in the distribution of

the film during an experiment would occur before definite measurements were taken.

8 The curves in Fig. 2 show the difference in the static pressure necessary to force air at various average velocities through clean and dirty tubes as specified in the figures. These curves and Table 2 give very interesting data. They show that a very small but irregular distribution of dirt upon the surface of smooth tubes will have a very decided effect upon the amount of air that will pass through the tube for a given static pressure. The effect is not merely a

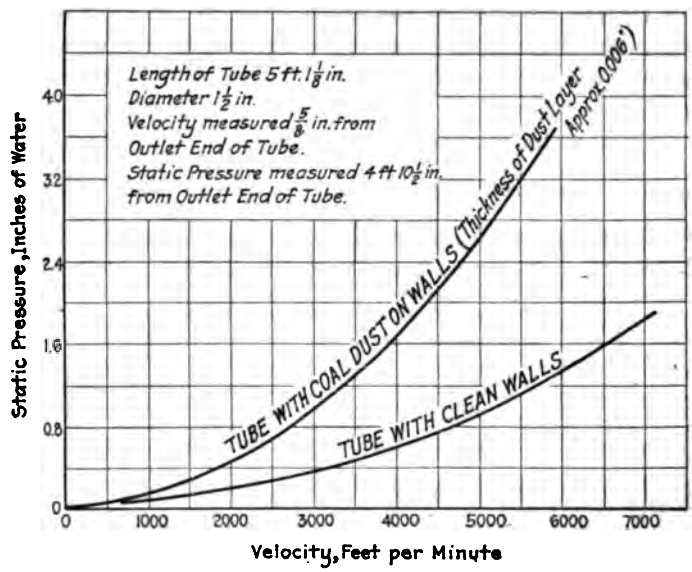


FIG. 2 EFFECT OF DUST IN TUBE

diminution in the effective area of cross-sections but must give rise to turbulency since the ratio of the average velocity to the maximum is so much influenced by the presence of the dirt. The diminution in this ratio due to the dirt must depend chiefly upon the manner in which the dirt is distributed. At any rate the distribution must be of considerable more importance than the amount per unit area as can be seen by reference to Table 2 and Fig. 2. The values of the static pressure to maintain an average velocity of 3000 ft. per min. through the tubes when clean and dirty as outlined above are given in Table 3.

9 While the results of the present experiments should be considered as only preliminary, they nevertheless bring out some rather interesting points. The striking features about them are to be found (*a*), in the way in which the velocity distribution changes along a tube; (*b*) in the relative relation of the ratios of average velocity to maximum velocity for different tubes as well as different velocities; and

TABLE 3 STATIC PRESSURE IN INCHES OF WATER TO MAINTAIN AN AVERAGE VELOCITY OF 3000 FT. PER MIN.

Diam. of Tube, in.	Clean	Dirty
1½	0.37	0.94
¾	0.80	3.20
½	1.00	5.00

(*c*) in the influence that dust collected upon the wall of a tube exercises upon the total flow of air under a definite static pressure.

10 In conclusion I take pleasure in acknowledging my indebtedness to the Westinghouse Electric and Manufacturing Co. in whose research laboratory the work was done, for furnishing the facilities for carrying out the experiments, and in particular to Mr. C. E. Nolan for his able assistance throughout the work.

DISCUSSION

H. Wade Hibbard. At the end of Par. 4 and the beginning of Par. 5, the author says that it is customary to assume that steady distribution within the pipe is attained in a length of tube equivalent to ten times its diameter. His tests on a tube 1½ in. in diameter showed that it required a length equivalent to 50 diameters to obtain a stable distribution. In a 3-ft. pipe, 800 ft. long, designed for carrying sawdust and shavings, a piece of metal, accidentally carried along with the blast, will dent the conduit as much at the end as at the beginning.

Mayo D. Hersey. Has the author compared his experimental results with the well-known formula of Dr. Lees? This is recognized in England as covering the law of flow of fluid through any size of pipe. It would be of value to have the comparison made.

THE AUTHOR. If the author understands Professor Hibbard correctly, he means to infer that it is impossible to get a steady distribution of the air stream in any tube regardless of its length. The fact that the piece of metal carried along with the air blast in a 3-ft. pipe, 800 ft. long, dents the conduit apparently as much at the end as at the beginning, says practically nothing with respect to the uniformity of distribution of the velocity along the channel. In this case we are dealing with objects heavier than air and we may expect them to go through more or less regular oscillations back and forth from the walls into the air stream. Such tubes would by no means have a uniform smooth surface and furthermore the density of sawdust, etc., would be a much greater near the lower part than at any other section of the tube. Turbulency exists for various velocities in all tubes at certain velocities, but for a given tube we can easily have a regular turbulency set up so as to give the resultant effect of a steady distribution for the velocity in question.

As yet there has been no comparison made between the experimental results and the well-known formula of Dr. Lees as suggested by Mr. Hersey. This comparison will be made very shortly however, in connection with some further work on the flow of air through channels and tubes.

No. 1743

SYMPOSIUM ON CASTINGS

THE Sub-Committee on Foundry Practice has secured for the Society a set of papers on the products of the several types of foundries, believing that this information would be of greater value to the members than data as to the details of foundry practice.

Mechanical engineers are called upon to decide or advise on the selection of the proper kind of a casting for a given piece of construction and should therefore know or have available reliable information as to the present state of the art in foundry practice. The Committee has accordingly endeavored to have this information presented in such a way as to not trespass on the territory of the American Foundrymen's Association as to the art or how castings are made; on that of the American Society for Testing Materials as to specifications or methods of testing castings; nor on that of the Mining Engineers and Chemists as to the production of the metals from which the castings are made.

Each paper is supposed to outline the special properties of that particular kind of casting and where it is to be preferred to other forms of cast metal. Also to outline what is actually being done so that engineers and designers may know what they ought to be able to secure from first-class commercial foundries at the present time.

No attempt has been made to secure information as to what is going on in the line of research work in the foundry, as it was thought best to present a reliable statement of what foundries are now doing in a commercial way rather than of what it may be possible for them to do in the future.

The Committee has endeavored to secure the highest available authority for each of the several common types of castings and is under great obligation to these men who have written the papers and presented them to the Society. This is especially true of the several authors who are not members of this Society.

There will doubtless be many differences of opinion as to some of the statements made in these papers, but it is sincerely hoped that the members will freely question all points at issue so that after a full discussion the authors may be able to complete their papers in such a way that the set as a whole will be generally accepted as a thoroughly reliable report on the products of modern foundry practice.

W. W. Bird, *Chairman*
Stanley G. Flagg, Jr.
Benj. D. Fuller
Robt. E. Newcomb
Chas. E. Knoeppel
Wilfred Lewis
Richard Moldenke
Sub-Committee on Foundry Practice

No. 1743*a*

MALLEABLE CASTINGS

BY ENRIQUE TOUCEDA,[1] ALBANY, N. Y.
Non-Member

Malleable-iron castings, according to the author, are regularly made weighing as high as 500 lb., with an ultimate strength of 50,000 lb. per sq. in. and an elongation of 10 per cent or over. With these properties in mind he compares such castings with gray-iron and steel castings, showing that with respect to the former the malleable casting has equal physical properties, costs no more, is lighter in weight and is not liable to breakage. More than 70 per cent of the steel castings produced in this country, it is stated, are not annealed and are in a state of internal strain. Annealed malleable castings are free from such strain, are less expensive to produce, and in the author's opinion will stand more abuse in service. Their use is therefore indicated and recommended for any part of such size as can be successfully made where strength combined with ductility and low cost per pound is essential.

IN general, the selection of the proper kind of a ferrous casting for use in a given piece of construction is dependent upon a number of considerations, chief among which may be mentioned the following:

a Ability to successfully withstand the abuse to which it will be subjected in service

b High static and dynamic strength, which implies minimum sections for the strains involved, this in turn implying low cost for raw castings, as such are not only less costly in direct proportion to their strength, but their appearance is enhanced thereby

c Ease of machining, accompanied by a good surface on faced and turned parts, and freedom from unsoundness; which implies beauty of finished castings and low cost due to less loss from defective castings

d Smoothness of surface, and trueness to pattern.

2 In discussing the foregoing a fact should be remembered that is frequently overlooked, viz., that the gray-iron, the malleable-iron and the steel casting have each a legitimate field of their own

[1] Consulting Engineer.

Presented at the Spring Meeting, St. Louis, Mo., May 1920 of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

quite sharply defined by virtue of certain peculiarities possessed by each particular product. If a bedplate is desired for a certain large and ponderous machine, so dimensioned as to possess sufficient mass to enable it to absorb vibration, it would be unwise to specify that it be made of malleable cast iron. The malleable-iron founders are not at the present time equipped to cast and anneal castings of such size, and as cheapness and weight — obviously not high strength per square inch — are the dominant requirements in this case, the malleable-iron casting should not be specified. The same remarks hold good in the case of the steel casting, except of course that it is practical to make such castings as large as any made of gray iron.

GRAY-IRON CASTINGS VS. MALLEABLE-IRON CASTINGS

3 The true legitimate field of gray iron, in the writer's opinion, should and will in time be restricted to the production of that size of casting that cannot successfully be produced in malleable iron and is too costly to produce in steel, while it is peculiarly well fitted for such castings as are designed to sustain static compression, for which purpose the gray-iron casting is without question superior to either of the other two. The average ultimate tensile strength of the gray-iron casting as measured by the test bar does not exceed 20,000 lb. per sq. in., while it possesses practically no ductility. The writer can state with positiveness that in the case of at least sixty of the malleable-iron foundries the average ultimate strength and elongation of their product, as measured by the standard test bar, are 51,000 lb. and 12.50 per cent, respectively.

4 It can safely be stated that if a gray-iron and a malleable-iron casting are designed to perform the same function, the design of each being based strictly upon its physical properties, in view of the great difference in their relative strengths, coupled with the fact that a larger safety factor must be used in the case of the former due to its inherent brittleness, the latter can safely be made more than one-third lighter. Inasmuch as this difference in weight will in large measure counterbalance the difference in cost per pound with the substitution of a ductile casting for one that is brittle and is actually more easy to machine, the writer believes that when this fact is fully understood and its significance appreciated, many castings now made of gray iron will be replaced by malleable, with the result that the gray-iron-casting field will of necessity be narrowed

to this extent. It is therefore logical to conclude, as between these two products, that if the part can be successfully cast of malleable iron, the following statements hold true:

a For equal physical properties it is as cheap per raw casting
b It is as cheap per machined casting
c It will have a more pleasing appearance when assembled
d It will be lighter in weight
e Transportation charges will be less
f The weight per square foot on the floor will be less
g Breakage during transportation will be eliminated.

STEEL CASTINGS VS. MALLEABLE-IRON CASTINGS

5 If the same line of reasoning be used as between the malleable iron and the steel casting, it can be stated at the start, that neither can successfully trespass in certain directions in the field of the other, for each, as will be explained, has certain limitations that control the situation. There is hardly a limit to the size or weight in which the steel castings can be made, while as previously pointed out there is a limit in the case of malleable iron. On the other hand, there are castings that can be very nicely made of malleable iron that cannot well be made of steel, using the same pattern. When such is attempted, it will be found that the steel founder will request that he be permitted to thicken the sections in order to allow for the lesser fluidity of his metal.

6 The American Society for Testing Materials standard specifications for steel castings deal with two classes: Class A, for which chemical but no physical requirements are specified and which need not be annealed unless so specified; and Class B, which is divided into three grades, hard, medium, and soft, it being specified that all of these grades shall be properly annealed, and also conform to certain chemical and physical requirements. While the ultimate strength of both Class A castings and the soft grade of Class B castings may average higher than that of malleable iron, their yield point will average lower by some 4000 lb. Consequently for equal stiffness the malleable-iron casting has an advantage in the case of equal sections.

7 Inasmuch as the A.S.T.M. Specification for Class A castings states that they need not be annealed unless so specified, it is to be expected that the majority of these castings, do not receive that treatment. This means the presence of internal strains in

such castings, minimized to a certain extent, however, if baked molds are used. The writer knows of quite a number of large steel-casting companies that turn out a large daily tonnage of castings that are never annealed and assumes that in such cases they are made in accordance with specifications other than those of the A. S. T. M., and after considerable inquiry he has come to the conclusion that 70 per cent of the steel castings produced in this country are not annealed.

8 It happens that the heat treatment given the malleable-iron casting during the period in which the white iron is being converted into the malleable product, is carried on at a temperature just slightly in excess of the critical range, a temperature which yields practically the finest crystallization that the metal can attain, while the cooling from this temperature is effected at the rate of about 10 deg. per hr. In consequence of this the malleable-iron casting is not only fine-grained but is invariably free from internal strains. On the other hand, not only are unannealed steel castings in a state of internal strain, but they possess a coarsely crystalline structure corresponding to the temperature of solidification. In view of all this, the writer believes that if a given casting be made of malleable iron it will stand more abuse in service than will a similar unannealed steel casting.

9 If the cost of the raw casting is the dominant factor, then unquestionably preference must be given to the malleable-iron casting, as the difference in cost is considerable. If based upon the cost of the finished casting, the difference between the cost of the two will be much greater, due to the greater ease with which the malleable casting can be machined, which difference will vary directly in proportion to the amount and character of the machining operations.

10 The malleable-iron casting is characteristically free from blow-holes, while this is acknowledged to be one of the shortcomings of the steel casting; consequently, there will be less scrap in the case of the former. If rust-resisting properties are a consideration, as in the case of castings for refrigerator cars, the malleable-iron casting should be given preference over steel.

11 Concerning the developments in the malleable-iron industry due to the war, it can be stated that none were occasioned along metallurgical lines, for the reason that concentrated effort in this direction had been previously made in a very thorough and complete manner through the concerted action of a large number

of malleable-iron manufacturers, and such matters were well in hand even at the start of the war. Very few of the officers connected with the Ordnance Department were familiar with the physical characteristics of these castings, and the exigencies of the situation due to unpreparedness, coupled with the customary "red tape," served to a great extent to prevent those from substituting such castings for products with which they were acquainted. Owing to this the malleable-iron casting was not used for many purposes for which it was eminently fitted. As the product had not heretofore been used in connection with ordnance material, some development took place to this extent at least, for it was finally discovered by the Ordnance Department, at a rather late date however, that many castings formerly made of brass, bronze and steel could advantageously be replaced by malleable iron at a much lower cost. A very large tonnage was used in the manufacture of automobiles, trucks, tractors, tanks, etc., while a certain tonnage was used for hand and rifle grenades. Had the war lasted another year, the opinion is ventured that a large tonnage would have been used for various purposes aside from those mentioned.

PROPERTIES AND LIMITATIONS OF MALLEABLE-IRON CASTINGS

12 In commercial practice the engineer who has placed his contract with a reliable foundry can depend upon a uniformity of product that will rarely have a yield point lower than 31,000 lb. per sq. in. and frequently as high as 33,000, an ultimate strength less than 50,000 lb., or an elongation less than 10 per cent. He can quite safely depend upon the integrity of the casting, for the reason that the founder has finally learned through costly experience that freedom from shrinkage depends not only upon correct gate emplacement, but more particularly upon the use of large shrink heads, so located as to eliminate such defects. Castings have been produced commercially as long as 5 ft., with sections at some parts as thick as 3 in. In regular practice castings varying in weight from 300 to 500 lb. are made daily, while it would be difficult to place a limit on how small they can be run.

13 The wearing properties of this metal correspond to what can be expected of wrought iron or dead soft steel, while as is well known its machining properties can hardly be excelled. As far as the ordinary uses of malleable cast iron are concerned, the writer feels quite certain that no beneficial results can follow from its

heat treatment. In a limited number of cases, however, such as in the production of cheap hatchets, cleavers, and axes, malleable iron can be made file-hard by heating to just above the critical range and quenching.

COMPARATIVE COSTS OF GRAY-IRON, STEEL AND MALLEABLE-IRON CASTINGS

14 A general statement relative to the comparative cost of manufacture of the three products mentioned must obviously involve the consideration of many different cases and conditions. If it were not for the fact that in the case of the manufacture of malleable iron the annealing (which lasts from six to seven days and is a rather costly part of the process) must be taken into account, an approximate estimate could be made by figuring out the cost of the metal in the ladle for each of the three products; but the varying cost of molding in each case and the fact that the sprue can be easily knocked off from both gray and malleable-iron castings while it must be machined or cut off from the steel casting, render it rather difficult to make a comparison of the costs of the raw castings. In general, however, gray-iron castings run heavier than do those made of malleable iron. Consequently there is less sprue and therefore less remelt. Also there is less variation between the composition of the sprue and that of the charge and as a consequence a much larger percentage of scrap can be used in the mixture when making gray-iron than is possible in the case of the malleable castings. Assuming the highest fuel ratio in each case, 10 of iron to 1 of coke for the cupola, and 3 of iron to 1 of soft coal in the air furnace, it will be seen that the cost for fuel is less in the former than in the latter case. The same holds true in connection with furnace maintenance. Aside from cleaning and clipping, the gray-iron casting is finished when the metal fills the mold, while the white-iron castings must be cleaned, taken to the annealing room, packed in saggars, and then charged into the annealing oven and heat-treated for a period in most instances of seven days. The saggars must then be removed from the annealing oven, the castings again cleaned and sorted, to the cost of which must be added that of the fuel used for heat treatment, oven maintenance, supervision, and overhead. It can therefore be very easily seen that there must be considerable difference between the cost of production of these two products. The writer believes that on an

average it will easily cost 1 cent a pound more to produce malleable-iron than it will the gray-iron castings, and at least 30 per cent more to produce steel castings than malleable.

15 The malleable-iron casting is regularly used and can be recommended in the fabrication of railway cars, agricultural implements, motor vehicles, tractors, link belts, chain, fittings, stoves, etc., and is adapted for use for any part of such size as can be successfully made where strength combined with ductility is a necessity and low cost per pound an essential.

No. 1743b

DIE CASTINGS

By Charles Pack,[1] Brooklyn, N. Y.
Non-Member

The die-casting process, according to the author, is best adapted to alloys of comparative low fusing points, such as those of zinc, tin, lead and aluminum. While it is stated that no general rules can be laid down governing the design and application of die castings, the paper nevertheless outlines the general properties of the various alloys used and their fields of application and gives particulars regarding such limitations as maximum weight of casting, minimum wall thickness, minimum number of threads and minimum diameter of holes that can be cast, draft for cores and side walls, etc.

DIE CASTINGS may be defined as castings made by forcing molten metal, under pressure, into a metallic mold or die. It is erroneous to assume that all die castings have similar properties, since it is apparent that the properties of the die casting will depend upon the nature of the alloy used. The die-casting process is best adapted to alloys of comparatively low fusing points which may, for convenience, be divided into the following groups:

Group A Zinc Alloys, consisting essentially of zinc alloyed with tin, copper, or aluminum
Group B Tin Alloys, consisting essentially of tin alloyed with copper, lead, or antimony
Group C Lead Alloys, consisting essentially of lead alloyed with tin or antimony
Group D Aluminum Alloys, consisting essentially of aluminum alloyed with copper.

2 No general rules can be laid down governing the design and application of die castings since the art depends largely upon the skill of the designers, and quite frequently a part may be considered as impractical from a die-casting standpoint which, if measured by given standards, may be redesigned and die-cast very successfully.

[1] Doehler Die Casting Co.

Presented at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers.

Nevertheless, the writer will endeavor to outline briefly the general properties of the alloys used, their fields of application and their limitations.

GROUP A — ZINC ALLOYS

3 *Typical Alloy.* A typical zinc alloy for die castings is that whose composition is given below, together with particulars regarding its general properties and the casting limits which must be observed.

Zinc	87.5 per cent	Copper	4.0 per cent
Tin	8.0 per cent	Aluminum	0.5 per cent

Properties:

Color	Silver white
Weight per cu. in.	0.253 lb.
Melting point	780 deg. fahr.
Initial fusing point	275 deg. fahr.
Tensile strength	16,160 lb. per sq. in.
Elongation	2 per cent
Compressive strength	27,670 lb.
Pressure required to shorten bar 1 in. diameter	10 per cent
Hardness number (Brinell)	64.6

Casting Limits:

Maximum weight for casting	8 lb.
Minimum limit of wall thickness	$\frac{1}{10}$ in.
Small castings	$\frac{1}{16}$ in.
Variations from drawing dimensions per inch of diameter or length	0.001 in.
Cast Threads (minimum number):	
External	24 per inch
Internal	depends on conditions, often cast
Cast Holes: minimum diameter	0.031 in.

(Depends largely upon the depth and thickness of casting.)

Draft: Cores, 0.001 in. per inch of length or diameter. Side walls, 0.001 in. per inch of length.

4 *General Design.* Sections of castings should be as uniform as possible. Sharp corners should be avoided and fillets added whereever permissible. Undercuts in castings should be avoided wherever possible.

5 *General Remarks.* Alloys of this type are corroded by any alkaline or aqueous solutions of any salts. Castings may be polished to a high luster, but soon tarnish when exposed to ordinary atmospheric conditions. Castings made from this alloy may be readily plated with nickel, copper, brass, silver, or gold. When properly

plated such castings will retain their luster as well as those made from brass or bronze.

6 *Applications.* Castings made from this alloy should not be used for parts that are subjected to severe stress or sudden shock in service. They are used extensively for parts of phonographs, calculating machines, drinking-cup, cigar, candy, stamp and gum vending machines, magneto housings, automobile-body trimmings, pencil-sharpening machines, time-recording devices, stamp-affixing machines, and for many other devices of a kindred nature.

GROUP B — TIN ALLOYS

7 *Typical Alloys.* The five compositions given below are typical die-casting alloys of the tin group:

	Tin Per Cent	Copper Per Cent	Lead Per Cent	Antimony Per Cent
No. 1	90	4.5	0	5.5
No. 2	86	6	0	8
No. 3	84	7	0	9
No. 4	80	0	10	10
No. 5	61.5	3	25	10.5

8 Alloy No. 1 is a so-called "genuine babbitt" metal and was used very extensively during the war for main-shaft and connecting-rod bearings on all American-made aeroplanes and motor trucks. No. 2 is somewhat harder and is used extensively for bearings in internal-combustion engines. No. 3 is somewhat harder than alloy No. 2 and is the S. A. E. standard for high-grade internal-combustion-engine bearings. No. 4 is in general use for light bearings on stationary motors. No. 5 is a bearing metal for light duty and is used on a large number of moderate-priced automobiles for main-shaft and connecting-rod bearings.

9 In addition to the five compositions mentioned, hundreds of similar alloys may be made having various specific properties. A study of these alloys, however, would prolong this paper unduly, and in the opinion of the writer, is beyond its scope. The die-casting process, it may be said, is applicable to any of the alloys of this group and it may be left with the engineer to specify the alloy best suited to his requirements.

General Properties:

Maximum fusing point................................450 deg. fahr.
Weight per cu. in...........................depends on lead content

Casting Limits:

Maximum weight for casting .... 10 lb.
Limit in wall thickness .... $\frac{1}{32}$ in.
Variations from drawing dimension per inch of diameter or length .... 0.0005 in.
Cast Threads, minimum number:
External .... 27 per inch
Internal .... depends on conditions, often cast
Cast Holes: minimum diameter .... 0.031 in.
(Depends on depth and thickness of casting)
Draft: Cores, 0.0005 in. per inch of length and diameter. Side walls, 0.001 in. per inch of length.

10 *Applications.* Tin alloys find their largest field of application in their use as bearings for internal-combustion engines. They are also used for parts of soda fountains, cream separators, milking machines, surgical apparatus, galvanometer's parts, player pianos, etc., where a tensile strength of over 8000 lb. per sq. in. is not essential and where resistance to corrosion is of importance. They are not affected by water, weak acid, or alkaline solutions, and when free from lead, are extensively used for food-container parts.

GROUP C — LEAD ALLOYS

11 *Typical Alloys.* The following four compositions are those of lead alloys widely used in the production of die castings:

	Lead Per Cent	Tin Per Cent	Antimony Per Cent
No. 1	83	0	17
No. 2	90	0	10
No. 3	80	10	10
No. 4	80	5	15

General Properties:

Weight per cu. in. .... depends on lead content
Maximum fusing point .... 600 deg. fahr.

Casting Limits:

Maximum weight for casting .... 15 lb.
Minimum wall thickness .... $\frac{1}{32}$ in.
Variation from drawing dimensions per inch of diameter or length .... 0.001 in.

Cast Threads, minimum number:
External .... 24 per inch
Internal .... depends on conditions, often cast

Cast Holes: minimum diameter......................0.031 in.
(Depends on depth and thickness of casting)
Draft: Cores, 0.0005 in. per inch of length and diameter. Side walls, 0.001 in. per inch.

12 Alloy No. 1 is generally known as C. T. (Coffin Trimming) metal, due to its extensive use in the manufacture of coffin trimmings. This alloy is also a good bearing metal for light duty and is employed for thrust washers and camshaft bearings on light internal-combustion engines. No. 2 is somewhat softer and more ductile than No. 1. No. 3 is much used for light bearing duty, being somewhat tougher and stronger than Nos. 1 and 2. No. 4 is somewhat harder than No. 3 but less ductile. Many similar alloys may be compounded, all of which may be die-cast readily.

13 *Applications.* Lead alloys may be used where a metal of non-corrosive properties is desired and where a tensile strength of not over 8000 lb. per sq. in. will suffice. They are used extensively for fire-extinguisher parts, low-pressure bearings, oranmental metalware, and many parts that come in contact with corrosive chemicals. They should not be used for parts that may come in contact with foods or that may be handled often in service, since the poisonous properties of lead and lead alloys are well known.

14 The main advantage of these alloys lies in their comparatively low cost, but their high specific gravity must be considered, some lead alloys having a specific gravity double that of the zinc alloys.

15 During the war lead alloys were used for all hand-grenade fuse parts and many millions of these parts were made. Lead-alloy die castings were also used for thermite grenades, offensive grenades, trench-mortar fuse plugs and many other parts where a non-corrosiveness was an essential requirement.

GROUP D — ALUMINUM ALLOYS

16 *Typical Alloy.* A typical aluminum alloy for use in die-casting work has the following composition:

Aluminum, 92 per cent; Copper, 8 per cent.

Properties:

Colo ....Silver white
Weight per cu. in.......................................0.115 lb.
Melting point...................................1150 deg. fahr.
Tensile strength.............................21,000 lb. per sq. in.
Elongation...1.5 per cent
Hardness number (Brinell)..................................60.5

Casting Limits:

Maximum weight for castings . . . . 5 lb.
Minimum wall thickness . . . . $\frac{1}{16}$ in.
Variation from drawing dimensions per inch of diameter or length . . . . 0.0025 in.
Cast Threads, minimum number:
External . . . . 20 per inch.
Threads are cast oversize 0.01 in. to be chased to size.
Internal threads rarely cast.
Cast Holes: Minimum diameter 0.093 in. and not deeper than 1 in. Larger cores may be cast much deeper; smaller holes may be spotted to facilitate drilling
Draft: Cores, 0.015 in. per inch of diameter or length. Side walls, 0.005 in. Cores of less than $\frac{1}{4}$ in. diameter to have 0.005 in. draft per in. of length and diameter.

17 The composition described above is well known in the arts as No. 12 alloy and is used very extensively for automobile and aeroplane parts. By varying the copper content harder or softer alloys may be obtained, all of which may be die-cast successfully.

18 *Applications.* Aluminum die castings find wide employment in the manufacture of parts of automobiles, such as spark and throttle control sets, magneto parts, battery ignition and lighting systems, speedometers, etc. They are also used for parts of vacuum sweepers, phonographs, milking machines, vending machines, etc.

BRASS AND BRONZE DIE CASTINGS

19 Die castings made from various types of brasses and bronzes were put on the market as early as 1910, but have never been successful commercially. At the present time there is only one die-casting manufacturer producing brass die castings in any appreciable quantity.

20 It is a comparatively simple matter to produce a small quantity of sample brass die castings, but no material has yet been found for die-making purposes, which will withstand the continuous action of molten brass and at the same time retain its shape, surface and size. The die casting of brass and bronze must be considered as in the experimental stage at the present time, with little or no immediate prospect of the solution of the problem.

DEVELOPMENTS DUE TO THE WAR

21 The most important development in the art of die casting during the war was the perfection of the process of die casting alumi-

num and its alloys. Prior to 1914 all die castings were made from zinc, tin, or lead alloys. The die casting of aluminum was in the experimental stage and no aluminum die castings had been produced commercially at that time. In making this statement the writer is aware of the fact that many aluminum pistons and similar parts had been cast in metallic molds. These castings, however, were poured by gravity and were comparatively crude. No attempt was made to cast small holes, slots or threads, or to work to any such degree of accuracy as that found in modern pressure die castings.

22 The first commercial aluminum die castings appeared on the market in 1914 and during the war a suitable steel was developed for making the dies for this process that would withstand the action of molten aluminum without cracking, a problem the solution of which was essential to the development of the industry.

23 The part that this development played in the winning of the war will be readily appreciated when it is stated that at the cessation of hostilities there were being produced about one million aluminum die castings daily in this country for parts of gas masks, machine guns, aeroplanes, motor trucks, motor ambulances, surgical instruments, canteens, field binoculars, and many other appliances of war.

COMPARATIVE COST OF DIE CASTING

24 The cost of die castings cannot be computed on the pound basis since it depends on the design of the piece, the number and position of the cores, the quantity to be produced and certain other factors. For comparative purposes it may be stated that at the present time tin-alloy castings are the highest in cost, being followed by those of aluminum alloy, zinc alloy and lead alloy in the order named.

25 In considering the use of die castings it is well to bear in mind that on a pound basis die castings are far more expensive than iron sand castings where the machining cost is not considered. As the zinc alloys whose properties are similar to cast iron, cost from $200 to $275 per ton in ingot form, it is apparent that the substitution of a die casting for an iron casting can only be considered when the machining cost is sufficient to compensate for the difference in cost of the raw materials.

No. 1743c

ALUMINUM CASTINGS

By Zay Jeffries,[1] Cleveland, Ohio
Non-Member

In this paper the author first gives particulars of the various aluminum alloys used for casting purposes. He then deals with the metallography and physical properties of these alloys, following which are notes on maximum and minimum weights, tolerances on dimensions and machining data. Later sections deal respectively with heat treatment, aging, effect of thickness of section and rate of chill on physical properties of aluminum alloys, fatigue resistance, special places where the use of aluminum castings is indicated by reason of their lightness and ease of machining, and with the selection of alloys for special purposes.

THE aluminum-castings industry can hardly be said to be more than fifteen years old. It is true that castings were made prior to 1905, but in such small quantities as to be insignificant in comparison with the present magnitude of the industry. The annual production of aluminum at the present time is about three hundred million pounds. The percentage of this used in castings is not definitely known, but it is estimated at about one-half the world's production. While the automotive industry uses the bulk of the aluminum castings, there is hardly an industry of any importance that does not use some. These castings find their way into many homes in the form of kitchen utensils, vacuum-sweeper parts, etc. The use of aluminum castings is increasing rapidly in the electrical industry.

2 The aluminum-castings industry was developed largely in brass foundries and even today it will be found in many instances that companies manufacturing aluminum castings also manufacture brass castings. Rule-of-thumb methods have been followed in developing the aluminum-casting industry and even as recently as 1917 the quality of castings was so uncertain that aeroplane manufacturers and aviators had to be reassured of the reliability of the

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Presented at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers.

metal for aeroplane construction. In May 1917, at the request of Messrs. Manley and Cardullo of the Curtiss Aëroplane Company, an investigation was made of the properties of aluminum castings then being used for aeroplane crankcases and other parts. It was found that the quality was very poor, the physical properties being neither uniform nor sufficiently high.

3 Immediate steps were taken to introduce technical control in the foundries to insure against wrong metal compositions and abuse of metal during the melting operations, and also to insure the maintenance of the proper pouring temperature. The results were very satisfactory, as is evidenced by the production record of Liberty engine castings in which a minimum tensile strength of 18,000 lb. per sq. in. was specified and the rejection for failure to pass specifications was practically nil. Even in such highly stressed parts as those in the Liberty engine, no failures of aluminum castings in service have come to the author's attention.

ALUMINUM ALLOYS FOR CASTINGS

4 The principal aluminum alloy used for castings in the United States consists of about 92 per cent aluminum and 8 per cent copper and is generally known as Aluminum Company of America's No. 12 alloy or S. A. E. specification No. 30. This alloy when cast in a ½-in. test bar in green sand and tested without machining off the skin should give an average tensile strength of about 20,000 lb. per sq. in. and an average elongation of about 1.5 per cent in 2 in. A modification of this alloy having somewhat better physical properties is now finding considerable favor in castings for the automotive industry. This alloy has an analysis of 7.5 per cent copper; 1.5 per cent zinc; 1.2 per cent iron, and the remainder aluminum. The tensile strength of this alloy will average about 21,000 lb. per sq. in. and the elongation will be somewhat greater than that of No. 12 alloy.

5 S. A. E. Specification No. 32 is another alloy of aluminum and copper which finds considerable use, especially in parts which need to be water-tight. This alloy contains from 11 to 13.5 per cent copper, the remainder being commercial aluminum.[1] Its tensile strength when the test bar is cast in green sand about ½ in.

[1] Commercial aluminum always contains iron and silicon in amounts of 0.25 per cent and upward according to the grade of ingot.

in diameter and broken without machining off the skin should show an average of 20,500 lb. per sq. in. and the elongation will be practically nil. This alloy is used for carburetors, pumps, manifolds, radiators, engine cylinders, etc.

6 Another aluminum-copper alloy containing about 5 per cent copper is used for cast automobile bodies. This alloy has an average tensile strength of about 18,000 lb. per sq. in. and an average elongation of 3.0 per cent.

7 For pistons cast in permanent molds an alloy containing 10 per cent copper, 1.25 per cent iron, 0.25 per cent magnesium is extensively used. This alloy gives a good wearing surface both for the piston against the cylinder wall and for the bearing of the wristpin in the bosses. The pistons and camshaft bearings for the Liberty Engine were made from this alloy, the former being cast in permanent molds.

8 While the aluminum-copper alloys have been used mostly in the United States, light alloys containing zinc have been used more extensively abroad. The most common alloy in use in England for sand castings contains 13.5 per cent zinc and 2.75 per cent copper, the remainder being commercial aluminum. The specific gravity of this alloy is 3.0 and its tensile strength when cast in sand should be over 25,000 lb. per sq. in. and the elongation should exceed 1 per cent. In England the test bars of this alloy are cast in chill molds and the minimum physical-property requirements are a tensile strength of 11 long tons (24,640 lb.) per sq. in. and an elongation of 4 per cent. This alloy is used in England as a substitute for our No. 12 alloy.

9 An alloy containing 10 per cent zinc and 2.5 per cent copper is put out in ingot form in England by the British Aluminum Company. This alloy should have a tensile strength of more than 22,000 lb. per sq. in. and an elongation of more than 2 per cent in 2 in.

10 Aluminum Manufactures, Inc., use an alloy containing 7 per cent zinc, 2.75 per cent copper and 1.5 per cent iron. This alloy has an average tensile strength of about 27,500 lb. per sq. in. when cast in green sand and an average elongation of about 4.5 per cent.

11 Zinc is added to aluminum up to 33 per cent. As the zinc increases up to this amount the tensile strength increases and the elongation decreases. The 33 per cent zinc alloy has been used considerably in the past but it is too brittle to find extensive employment in the industries in competition with more ductile alloys which also have lower specific gravities.

12 At the Westinghouse Electric and Manufacturing Company an alloy composed of 95 per cent aluminum and 5 per cent magnesium is used for starting-motor frames and other automobile work where extreme lightness and good machining qualities are essential. The specific gravity of this alloy is 2.47, the average tensile strength about 27,000 lb. per sq. in., and the elongation about 3 per cent.

13 At the United States Navy Yard, Washington, D. C., an alloy containing 2 per cent copper and 1 per cent manganese is used for all aluminum-casting work. It is used for parts for gun sights, optical instruments, loading-tray sights for naval mounts, mine-laboratory experimental work, parts for nitrating machinery for use in the manufacture of powder, etc. This alloy has a tensile strength of about 18,000 lb. per sq. in. and an elongation of about 8 per cent. Its proportional limit is very low, often not exceeding 2000 lb. per sq. in.

14 An alloy containing 98 per cent commercial aluminum and 2 per cent manganese is also used for certain aluminum castings which should resist the corrosive action of chemicals used in rubber molding.

METALLOGRAPHY OF ALUMINUM ALLOYS

15 When copper is added to aluminum the compound $CuAl_2$ is formed. This dissolves in solid aluminum up to about 4 per cent copper at 500 deg. cent. and the solubility decreases to less than 1 per cent at room temperature. Above about 4 per cent copper a eutectic forms between $CuAl_2$ and aluminum-$CuAl_2$ solid solution. The tendency for the $CuAl_2$ to separate out in the free state is very marked, that is, the solid solution seems to be easily divorced from it. The variable solubility of $CuAl_2$ with change in temperature makes it possible to change the properties of the aluminum-copper alloys by heat treatment.

16 Zinc forms the compound Zn_3Al_2 which is soluble in aluminum up to 40 per cent of zinc.

17 Magnesium forms Mg_4Al_3 which is soluble in solid aluminum up to about 13 per cent magnesium at the eutectic temperature (450 deg. cent.). The solubility of the compound decreases with the temperature to about 5 per cent magnesium at 300 deg. cent. and probably to still lower values at atmospheric temperatures. Under equilibrium conditions a eutectic is formed between the saturated solid solution and the compound Mg_4Al_3 when the magnesium content

exceeds 13 per cent. Under normal conditions of cooling in sand, the eutectic is formed as a network around the grains when the magnesium content is over about 6 per cent. Magnesium also combines with the silicon present in commercial aluminum, forming the compound Mg_2Si. This compound is very brittle and renders the metal containing it in excess brittle.

18 Iron, manganese, and nickel form $FeAl_3$, $MnAl_3$, and $NiAl_3$, respectively. These constituents are very slightly soluble in solid aluminum and separate out in the form of needles when more than, say, 0.5 per cent of any of the elements is present. The

FIG. 1 MICROGRAPH OF 10 PER CENT COPPER ALLOY SHOWING A NETWORK OF $CuAl_2$. MAGNIFICATION 250 DIAMETERS

$FeAl_3$ needles have a strengthening effect on most aluminum castings by virtue of the fact that the normal eutectic network is made less continuous. As the fracture in nearly all aluminum castings takes place at the eutectic network, the $FeAl_3$ needles make the path of rupture greater and hence increase the breaking load. By increasing the breaking load more deformation is forced upon the more ductile excess material, thus producing a higher elongation. Iron in amounts up to 1.5 to 2 per cent may therefore be quite beneficial in aluminum castings.

19 Figs. 1 to 5, inclusive, show micrographs of the various constituents described above.

Fig. 2 Micrograph of Alloy Containing 1.7 Per Cent Silicon and 4.5 Per Cent Magnesium, Showing the $Al-Mg_2Si$ Eutectic. Magnification 250 Diameters

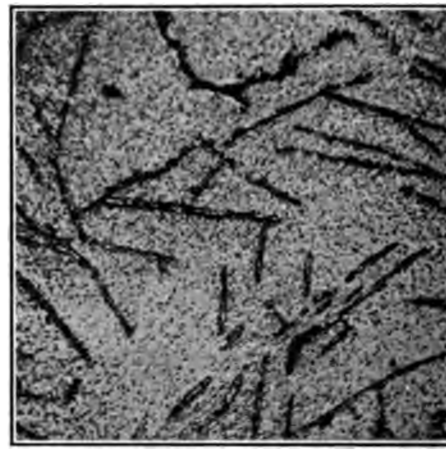

Fig. 3 Micrograph of Alloy Containing 3 Per Cent Iron, Showing Needles of $FeAl_3$. Magnification 250 Diameters

Fig. 4 Micrograph of Alloy Containing 1.35 Per Cent Nickel, Showing Needles of $NiAl_3$. Magnification 250 Diameters

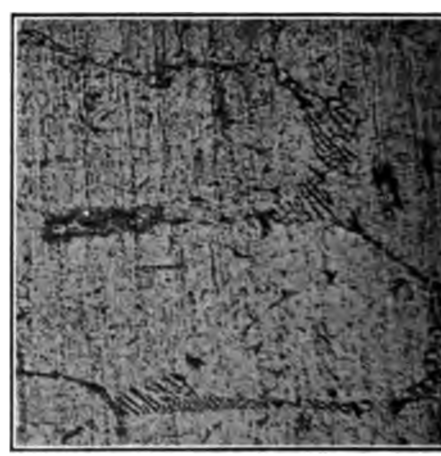

Fig. 5 Micrograph of Alloy Containing 0.5 Per Cent Manganese, Showing $MnAl_4$. Magnification 250 Diameters

PHYSICAL PROPERTIES OF ALUMINUM AND ITS ALLOYS

20 *Hardness.* The Brinell hardness of pure aluminum is about 25 and that of the hardest aluminum alloy is more than 120. Where hardness is the controlling factor, any value between 25 and 120 can be produced.

21 *Coefficient of Expansion.* The coefficient of expansion of pure aluminum is about 0.000025 per deg. cent. The coefficient of expansion of the alloys varies between 0.000022 and 0.000027. Alloys containing about 5 per cent magnesium have the higher coefficient of expansion and the high-copper alloys show the lower values.

22 *Heat Conductivity.* The heat conductivity of pure aluminum is about $0.50K$ or about 48 per cent of the conductivity of copper at ordinary temperatures, and varies but slightly with temperature. The heat conductivity of aluminum alloys is less than that of the pure metal, varying between 25 per cent and 48 per cent of the conductivity of copper, the general rule being that the greater the percentage of alloying elements, the lower the heat conductivity.

23 *Electrical Conductivity.* The electric resistivity of pure aluminum is about 2.94 microhms per cu. cm. The electric resistivity of the alloys is higher than that of pure aluminum, ranging up to 5.70 microhms per cu. cm.

24 *Melting Points.* The melting point of pure aluminum is 658 deg. cent. The lowest melting points of the useful alloys range around 580 deg. cent.

25 *Crystallization Shrinkage and Allowance for Pattern Shrinkage.* The crystallization shrinkage of pure aluminum is about 5 per cent by volume, and a further 5 per cent decrease in volume takes place on cooling from the melting point to room temperature. The crystallization shrinkage is somewhat less in the alloys. The pattern shrinkage allowed for all aluminum alloys is $\frac{5}{32}$ in. per ft.

26 *Specific Gravity.* The specific gravity of pure aluminum is about 2.7 and that of the alloys is not far from that calculated from the specific gravities of the various constituents. An alloy containing 95 per cent aluminum and 5 per cent magnesium has a specific gravity of about 2.47 and one containing 67 per cent aluminum and 33 per cent zinc a specific gravity of about 3.3. These represent minimum and maximum values for the usable aluminum-base alloys.

27 *Modulus of Elasticity.* The modulus of elasticity of aluminum is about 10,000,000 lb. per sq. in. Values as low as 8,500,000 and as high as 11,000,000 have been obtained.

GENERAL OBSERVATIONS ON ALUMINUM CASTINGS

28 *Classes.* Aluminum castings may be divided into three classes according to the method of manufacture, namely,

1 Sand Castings
2 Permanent-Mold Castings
3 Pressure Die Castings.

29 Sand castings are so named because they are cast in either green or dry sand. The method of molding is very similar to that used for other metals except that the gates, risers, and chills must be used to suit the particular properties of aluminum and its alloys. Such parts as crankcases for gas engines, oilpans, manifolds, steering-wheel spiders, transmission housings, differential carriers, etc., are usually cast in sand.

30 Permanent-mold castings are made in iron or other metal molds and consequently have a greater degree of chill than the sand castings. The physical properties of the various alloys vary in proportion to the degree of chill. Molds for the permanent-mold castings are provided with ample gates and risers for feeding, consequently the permanent-mold casting is non-porous as distinguished from the pressure die casting, which is always porous in the interior. As a general rule the permanent-mold process is limited to small parts with simple shapes.

31 Pressure die castings are considered in a paper on Die Castings presented at this session by Mr. Charles Pack.

32 *Maximum and Minimum Weight.* Aluminum sand castings have been regularly made weighing as high as 480 lb. each, but one of from 150 to 200 lb. is usually considered a large casting. The minimum weight may be taken as 1 oz., but neither this weight nor 480 lb. should be considered as an extreme limit.

33 In the permanent-mold process the maximum weight is about 20 lb. and the minimum weight about 1 oz. Permanent-mold castings have been made weighing as high as 150 lb. but these large castings should be considered as special.

34 *Section Thickness.* On gas-engine oilpans the walls should be $\frac{3}{16}$ in. or more in thickness and on crankcases a wall thickness of $\frac{7}{32}$ in. or greater is common practice. Fillets should have large radii, especially where heavy sections join thin sections. The best thickness on pipes, manifolds, etc., is $\frac{1}{8}$ to $\frac{5}{32}$ in. The minimum thickness on permanent-mold castings is $\frac{3}{32}$ to $\frac{1}{8}$ in.

35 *Finish.* On large sand castings $\frac{1}{8}$ in. should be allowed for finish and on bench work $\frac{1}{16}$ in. A finish of $\frac{1}{32}$ in. should be allowed for disk grind.

36 The finish on permanent-mold castings up to 3 in. in diameter, for example, pistons, should be about 0.075 in. on the diameter, and the finish on the piston head should be 0.045 in. The finish on permanent-mold pistons over 3 in. in diameter should be about 0.093 in.

37 *Tolerances on Dimensions and Weights.* In sand castings a tolerance of $\frac{1}{32}$ in. in thickness should be allowed and in permanent-mold castings the tolerances are plus or minus 0.010 in. The weight tolerances on sand castings using metal-pattern equipment are plus or minus 3 per cent and with wood patterns plus or minus 5 per cent. The weight tolerances on permanent-mold castings are plus or minus 2 per cent.

38 *Machining.* For roughing cuts speeds of 500 to 700 ft. per min. and a feed of $\frac{1}{16}$ in. to $\frac{5}{32}$ in. are recommended, according to the nature of the work. No lubrication is recommended for roughing. For extra fine finish a speed of about 125 ft. per min. is recommended, but speeds up to 600 ft. can be used and satisfactory finish obtained. For turning, the finishing feeds should be from 0.01 in. to 0.125 in., and for planing up to 0.25 in. A good lubrication mixture for finishing is 70 parts kerosene and 30 parts lard oil.

39 Aluminum alloys take a fine finish by grinding. Good results have been obtained using a No. 40 grain crystolon wheel and a mixture of 10 per cent lard oil and 90 per cent water for lubrication. The grains of the crystolon wheel even when dislodged do not stick in the metal.

HEAT TREATMENT OF ALUMINUM-ALLOY CASTINGS

40 Experiments have indicated that the tensile properties of aluminum alloys are considerably improved by heat treatment, but these developments are so new and have gained so little headway in production that the subject will not be treated in this paper.

41 Aluminum castings, especially those containing copper, may have their volumes permanently changed by heat treatment. Also the hardness may be changed. These facts are taken into consideration in connection with the heat treatment of aluminum-alloy parts such as pistons which will have to operate at an elevated temperature. The pistons are given a heat treatment which will cause permanent

growth to such an extent that no additional change of volume will take place during normal running.

AGING

42 Nearly all aluminum-alloy castings change in physical properties at room temperature after casting. This change in properties is called "aging" and is caused by the same internal changes which change the properties of the alloys by heat treatment. The general effect of aging is to increase tensile strength and decrease elongation. The elongation may be reduced by aging from 2 per cent to 1 per cent in the 8 per cent copper alloy, or from 8 per cent

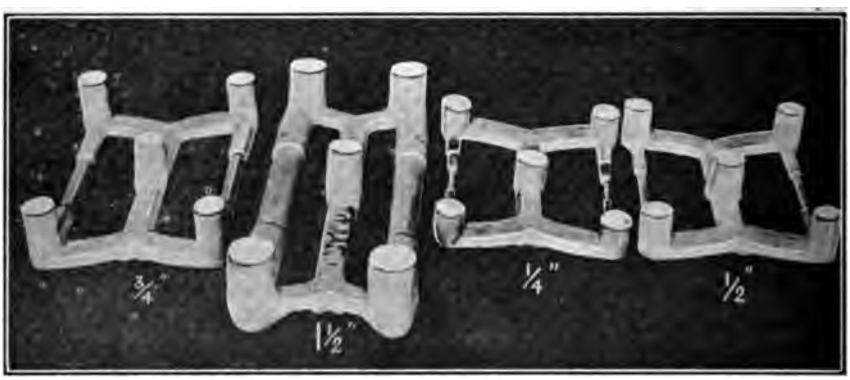

FIG. 6 METHOD OF CASTING ROUND TEST BARS

to about 5 per cent in one of the more ductile zinc alloys. This aging should cause no worry to engineers because, although it was not suspected in the past, the alloys which have been used for years have aged without harmful effects.

EFFECT OF THICKNESS OF SECTION ON PHYSICAL PROPERTIES OF ALUMINUM ALLOYS

43 In general, the tensile strength and elongation of the aluminum alloys decrease as the thickness of the section increases. In the case of round test bars cast as shown in Fig. 6, the decrease in tensile strength in an alloy containing 8 per cent copper and 92 per cent aluminum is from approximately 25,000 lb. per sq. in. and 3 per cent elongation in the ¼-in. test bar, to 14,000 lb. per sq. in. and less than

1 per cent elongation in the 1½-in. test bar, with proportional intermediate values for the ½-in. and ¾-in. bars.

44 It appears that this alloy on account of its low elongation does not show up so well when tested in a large piece as would be indicated from the results obtained from small test bars machined from the larger pieces. For example, small test bars cut from the 1½-in. bars averaged about 19,000 lb. per sq. in. and the elongation was more than 1 per cent, whereas the large bars tested 14,000 lb. per sq. in. and less than 1 per cent elongation. The tensile strength of one of the zinc alloys described earlier decreases at an approxi-

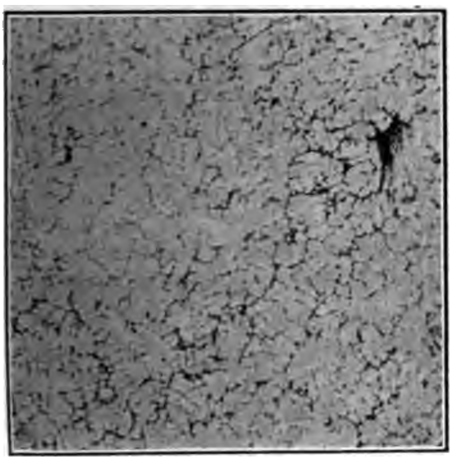

FIG. 7 GRAIN SIZE IN A ROUND TEST BAR OF 8 PER CENT COPPER ALLOY, ¼ IN. IN DIAMETER, CAST IN GREEN SAND. MAGNIFICATION 50 DIAMETERS

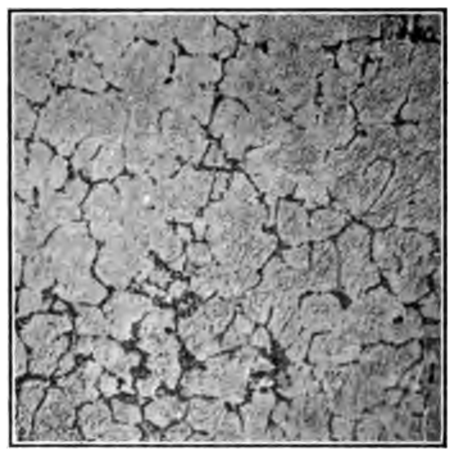

FIG. 8 GRAIN SIZE IN A ROUND TEST BAR OF 8 PER CENT COPPER ALLOY 1½ IN. IN DIAMETER, CAST IN GREEN SAND. MAGNIFICATION 50 DIAMETERS

mately linear rate from about 30,500 lb. per sq. in. in a ¼-in. test bar to 19,500 lb. in a 1½-in. test bar. Small bars cut from the 1½-in. bar tested about the same as the 1½-in. bar itself. The elongation of the zinc alloy decreases also with increase in thickness of section. It is thought that the greater ductility of this alloy is responsible for the similarity of results between the small test bars cut from the large bars and the large bars themselves.

45 The lower strength and ductility of the large sections are due to a combination of variation in grain size and soundness of the casting. The smaller sections are as a rule better fed during solidification and since they solidify more rapidly than the thicker sections,

they are also finer-grained. Figs. 7 and 8 show the variation in grain size in the 8 per cent copper alloy when cast in $\frac{1}{4}$-in. and $1\frac{1}{8}$-in. bars.

EFFECT OF RATE OF CHILL ON THE PHYSICAL PROPERTIES OF ALUMINUM ALLOYS

46 As a general rule, the more quickly aluminum alloys solidify the smaller the grain size and the higher the tensile strength and elongation. No. 12 alloy, for example, when cast in large sections in sand may have a tensile strength as low as 14,000 lb. per sq. in. and an elongation of less than 1 per cent. When cast in small sections in a chill mold its tensile strength may be 31,000 lb. per sq. in. and its elongation as high as 6 per cent. Figs. 9, 10, 11, and 12 show the effect of degree of chill on the structure of an alloy containing 92 per cent aluminum and 8 per cent copper.

FATIGUE RESISTANCE OF ALUMINUM ALLOYS

47 In sand-cast bars of No. 12 alloy the fatigue resistance as measured on the White-Souther machine is about 14,000 lb. per sq. in. maximum stress for 500,000 reversals and about 8500 lb. per sq. in. maximum stress for 16,000,000 reversals. The proportional limit of this alloy is about 5000 lb. per sq. in. The stress-strain curve departs very gradually from a straight line after the proportional limit is exceeded, as is shown in Fig. 13. It is believed that the proportional limit of the aluminum alloys does not represent a breakdown of the main metallographic constituent, as in a single-component metal like wrought iron, but that it simply breaks some of the unfavorably situated, brittle aluminum compounds and thus causes slight permanent set. The fatigue life of the aluminum alloys is quite unusual, inasmuch as the safe limit seems to be equal to or higher than the proportional limit as measured by the extensometer. Chill-cast alloys are even better than the sand-cast alloys for fatigue resistance.

48 In a very comprehensive paper entitled Aluminum Alloys for Aeroplane Engines, by Prof. F. C. Lea, published by the Royal Aeronautical Society in 1919, fatigue tests on various aluminum alloys are reported, together with stress-strain diagrams. These tests show a minimum fatigue range in the Wöhler test of 12,000 lb. per sq. in. (− 6000 to + 6000) for 12,000,000 reversals on an alloy containing 12 per cent copper. The proportional limit of this alloy

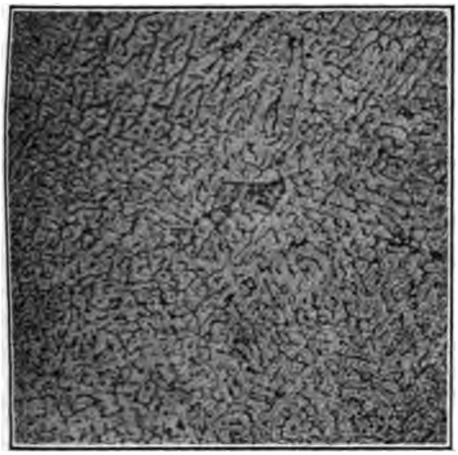

Fig. 9 Micrograph of Aluminum Alloy Containing 8 Per Cent Copper, Showing Grain Size Produced by Casting Cylinder 1¼ in. in Diameter in Iron Mold. Magnification 50 Diameters

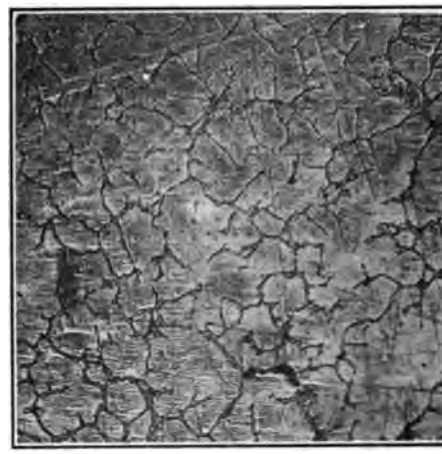

Fig. 10 Micrograph of Aluminum Alloy Containing 8 Per Cent Copper, Showing Grain Size Produced by Casting Cylinder 1¼ in. in Diameter in Green Sand. Magnification 50 Diameters

Fig. 11 Specimen Shown in Fig. 9 but at Higher Magnification to Show Structure of Network Surrounding the Grains. Magnification 250 Diameters

Fig. 12 Specimen Shown in Fig. 10. Higher Magnification Brings Out the Structure of the Network Around the Grains. Magnification 250 Diameters

was 5000 lb. per sq. in., although the amount of permanent set up to 6000 lb. was very slight.

GENERAL CONSIDERATIONS REGARDING USE OF ALUMINUM CASTINGS

49 As an engineering metal aluminum is in a class by itself because of its low specific gravity. Its use is determined by many factors, among which are the following:

a Where lightness is of prime importance, aluminum in

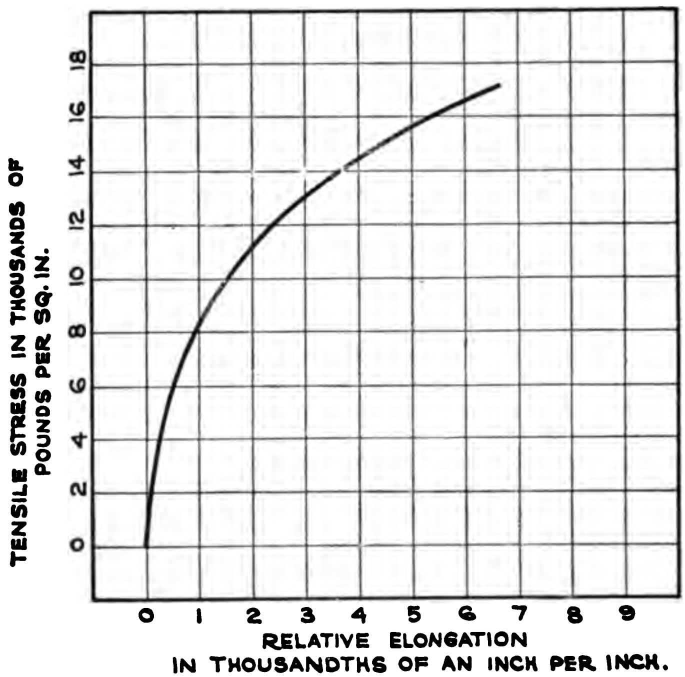

FIG. 13 STRESS-STRAIN DIAGRAM OF ALLOY CONTAINING 92 PER CENT ALUMINUM AND 8 PER CENT COPPER

sheet form has about seven times the stiffness of steel for equal weights. For a given rigidity in any metal structure aluminum will weigh less than any other metal used in engineering. For such parts, therefore, as aeroplane engine crankcases, camshaft housings, oilpans, etc., where lightness and rigidity are of prime importance, aluminum alloys find extensive use.

b In the manufacture of gas-engine crankcases it is found that the aluminum castings can be machined about three times as rapidly as cast-iron parts, and consequently the first cost of plant installation and labor cost of machining is greatly decreased by the substitution of aluminum for cast iron The lightness of the crankcase is the chief advantage because the weight of the engine per horsepower is reduced.

c Where the lightness of reciprocating parts like pistons. is of great moment, such as in high-speed gas engines, the use of aluminum is very beneficial. The aluminum piston also has the advantage of high heat conductivity, which is sometimes of greater importance in a gas engine than reduction in weight.

d For reducing unsprung weight in motor vehicles, aluminum is particularly valuable. This includes such parts as differential carriers, rear-axle housings, brake shoes, hub caps, wheels, etc.

e Aluminum alloys, especially with moderate pressures, function nicely as bearings against hardened steel. As an example, the Liberty Engine camshaft was run in aluminum bearings, as were also the rocker arms. Aluminum alloys, however, do not bear well against soft steel.

f For equal volumes aluminum is cheaper than brass or bronze. Aluminum alloys find extensive use as a substitute for brass and bronze castings where freedom from atmospheric corrosion is of prime importance and where the strength requirements are not too severe.

SELECTION OF ALLOY

50 For aluminum castings not highly stressed, No. 12 alloy should be used. Where freedom from leaks is the main requirement, the 12 per cent copper alloy is good and the more-ductile zinc alloy should be used for highly stressed parts. Present-day engineering materials are required to stand abuse rather than normal use. Ductility is essential if an alloy is to stand abuse, and this is the main reason for the use of the aluminum alloy containing 2.75 per cent copper, 1.5 per cent iron and 7 per cent zinc.

No. 1743*d*

STEEL CASTINGS

By John H. Hall,[1] High Bridge, N. J.
Non-Member

This paper deals briefly with the physical properties of steel castings and their improvement through annealing and heat treatment, cost of castings, and the classes of work for which steel castings should be specified. Castings are now made, it is stated, with physical properties nearly twice as good as those of the ordinary commercial product, and where the service demands unusual strength, toughness and resistance to wear and fatigue, their use is urged even though they machine somewhat less freely.

NOTWITHSTANDING the fact that the art of steel founding had been brought to a high state of perfection before the war as regards soundness of castings produced and their accuracy to dimensions, the general character of the output of a steel foundry, as first-rate foundrymen will agree, has been limited in great measure, not by what it is able to make, but by the demands of the customer. This has been, and unfortunately still is, due to the fact that the buyer of castings inclines too much to the viewpoint of the efficiency engineer in the machine shop, and is willing to subordinate everything but actual soundness to ease and speed of machining — in fact, he generally insists that the castings be made with an eye first of all to the latter requirement. This attitude, it may be said, is justified in part by the fact that in meeting keen competition, a few minutes or even seconds of extra maching time per casting mean the difference between profit and loss on a job. Moreover, it has been difficult in past years to procure anything better than an average steel casting, which would not show especially good physical properties under test.

2 At the time the war broke out, however, the electric furnace was coming rapidly into use in steel foundries, and the maker

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of small castings especially was beginning to find that he could produce a much better article than he had previously thought possible. The war emergency hastened the introduction of the electric furnace to an enormous extent, so that progress has been very rapid in the last five years. It also made it imperative that steel castings be substituted for forgings in thousands of cases, which had two very marked effects upon the steel-founding industry, namely:

a The foundries had to learn, and did learn, to produce successfully castings of an intricacy that in former days would have challenged the skill of all but the very best shops

b The absolute necessity that the castings should possess physical properties that compared at least fairly well with those of the forgings previously used, stimulated the efforts of the foundrymen in this direction also, and resulted in many cases in the production of work that exhibited better physical properties than had previously been thought possible in castings.

3 The more observing users of castings, having learned during the war that it is possible to produce a far better steel casting than had formerly been supplied, have in some cases profited by their recently acquired knowledge, and are demanding and securing the better product. The testimony of some of the very best makers of small steel castings, however, is to the effect that the buyers of their wares still adhere to their old practice of requiring first of all a freely machining casting and will not tolerate improvement in the quality of the steel if it interferes in the slightest with ease of machining. This attitude is unfortunate and really short-sighted, as in the long run it can have but one result. Those who maintain it are sure, sooner or later, to suffer from the competition of concerns making and machining their own castings, and willing to sacrifice a little to better quality. It is true at the present time that the very best steel castings made are used for the most part by those who make them, and for certain classes of product that are subjected to very severe service. Castings are now being made that show all-around physical properties nearly twice as good as those of the ordinary commercial steel casting, but the average machine-shop proprietor absolutely cannot be persuaded to use them, simply because they do not machine quite so freely as those he has been accustomed to in the past.

RANGE OF WEIGHT OF STEEL CASTINGS PRACTICALLY UNLIMITED

4 There are practically no limitations to the maximum and minimum weights of steel castings that can be produced commercially. The foundryman using a small bessemer converter, a crucible furnace, or a small electric furnace can and does make castings with sections as light as $\frac{3}{8}$ in., and in some cases even $\frac{1}{4}$ in., provided their size and intricacy is not such as to make this impossible. On the other hand, the open-hearth furnace will pour castings as heavy as are ever called for, some poured in the last few years weighing 150 tons or even more, and with maximum sections of several feet. In fact, the steel foundryman is not so much concerned with the actual thickness of any particular section as with the combination of thick and thin sections in one and the same casting. When absolutely necessary, he can execute a casting of great intricacy in which thick and thin sections occur, often in exasperating combinations; but it is an undoubted fact that if the designer of castings would take counsel more often with the foundryman, he would obtain advice that would help him to secure a far better casting, and one that would more successfully meet the service it is called upon to stand.

PHYSICAL PROPERTIES OF STEEL CASTINGS

5 The physical properties of the average commercial steel casting depend largely upon whether or not the casting has been annealed. Perhaps 50 per cent of the steel castings made in this country are shipped without annealing. This statement is not made on actual knowledge and is consequently subject to criticism as too high or too low, depending upon whether the critic is or is not a steel foundryman. The American Society for Testing Materials specifications for steel castings call for two classes, A and B. Class A castings "need not be annealed unless so specified." The carbon of these castings is specified not over 0.30 per cent, and the phosphorus not over 0.06 per cent (at present 0.07 per cent is allowed). Such castings will test about as follows:

Tensile strength, lb. per sq. in.	60,000
Yield point, lb. per sq. in.	25,000
Elongation in 2 in., per cent	10 to 15
Reduction of area, per cent	15 to 25

6 The A.S.T.M. specifications do not call for tensile or bending tests on Class A castings. These specifications are intended

to cover only such castings as are not subject to very severe service, and in which the chief characteristics desired are soundness and comparatively easy machining.

7 The writer is aware that some engineers are willing to accept unannealed steel castings for quite severe service, even for parts of locomotives and cars. The properties of the metal, as shown by the ordinary static tensile and bending tests, are considered good enough to justify the use of the unannealed steel. Of course the only steel so accepted is quite low in carbon, and accordingly fairly tough, at least in static tests.

8 From a fairly long acquaintance with cast steel, however, the writer is forced to state that in his opinion this policy cannot be justified. Our knowledge of endurance or "fatigue" tests teaches us that a coarsely crystallized metal such as unannealed cast steel cannot have a really high resistance to severe alternating stresses, and nothing like as high as the same metal will have after the coarse structure has been broken up by annealing and replaced by a finely crystalline condition.

9 It is when we come to consider resistance to sudden heavy shocks, however, that the danger of using unannealed cast steel for severe service strikes us most forcibly. Tests made by the writer have shown that under heavy shock cast steel even as low in carbon as 0.10 per cent breaks almost without bending, and with very little absorption of energy. In other words, the metal is truly brittle under impact test. After ordinary annealing the resistance to shock is greater, and the test piece bends through a greater angle before rupture. If the steel under test has been heat-treated by rapidly cooling and reheating, the shock resistance and angle of bend are increased to an even greater extent.

10 These tests were made on a Frémont testing machine; but the writer has made many tests on full-sized castings that amply justify him in his opinion that the use of unannealed cast steel for service calling for ability to resist sudden heavy shocks cannot be justified unless the sections of the piece can be made very heavy for the strength desired — in other words, unless weight per unit of strength can be disregarded.

11 The A.S.T.M. Class B castings must not exceed 0.05 per cent phosphorus or sulphur. (At present this has been temporarily increased to 0.06 per cent.) No other chemical specifications are given, the analysis other than phosphorus and sulphur being left to the judgment of the foundryman. Annealing, however, is

specified. These castings are divided into three grades (hard, medium, and soft) which must give physical properties as follows:

	Hard	Medium	Soft
Tensile strength, lb. per sq. in.	80,000	70,000	60,000
Yield point, lb. per sq. in. (=0.45 T. S.)	36,000	31,500	27,000
Elongation in 2 in., per cent	15	18	22
Reduction of area, per cent	20	25	30
Cold bend around 1-in. pin, deg.	..	90	120

12 Reliable foundries can meet these specifications with practically no trouble beyond the expense necessarily due to meeting the requirement that castings be kept together by heats as poured, and the cost of making the test specimens and carrying out the tests. Most specifications as now written call for so many test pieces that the cost of producing the castings is considerably increased. This is particularly true when it is specified that a test shall be attached to each and every casting over a certain weight. The writer has known cases in which this specification was literally insisted upon where as many as fifty almost identical tests were pulled from a heat of open-hearth steel. The A.S.T.M. specifies a test attached to every casting weighing over 500 lb. unless the sections are such that a satisfactory test piece cannot be attached to it. In that case, and for castings under 500 lb., two test bars may be cast to represent a heat, or "in the case of small or unimportant castings, a test to destruction on three castings from a lot" may be used. This test shall show the material to be "ductile, free from injurious defects, and suitable for the purpose intended." This provision for testing small castings to destruction in place of using test bars is an excellent one and should be used more than it is.

ANNEALING AND HEAT TREATMENT

13 In order to meet specifications similar to those of the A.S.T.M., Class B, annealing is essential. The annealing practice followed is almost invariably such as to give the castings the greatest possible softness and ease of machining, and no heat treatment as the term is usually interpreted is needed to meet these specifications if the steel is well made in the first place. Castings annealed in this manner should machine very freely or fairly freely, depending upon whether they are of "soft" or "hard" grade, as prescribed. If the first cut is deep enough to get well under the

scale left by annealing and if the casting is free from adhering sand, it should give no trouble in machining.

14 Heat treatment as usually understood, that is, annealing the castings at a proper temperature, cooling more or less rapidly, and then heating a second time, produces castings with very much better physical properties than those of plain annealed cast steel, and at the same time produces a somewhat harder and much-finer-grained steel which has far better wearing qualities and much better resistance to fatigue than is possible in the case of castings which are simply annealed. In extreme cases, by using heat treatment and special analysis, it is possible to produce castings that will show as good tensile strength as "Class B Hard" of the A.S.T.M. specifications, and equal or exceed the elongation, reduction of area and bend of "Class B Soft" castings. The yield point especially is raised by heat treatment, and the wearing properties are improved. The resistance to fatigue is also greatly increased, and, most important of all, the power to resist sudden shock is enormously improved.

15 The cost of such heat treatment, if substituted for plain annealing when the castings are made, need not be higher than that of plain annealing, provided the foundry is properly equipped to carry out heat treatment. It must be confessed, however, that the average foundry today is not so equipped, and in many cases does not have on its staff a man who thoroughly understands heat treatment. In the writer's opinion this state of affairs is due as much to the buyer of castings as to the maker, because the buyer as a rule is not at all interested in the superiority of heat-treated or special-analysis steel castings as soon as he finds that such castings machine a little less freely than the common variety. The writer has had violent protests from customers because some of his castings gave a long, curling chip in the lathe, as a forging does, instead of the usual crumbly chip of the common steel casting. The fact that the physical properties of the castings were about as follows:

Tensile strength, lb. per sq. in.	85,000
Yield point, lb. per sq. in.	55,000
Elongation in 2 in., per cent	25
Reduction of area, per cent	50
Cold bend around 1-in. pin, deg	180

did not interest the customer in the least; and yet he had come to the foundry in which the writer was employed, and asked particularly for a casting to resist extremely severe fatigue and shock!

COST OF STEEL CASTINGS

16 The cost of steel castings, of course, varies with the size weight, and intricacy of the pattern to be executed. At the present time heavy castings such as are suitable for production in open-hearth foundries can be obtained for from 8 cents per lb. up, depending upon the difficulty of the job, while small and difficult castings command a price of about 12 cents per lb. up. No upper limit for price can very well be given, as very small or difficult castings, on which a piece price is often quoted, may go as high as $1.00 per lb. or even higher: but the foregoing figures are close enough for comparative purposes.

WHERE STEEL CASTINGS SHOULD BE SPECIFIED

17 Steel castings should be used for all cases where a forging is not suitable or is prohibitive in cost, and where the service to be rendered calls for strength and toughness combined. To enumerate the uses of steel castings would be virtually impossible. It is enough to say that cast iron is suitable for service in which the stresses are chiefly compressive, or where the stresses to be endured can be taken care of by increasing the size of the casting, regardless of the resulting weight. In fact, in many cases where cast iron is used, as in bedplates of engines and heavy machines, great weight for a given strength is even desirable. Where moderate strength and toughness, combined with fairly high resistance to wear and fatigue, are desired, a malleable-iron casting will give good satisfaction. When, however, the service demands all the strength, toughness, and resistance to wear and fatigue that are obtainable in a cast metal, a steel casting should be specified; and if the service is unusually severe, the engineer should use the best cast steel he can buy, and should not allow himself to be influenced in his choice by the fact that the metal does not machine as freely as inferior cast steel or malleable iron.

No. 1743e

GRAY-IRON CASTINGS

By Richard Moldenke, Watchung, N. J.
Member of the Society

After an introductory discussion of the carbon situation in cast iron, the author deals briefly with its chemical composition, giving in this connection a table of recommended compositions for many varied purposes. He then indicates the extent to which specialization has taken place in iron founding and enumerates the various classes of foundries and the lines of work to which they are best adapted. Dealing with recent tendencies in gray-iron foundry practice, he calls attention to the very high sulphur content now met with in castings — a result of the use of abnormal proportions of scrap during the war because of the high price of pig iron — and gives brief particulars of an economical duplexing process by means of which the sulphur in molten cupala metal may be brought down in an electric furnace from, say, 0.12 *per cent to* 0.05 *per cent and even lower.*

IN working out problems in machine design the mechanical engineer is constantly confronted with requirements for those parts he wishes to make of cast iron which make it necessary for him to be informed upon the latest developments of the foundry industry. Where to use a casting and what metal or alloy to make it from should be pretty well known by now, but what class of cast iron to use and what kind of a foundry to get it from may be matters open to doubt in these days of rapid changes.

2 Cast iron, with all its lack of homogeneity and consequent requirement of large factors of safety, is nevertheless of such variety in physical properties that few engineers are familiar with the many uses to which it can be put satisfactorily. It may run from glass-hard to dead-soft, from a condition of brittleness to one of considerable toughness and elasticity, appear silvery to black in fracture, may be made acid-proof and also yield to acid attack promptly; and, strange to say, may exhibit some of these diametrically opposite properties in the same casting. Therefore, even though he depends upon the foundryman to deliver him the kind of casting he needs, the engineer should have some knowledge of the underlying causes for the peculiar properties exhibited by cast iron when put into the form of castings.

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THE CARBON SITUATION IN CAST IRON

3 To understand the nature of the complex material known as cast iron it is necessary to go behind the returns of an "ultimate analysis," as it were, in order to get the "rational" analysis, or the grouping up of the several elemental constituents. Here it will be seen that the total carbon is present in two forms, part of it being a mechanically mixed crystalline graphite and the remainder being chemically combined. Then there are iron silicide, phosphide and sulphide, the last two compounds being distinctly crystalline when present in comparatively large proportions. The amount, distribution and physical condition of the graphite thrown out of solution at the moment of set are the more particular items of importance to the engineer in connection with the strength and machinability of the casting. While the total carbon of the metal is the same throughout the mass, the division into mechanically mixed graphite and combined carbon will be such that the highest proportion of the former will be found in the center of the section together with the least of the latter form, and vice versa at the periphery. Since the presence of combined carbon in increasing proportion means a corresponding increase in hardness, the metal at the surface of a casting will be harder than in the center. Further, the oxide of iron formed through contact of molten iron with a damp mold surface and the fluxes in the molding sand if not properly covered with adhering graphite, on interacting form an enamel which is very hard on the cutting tool, and hence this should be made to dig under the skin of the casting and lift the surface metal away as it cuts along under lubrication of the graphite present, and not be allowed to glide over the skin.

4 With the foregoing explanation of the carbon situation in cast iron, and remembering that graphite is in mechanical admixture only and consequently is an element of weakness by cutting up the continuity of the iron mass, it will be understood that fundamentally cast iron is a steel of given combined carbon content, with the other elements present in much higher percentages than one would expect in any steel. For instance, if the percentage analysis of the chips taken from the first eighth inch below the surface of a flat plate gave silicon 2.00, manganese 0.70, phosphorus 0.45 and sulphur 0.12; with graphite 3.20 and combined carbon 0.30; then that part of the plate was for practical purposes a 30 carbon steel containing much graphite, double the usual sulphur and ten times as much phosphorus as a steel ought ordinarily to contain. The

analysis for the carbons is advisedly given as representing the first eighth inch of the flat plate and not the whole of the plate, as would be the case for the other items of the analysis. The fact that the proportions of the two forms of carbon vary from surface to center conclusively shows that while the total carbon of a cast-iron analysis may be perfectly reliable, the items "graphite" and "combined carbon" in such an analysis are worthless unless the actual point from which the sample has been taken is indicated.

CONSIDERATIONS REGARDING CHEMICAL COMPOSITION OF CASTINGS

5 So far as the chemical composition of cast iron is concerned, it may be said that silicon as the chief determining factor of the condition of the carbon present may run from as low as 0.35 per cent in deeply chilled castings up to 3.25 per cent in the very lightest of hardware and novelty work. The absence of silicon allows the carbon to remain in chemical combination with the iron. As the percentage of silicon increases the iron loses its power to hold the carbon in combination and it is thrown out as graphite at the moment of set. The higher the silicon, the greater the graphite percentage. A factor affecting the silicon content to be employed is the rate of cooling of the metal when a mold is filled with molten iron. The thinner the section the greater the power of the iron to hold the carbon in chemical combination in spite of the silicon present. Hence high silicon is necessary to prevent thin sections from turning out hard, and since the foundryman must deliver his castings in condition to machine properly, he naturally regulates the silicon of his mixture to care for the thinnest section. He has the further difficulty of contraction with which to contend. Cast iron with the carbon all combined — or "white iron" — has a casting contraction of ¼ in. to the foot, whereas with the carbon all as graphite this contraction (erroneously called "shrinkage") is only half as much. If the foundryman were to aim for strong iron with a reasonable machinability for the heavier sections of the casting to be made (the blueprint showing that these sections go under the planer, etc.), making his mixtures accordingly with high steel-scrap percentages and comparatively low silicon, the thin rib and brackets of the casting would come out hard and brittle, if not actually white in fracture, and the greater contraction of these in cooling from the molten state would cause bad warping and internal strains. The mechanical engineer should therefore study the metallurgical side of the foundry requirements so that he may design

TABLE I RECOMMENDED ANALYSES FOR VARIOUS CLASSES OF CASTINGS

Castings	Light					Medium					Heavy				
	Si	Mn	S	P	T.C.	Si	Mn	S	P	T.C.	Si	Mn	S	P	T.C.
Acid-resisting	2.00	0.75	0.05*	0.20*	3.25*	1.50	1.25	0.05*	0.20*	3.25*	1.00	1.25	0.05*	0.20*	3.25*
Agricultural	2.50	0.60	0.06	0.75	3.75	2.25	0.70	0.08	0.70	3.50	2.00	0.80	0.10	0.60	3.25
Air Cylinders	1.90	0.70	0.08	0.50	3.40	1.50	0.80	0.09	0.40	3.25	1.00	0.90	0.10	0.30	3.00
Annealing Boxes						0.65	0.20	0.08	0.20	3.50					
Automobile Cylinders	2.25	0.65	0.08*	0.40*	3.25*	2.00	0.75	0.08*	0.40*	3.25*					
Balls for Grinding						0.75	0.50†	0.15*	0.40*	3.75†	0.50	0.50†	0.15*	0.40*	3.75†
Bedplates	2.00	0.70	0.08	0.60	3.75	1.75	0.75	0.10	0.50	3.50	1.50	0.80	0.12	0.40	3.25
Boiler Castings						2.00	0.80	0.06*	0.20*	3.50*					
Car Wheels											0.65	0.50	0.08	0.35	3.50
Chilled Castings												1.00	0.06*	0.20*	3.50*
Chills						1.00	0.50	0.05	0.20	3.00					
Crusher Jaws						1.00	1.00	0.04	0.20	3.50	0.80	1.25	0.06	0.20	3.25
Dies for Hammers						1.50	0.60	0.05	0.20	3.00	1.25	0.80	0.06	0.20	2.75
Dynamo Castings	2.50	0.50	0.05	0.75	3.75						2.15	0.50	0.06	0.50	3.25
Electrical Work	3.00	0.50	0.03	0.60	3.75	2.75	0.50	0.05	0.50	3.50					
Engine Frames						2.00	0.60	0.08	0.50	3.50	1.75	1.00	0.10	0.40	3.00
Fire Pots, Grates	2.25	0.60	0.05	0.20	3.50	2.00	0.80	0.06	0.20	3.25					
Fly Wheels	2.00	0.50	0.05	0.50	3.50	1.50	0.60	0.06	0.40	3.25	1.25	0.70	0.08	0.30	3.25
Friction Clutches	2.40	0.60	0.10	0.70	3.75	2.00	0.70	0.12	0.50	3.50					
Furnace Castings	2.40	0.60	0.05	0.60	3.75	2.15	0.80	0.06	0.50	3.50					
Gas-Engine Cylinders	2.00	0.70	0.08	0.40	3.25	1.50	0.80	0.09	0.30	3.00	1.25	0.90	0.10	0.20	2.85
Gears	2.25	0.60	0.08	0.70	3.75	2.00	0.80	0.09	0.60	3.50	1.50	1.00	0.10	0.50	3.25
Glass Molds, Pipe Balls						2.50	0.60	0.04	0.20	3.25					
Grate Bars						2.00	0.60	0.05	0.20	3.50					
Gun Iron						1.50	0.50	0.05	0.30	3.25	1.00	0.60	0.05	0.30	3.00
Hardware	2.50	0.70	0.08	0.80	3.75										
Heat-resistant Iron						2.00	0.80	0.06	0.20	3.25	1.50	1.00	0.06	0.20	3.00
Hydraulic Cylinders						1.50	0.80	0.05	0.40	3.25	1.00	1.00	0.08	0.20	2.85
Ingot Molds											1.25	0.80	0.06	0.20	3.75
Machinery Castings	2.50	0.60	0.08	0.70	3.75	2.00	0.80	0.09	0.60	3.50	1.50	1.00	0.10	0.50	3.25
Mine Wheels						0.90	1.00	0.10	0.20	3.00					
Ornamental Castings	2.75	0.60	0.06	0.90	3.75	2.25	0.70	0.08	0.80	3.50					
Pipe (Water)	2.25	0.60	0.06	0.80	3.75	2.00	0.80	0.08	0.70	3.50	1.50	1.00	0.10	0.60	3.50
Piston Rings	2.00	0.70	0.05	0.60	3.50	1.75	0.80	0.06	0.50	3.25					
Pulleys	2.40	0.50	0.05	0.70	3.75	2.15	0.60	0.07	0.60	3.50	1.90	0.70	0.09	0.50	3.25
Radiators	2.25	0.70	0.06	0.80	3.50										
Rolls (Chilled)							0.80	0.06	0.40	3.25		1.00	0.08	0.30	3.00
Soft Castings	2.60	0.50	0.06	0.60	3.75	2.40	0.60	0.08	0.50	3.50					
Soil Pipe	2.25	0.60	0.08	0.80	3.75	2.00	0.80	0.10	0.60	3.50					
Steam Cylinders	2.00	0.60	0.08	0.50	3.50	1.60	0.80	0.09	0.40	3.50	1.25	1.00	0.10	0.30	3.50
Stove Plate	2.50	0.50	0.06	1.00	3.75	2.25	0.60	0.08	0.80	3.50					
Valves	2.25	0.60	0.07	0.50	3.25	1.75	0.80	0.08	0.40	3.00	1.25	1.00	0.09	0.30	2.85
White Iron Castings						0.75*	0.20†	0.25*	0.75*	2.50†					

*Below. †Above.

his work with the least abrupt changes in section, avoid sharp corners and distribute his metal to the best advantage — in other words, give the casting a chance.

6 The total-carbon percentage has come to be a most important factor in the iron-castings industry. Of recent years the comparatively low-total-carbon iron made in the air furnace, and called "gun iron" from its use previous to the days of the steel cannon, has been paralleled by cupola metal in which great quantities of steel scrap are added to the mixture. Even in the air furnace the former long periods of refining to reduce the total carbon are shortened by adding all the way up to 30 per cent of steel scrap to the mixture of pig irons and home scrap. The product gets the erroneous name of "semi-steel" from those who are not informed on the properties of this grade of cast iron — for it is only a high-grade cast iron, with none of the properties of steel. Its importance to the engineer lies in the fact that the ultimate strength is nearly doubled by adding the steel scrap and that the comparatively cheap iron casting need not be replaced by the more expensive cast steel. The foundryman, however, has a problem before him in running these high-steel-scrap heats. Only the best of cupola and mold-gating practice will yield castings free from blowholes, cracks and serious internal shrinkages. The silicon percentage must be watched particularly, so that separation of graphite and combined carbon may leave the latter low enough for proper machinability. The effect of low total carbon (it may be brought down to 2.25 per cent, or just above the line of division into steel), if the combined carbon is kept normal, is to give an otherwise lower graphite percentage. Necessarily this graphite is finer in crystallization, and there being less of it than normal, the planes of separation in the solid metal are smaller and fewer, hence the greater strength. Moreover there is much less phosphorus present as the result of the low phosphorus of the steel added.

7 Manganese in the castings should not be allowed to run below 0.50 per cent nor over 1.00 per cent for normal machinable work. Phosphorus should be as low as possible, considering the pig irons that must be used, and for important and thick work had better run below 0.40 per cent. Very thin castings, such as art and architectural details in structures — which really correspond to the "stove plate" of the foundryman, require very nearly 1.00 per cent of phosphorus to come out in perfect detail. Table 1 gives recommended compositions of castings for many purposes. It is not wise for the engineer to specify the chemical composition of the castings

he requires, but if strength is important it should be specified, according to the standards of the American Society for Testing Materials.

SPECIALIZATION IN IRON FOUNDING

8 The foundry industry is pretty well divided up into specialized lines of work. If the engineer needs chilled castings he turns to the makers of chilled rolls. Many foundries, however, other than roll makers produce crusher castings, balls for grinding, and shoes and dies for stamp mills. Another important branch of foundry work is the making of cylinders. For the highest grade, foundries having air furnaces should be selected, particularly if they make a specialty of cylinders, piston rings and the like, and these will be found among the great foundry and machine works of the country building heavy machinery and engines (particularly of the marine type). Castings to withstand high pressures, such as those of air, steam and ammonia, are made by foundries specializing in air compressors, pumps, refrigeration machinery, and the general valve and fitting trade. For light parts to serve for ornamental purposes there are many stove shops to which recourse may be had. In general, however, where reciprocating parts or sections of intricate character and subject to severe service strains are required, it is safer to go to the large concerns making well and favorably known lines of product. The foundries connected with such establishments have the proper facilities and the trained men to undertake the work, and will be in position to advise improvements in design to give the metal a better chance for the purpose intended. Foundries of the class just mentioned usually have need of several mixtures of iron to supply the parts required for the line of machinery turned out. Hence they usually run a few charges of fairly low-silicon, high-steel-scrap metal for such castings as air-compressor cylinders, rolling-mill parts, and other work of high strength and density. The next charges may then care for ordinary machinery castings, and finally a few extra soft or high-silicon charges may care for pulleys and the very light-section product. Sometimes the process is reversed, all light, soft work being run first and the harder classes at the end of the heat.

9 On account of the lack of elasticity and its massiveness, cast iron is ideal for frames for machinery of all kinds, and as it can be locally chilled to give splendid wearing surfaces where needed, built-up structures of several materials can be avoided. The reduction in the total carbon through steel additions to the mixture

can be carried to a point when the chips taken from such a casting curl up almost as well as those from hard steel, and hence such metal serves very well where it is not essential that steel be used.

RECENT TENDENCIES IN GRAY-IRON FOUNDRY PRACTICE

10 A few words on the more recent tendencies in the production of gray-iron castings will not be amiss. The passing of the war has given us time to realize more fully the train of evils that have been its legacy to the foundry. A cataclysm such as this cannot but have disorganized everything it has touched, and so the foundry is manned today by operatives giving but part of the output of previous days, the character of the work is itself poorer, corrective measures are impossible of application, and hence the industry is far behind the present demand for castings. Until labor seeks work conditions will be no better, and prices are the rule which take into account all kinds of eventualities. So much for the operating end. On the technical side matters are just as bad. While there was a tremendous expansion of other industries due to war demands, the foundries simply pushed production to the utmost consistent with ability to get raw materials and men. Impossible prices for pig iron resulted in the use of so high a proportion of scrap in the mixtures that when the castings made during this period of stress eventually land into the scrap piles of the country — probably within the next twenty years — a serious problem of very high-sulphur and oxidized iron will confront the foundryman. Instances of 90 per cent scrap in the mixture were common and castings with over 0.20 per cent sulphur equally so.

11 The problem will have to be faced, and there are two possible solutions. The method heretofore employed in such a case is the increased use of pig iron in the mixtures. Before the war the common practice was to use not over 40 per cent scrap with 60 per cent pig. Now it is reversed. Consequently, instead of having the sulphur run about 0.08 per cent maximum in important work, it is more nearly 0.12 per cent. Ordinary work today touches 0.18 per cent without much comment. It is true that even a higher sulphur content will not militate against machinability provided the melting practice is of the very best, but the castings are more subject to danger from shock and casting strains.

12 The other method for correcting the high-sulphur tendency in daily work at present and for that to come, would be a desulphuri-

zation process. Until very recently no method of this kind was available, but fortunately the electric furnace has been drawn upon for this purpose and is doing the work with complete satisfaction. Instead of melting the charges from the cold state, as is the case in electric-steel developments, the regular cupola method is employed and the molten metal is transferred into an electric furnace having a basic lining. In this way the heavy current variations are avoided and by taking advantage of the comparatively cheap cupola-melting cost, that of the electric furnace for refining only is very materially reduced. It is a question whether more than a quarter of a cent is thus added to the cupola melting cost per pound of metal "duplexed," where current is cheap.

13 Molten cupola metal with possibly 0.12 per cent sulphur may thus be brought down to about 0.05 per cent and even lower. Moreover, the metal is highly superheated — cupola metal itself is intensely hot as compared with furnace iron — and thoroughly deoxidized. The castings made are therefore much better and sounder than the ordinary run, equaling charcoal-iron castings made in the air furnace. The first cost of the installation is high but soon pays for itself in the quality of work turned out. While developed during the war period it is none the less a logical sequence in the world advance of the art of making iron castings, and should be welcomed as a satisfactory way out of a very bad situation ahead.

14 In conclusion, the mechanical engineer is urged to turn his attention more specifically to foundry operation. The technical staffs of all the foundries of continental Europe, from manager down to assistants in the several production and testing departments, are all graduate engineers. The consequence is that castings are made strictly for the purpose intended and not merely to get by the machine shop safely, as is unfortunately so often the case here. Close coöperation between men trained in the science with those trained in the art of making castings can result only in good.

No. 1743f

BRASS AND BRONZE CASTINGS

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Following a short discussion regarding the terminology employed in designating alloys of copper, tin, zinc and lead, the author states certain precautions which an engineer should take when specifying an alloy for a given service, and then deals briefly with the various deoxodizers and fluxes used in the brass foundry. The latter half of the paper is devoted to an enumeration of the more important bronzes and copper alloys, their compositions, properties and uses being set forth in considerable detail.

THE terms "brass" and "bronze" are not as definite in their meaning as might be desired. The term bronze was formerly applied to an alloy of copper and zinc. Usage, however, has changed and it is now generally applied to an alloy of copper and tin, while the term brass is used to denote an alloy of copper and zinc. This change of nomenclature has led to a great deal of confusion. Even today we speak of crankpin, crosshead, or engine brasses when we have bronzes in mind; that is, a copper-tin alloy with comparatively little, if any, zinc in its composition. In general, it may now be said that without a modifying adjective brass is an alloy of copper containing a large amount of zinc; and bronze is an alloy of copper containing a relatively large amount of tin. In this country no general standard of terminology for these alloys has as yet been adopted; the British Institute of Metals has adopted one but theirs does not seem entirely satisfactory for our purpose. For the present, in this discussion, these alloys will be designated according to the prevailing commercial usage in this country; and other alloys will, as far as possible, be designated in the order of their preponderant constituents. Thus, the copper-tin-lead alloy is one containing, by weight, more copper than tin, and more tin than lead.

2 It is very generally understood that in designing machine parts which are to be made of cast materials a larger factor of safety

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is necessary than that required for rolled or forged material, although the compositions may be very similar or even identical. This in itself is not sufficient. The fact that the strength in any given casting varies in its different parts must also be taken into consideration, since it is impossible to make a casting with varying sections of uniform strength.

3 In general, the engineer should be cautioned against specifying alloys upon which he has only such knowledge as has been derived from laboratory tests alone, since it is impossible to determine, by ordinary experiment, all of the economic conditions which enter into the founding and tooling of a casting and the requirements of a subsequent satisfactory use of an alloy for any given service. To illustrate: Years ago the favorite Government composition known as gun metal (88 per cent copper, 10 tin and 2 zinc) was unreservedly specified for valves, steam connections and fittings. At the present time, however, no successful valve manufacturer would think of using this alloy for this purpose, although from purely laboratory results this and similiar high-tin alloys may appear to meet the requirements. The safe procedure is either to investigate and follow standard practice, or determine definitely why it should not be followed; or else to leave the composition and the guarantee of results to a responsible manufacturer and hold him accountable.

4 Aluminum should never be used in an alloy which is to be tight against leakage of either a gasoous or fluid substance. Aluminum has the peculiar property of forming a characteristic skin upon the surface of its alloys, due to its great affinity for oxygen. Appreciably less than one two-hundredths of one per cent of aluminum produces an appearance upon the surface which is very distinctly that of a brass or bronze casting. In fact, this surface film is often found in castings where the amount of aluminum present is so small that the chemist may be in doubt as to whether even a trace of it exists. In pouring a casting containing aluminum this surface film becomes entrained in the convecting currents as the molten metal fills the mold and often extends through the entire thickness of the wall of the metal and thus gives rise to porosity which causes leakage. For the foregoing reasons it is always desirable, when pouring, that the molds should fill with as little bubbling and splashing as possible. This applies to all alloys in general, but to those containing aluminum in particular. The "horn gate" or some form of gating by means of which the molten metal enters at the lowest point in the mold is very desirable.

5 In the alloying of copper and tin a low-fusing crystal forms known as the tin-copper eutectoid. This crystal is hard and brittle and in general should not be present in bronzes used for machine parts; and more particularly so if these parts are subjected to temperatures approaching that of high-pressure steam. This crystal always forms in ordinary foundry practice when tin is present in amount equal to 9 per cent of the copper content. In slow cooling the amount of this crystal is increased and slow cooling also causes it to appear with a lower percentage of tin. High-tin bronzes should neither be used under high temperature nor where they are subjected to severe shock. Without doubt this eutectoid crystal constitutes the most valuable bearing crystal for bearing metals to be found in any of the bronzes.

6 Considerable experimental work has been done at different times and in different laboratories on the subject of heat treatment of foundry bronzes and brasses which are to be used without subsequent forging or tooling. These experiments have been applied particularly to the copper-aluminum alloys and to the copper-tin alloy known as gun metal. Nothing, however, has as yet developed to bring this process of heat treatment into general commercial importance.

DEOXIDIZERS

7 *Magnesium* has a strong affinity for oxygen. It corrodes quite readily when exposed to moist air. It is the most natural deoxidizer for zinc. It is also very useful for deoxidizing copper in the making of pure copper castings requiring a high electrical conductivity. Unlike many other deoxidizing agents, a slight excess of magnesium will not materially reduce the electrical conductivity.

8 *Aluminum* also has a very strong affinity for oxygen, ranking next to magnesium in this respect. It is an excellent deoxidizing agent in all alloys where it is intended that it should be present. It produces a characteristic surface which is not adapted for soldering.

9 *Manganese* has a high affinity for oxygen, ranking next to aluminum in this regard. It is supplied commercially as manganese copper, which contains about 30 per cent of manganese. It is not only a deoxidizing agent when alloyed with copper but it is also a desulphurizer. When used in copper castings a mere trace greatly increases their electrical resistance.

10 *Zinc* is very generally used for alloying with copper. In its affinity for oxygen it ranks next to manganese but, owing to its

high affinity for copper and its tendency to form a solid solution, it is not as good a deoxidizing agent as might be expected.

11 *Silicon* has considerable affinity for oxygen when present in the molten state of an alloy. It is supplied to the trade in the form of silicon copper, a product of the electric furnace containing from 10 to 50 per cent of silicon. It became very popular as a deoxidizing agent in the brass foundry some fifteen years ago, but has largely been replaced by phosphorus for this purpose owing to the convenient form in which the latter is supplied. Silicon has the disadvantage that it cannot be used in any alloy containing lead, owing to its tendency to unite with lead and cause porous and spongy castings.

12 *Phosphorus* is now most universally used in the brass foundry as a deoxidizing agent for copper. New copper can be melted in a crucible and complete deoxidization effected with one-tenth of one per cent of phosphorus. It has the disadvantage, however, of reducing the electrical conductivity wherever it is used. The phosphorus-copper crystal (PCu_3) is very hard and a very small percentage of phosphorus tends to harden any of the alloys in which it is present. If bronze containing more than a trace of phosphorus is to be used as a bearing metal, it should also contain an appreciable amount of lead in order to minimize the abrasive character of the phosphorus-copper crystal.

13 *Boron Suboxide.* When this compound was first introduced it was the most successful deoxidizing agent for copper, in that not a trace of boron remains in the deoxidized copper, and the effect is exceedingly good, making a pure deoxidized copper with high electrical conductivity. It has recently been superseded, however, by boron carbide, which is even more active as a deoxidizing agent. Both require skill and expert knowledge to handle successfully in the foundry.

FLUXES

14 The use of fluxes in the brass foundry is of considerable importance. If pure metals could be melted without coming in contact with air, fluxes would not be required and therefore it is essential that proper fluxing be resorted to. Covering the surfaces of the molten metal with charcoal is always beneficial, that is with metals and alloys whose melting temperature is above that of the ignition point of the charcoal.

15 The function of a flux is twofold: first, it should dissolve in itself all the metallic oxides; and secondly, when in a molten state it

should completely cover the surfaces of the molten metal and thus exclude all air. A good flux should not be too destructive on the walls of the furnace or crucible. Satisfactory castings may be made by melting pure metals in a crucible and covering with charcoal. In the remelting of gates, risers and skimmers which have been more or less oxidized, it is necessary to use both a deoxidizing agent and a flux. The best two fluxes known are probably borax and fluor-spar. The first of these is too expensive for ordinary foundry use. The second, when pure, is too severe upon the walls of crucible and furnace. Various fluxes are now on the market containing, in part, fluor-spar and an admixture of other materials which tend to reduce the destructive action of the fluor-spar upon the furnace and crucible walls. Broken glass is often used and is quite satisfactory, especially if of a low fusing temperature. Glass however, has also a destructive action upon the crucible and furnace walls, so that the fluor-spar mixture fluxes upon the market are the best for the purpose.

16 Brass-foundry products do not compete directly with cast iron, cast steel or steel forgings in machine construction. They are specified by the designer only when certain properties are necessary, such as appearance, bearing value, chemical composition, and the like. These non-ferrous products never can compete directly in cost with the ferrous products for the reasons that copper is always higher-priced than iron and the price of tin is always subject to foreign speculation; while few of the minor constituents compare favorably in price with iron.

BRONZES

17 *Manganese Bronze* is the term generally applied to a composition which, properly speaking, is a brass instead of a bronze. It is an alloy of copper containing a high percentage of zinc, aluminum and iron, in which a small amount of manganese has been used. The beneficial effects in this alloy, due to the deoxidizing and iron-carrying properties of the manganese, are very striking. Beneficial results are obtained even though no more than a mere trace of manganese finally remains in the resulting product. In none of the brass-foundry products are the high merits of an alloy so much due to the results of skilful and intelligent foundry practice as in the case of this particular alloy. The composition is relatively cheap when the cost of its constituents is considered. The *cost* of the final product, however, is much higher than its composition would indicate, due to the following requirements necessary for the production of satis-

factory castings: The introduction of a small percentage of iron is necessary for proper results, and care and skill with proper equipment are indispensable for this operation; secondly, the amount of metal required for gates, heads and risers necessary for this alloy is unusually large, so much so that in the production of ordinary automobile castings only from 17 to 39 per cent of the entire metal melted finally appears in the form of finished castings; and lastly, the loss in melting is unusually large. For machine construction this alloy, of all brass-foundry products, is of first importance. In strength it successfully competes with steel castings and steel drop forgings although the price is from three to five times that of malleable castings. This, and its non-corroding properties make it a most desirable material in automobile and marine construction. As a foundry product it is a very desirable material as it has a very smooth surface. The labor for finishing it is in many cases so light compared with that required for finishing steel castings that it can often compete with steel castings. This is especially true in automobile construction. It has the necessary strength for brackets, standards and fixtures, so that it can replace both steel castings and forgings for these purposes. Manganese bronze has a field of usefulness in experimental work in that it can be produced with less delay than steel forgings, stampings or malleable castings, requiring as it does so little finishing. By varying the two principal constituents of manganese bronze the desired physical characteristics can be obtained. An increase of the percentage of copper increases its elongation, whereas an increase in zinc increases its ultimate tensile strength but decreases its elongation. When this material is made up with the necessary precautions as to purity and foundry methods, very excellent commercial results can be obtained. For example, a tensile strength of 90,000 lb. per sq. in. with 30 per cent elongation can be secured in a standard test coupon, whereas many producers consider 65,000 lb. tensile strength with 15 per cent elongation as fair.

18 *Aluminum Bronze* is an alloy of copper and aluminum containing from 4 to 11 per cent of aluminum and the remainder copper. In recent years, however, this alloy has been made up with an addition of from 1 to 6 per cent of iron. The introduction of iron produces twofold results: First, it causes the alloy to solidify with smaller crystals, that is, it produces a finer grain; and secondly, it adds a third hard crystal formed by the union of aluminum and iron which is much harder than the other two crystals. This is the only brass-foundry product which can compete commercially with man-

ganese bronze. With care it can be so made up as to have a somewhat higher ultimate strength than manganese bronze, although it does not possess a higher yield point or elastic limit. The drawback to this alloy is the inherent difficulty of making solid castings. The tendency to have blowholes is very characteristic and even extreme skill will overcome this difficulty only in a degree. Aluminum bronze may be used for bearing purposes, and recently, when containing iron, has had a limited use for motor-truck worm drives. The only advantage of this latter alloy over a copper-tin-bronze composition is its cheapness; for bearing purposes, however, it cannot compare with a copper-tin alloy.

19 *Acid-Resisting Bronzes.* These are some of the brass foundry products which formerly were of more importance than they are today, owing to the fact that stoneware and specially modified cast iron are now largely replacing these uses. There are, however, some uses for bronzes to resist sulphuric and sulphurous acids in mine service, pulp mills, and other industries. For these purposes two alloys are most serviceable; a copper-tin-lead alloy containing substantially 86 per cent copper, 12 per cent tin and 2 per cent lead; and aluminum bronzes containing a small iron percentage.

20 *Copper-Tin Alloys.* First among these alloys highest in its tin content may be mentioned bell metal, a composition containing from 16 to 25 per cent of tin. As the name implies, it is used for the manufacture of bells, gongs, steam whistles and the like. It is a resonant metal. For industrial purposes the compositions used are those having the lower percentages of tin, say from 16 to 18 per cent. The higher percentages of tin, especially between 20 to 25 per cent, are used for bells in which high tone quality is required. The production of these bells requires considerable skill and constitutes a special industry. In general, however, it may be said that for all purposes the constituents for these alloys should be extremely pure and every precaution against oxidation should be taken during melting and pouring, since the quality of the product is altogether dependent upon the purity of the alloy.

21 The next alloy of this type to be considered is a bronze used in turntables and movable bridges. This composition is given in the tentative A.S.T.M. specification as "Class A, Bronze Bearing Metals for Turntables and Movable Bridges." It is also given as "Grade A" in the American Railway Engineering Association specification bulletin of July, 1918. It contains 20 per cent of tin, and

phosphorus not to exceed 1 per cent. It is very hard, due both to its tin and its phosphorus contents. For best service conditions it should not be used except when placed between hardened steel plates. It is intended for pressures over 1500 lb. per sq. in. and slow-moving bearings.

22 A composition similar to the foregoing is one given as "Class B Bronze" under both of the immediately foregoing specifications and containing 17 per cent of tin with a permissible phosphorus content of 1 per cent. It does not seem wise to permit this amount of phosphorus in a bearing metal of this kind, however, if to be used as specified between "soft steel plates." Better service conditions can be obtained by the production of the necessary hardness through increase of tin content and reduction of phosphorus. An alloy containing 19 to 20 per cent tin and only a trace of phosphorus would give better service when used on soft steel. In a case of this kind where a copper-tin bronze is used between unhardened steel surfaces the amount of phosphorus present should be very limited owing to the exceeding hardness of the copper phosphide crystals.

23 The next alloy of importance as to its tin content is a bronze containing 11 per cent of tin and a maximum of 0.2 per cent of phosphorus. This alloy is the standard composition now in very general use for worm wheels in motor-truck drives and in the motor-car reduction worm gears. A feature of considerable importance in the making of this alloy seems to be that of producing the proper amount of chilling effect in the cooling of the castings. In some cases zinc has been added to an amount of 2¼ per cent; results show, however, that the addition of zinc is undesirable although it produces an additional hardness of the alloy as indicated by the Brinell and scleroscope tests. This additional hardness rather detracts from its bearing value, and, at the same time, makes it more difficult for tooling. It seems self-evident that the above alloy, as it now stands, could be improved by the addition of a small amount of lead, — not sufficient to produce undue weakening, — and that this would result in a threefold benefit: easier tooling, increased accuracy of tooled surfaces, and a higher bearing value.

24 *Copper-Tin-Zinc Alloy.* The most popular alloy of this class is the one known as ordnance bronze or gun metal. Its composition is 88 copper, 10 tin and 2 zinc. This alloy has come to be a very generally known composition in that it is a general-average bronze. However, with a more complete metallurgical knowledge

of the relative values of different compositions, it becomes of less importance, in that it is specifically neither a bearing bronze of high merit nor a bronze especially adapted for machine parts when maximum economy of alloy is considered. The high tin content produces in this alloy a high percentage of the tin-copper eutectoid, with the resulting disadvantages when the alloy is used for other than bearing purposes. The use of ordnance bronze for the construction of machine parts is, for two reasons, a wasteful one: It contains an unnecessary amount of tin, a high-priced constituent, and more serviceable alloys can be produced by the use of a lower percentage of tin. For machine parts, steam connections and valves, a composition having, say, 90 per cent copper, 6½ tin, 2 zinc and 1½ lead is a superior alloy for the reasons that it can be produced as a more economic foundry product, its price is less, its physical properties are superior, and lastly, its machining qualities are superior. On the other hand, the zinc in ordnance bronze distinctly detracts from its bearing value.

25 *Copper-Tin-Lead Alloys.* These alloys are strictly bearing compositions containing not in excess of 82 per cent of copper and a tin maximum of 11 per cent with a lead maximum also of 11 per cent. This range of alloy corresponds to "Class C" of the A.S.T.M. tentative specifications for Bronze Bearing Metals for Turntable and Movable Railway Bridges. These alloys are usually deoxidized with phosphorus which has a tendency to suspend or diffuse the lead, since in all these alloys the lead is held in mechanical suspension without any chemical union. The most generally used of this series contains 80 per cent copper, 10 tin and 10 lead.

26 To enumerate all of the copper-tin-lead alloys given by various authorities as bearing bronzes would constitute an almost infinite series. The whole matter, however, can be summed up in a few words: Lead is a desirable constituent in a bearing alloy but has the drawback of weakening the bronzes, therefore, for light service, or under conditions where a supporting oil film cannot always be maintained, as in car or truck bearings, the compositions containing the higher proportions of lead should be used. Where the punishment is severe the alloys should have a lower percentage of lead. As a rule it is economy to use as much lead as possible, since lead is a cheaper metal and it also adds materially to the bearing value of the alloy.

27 *Copper-Lead-Tin-Zinc Alloys* are those containing from 3 to 12 per cent of lead, from 4 to 6 per cent of tin and from 1 to 10 per

cent of zinc. This series of compositions includes several used for bearing purposes, especially those having the high lead limits. The addition of zinc in this alloy is for the purpose of reducing the cost; it does not add to the value of the alloy except in its physical strength and it detracts from its bearing value. The series includes the popular red-brass composition; 85 per cent copper, 5 tin and 5 zinc; and also a cheaper composition containing 77 per cent copper, 10 lead, 3 tin and 10 zinc used for low-pressure valves and plumbers' supplies.

DISCUSSION

THE following discussion on the six papers comprising the Symposium on Castings is grouped according to the subjects discussed.

MALLEABLE CASTINGS

J. L. HUGHES. In Par. 11 the author refers to the Ordnance Department as being unfamiliar with the physical characteristics of malleable iron. The Ordnance Department has been trying to make drop bombs of malleable iron and has searched the malleable foundries for satisfactory castings. A hydrostatic test of the bombs is necessary and the malleable castings are too porous to stand this test. Great difficulty has been experienced in inducing the malleable foundries to undertake the necessary research to produce satisfactory castings for this purpose. At present the Department is about to discontinue the use of malleable castings due to its inability to have them properly cast. Although the specifications regarding the hydrostatic test have been lowered and the castings varnished in an attempt to minimize porosity, satisfaction has not been obtained.

H. WADE HIBBARD. Thick sections in malleable castings have been feared by the designer and objected to by the manufacturer of malleable castings, and yet in Par. 12 we read "Castings have been produced commercially as long as 5 ft., with sections at some parts as thick as 3 in." With certain of our malleable iron foundries, the ability to produce malleable iron sections as thick as three inches or more has been known for some decades. The method of producing them seems to be to place a chill in the mold in contact with the thicker sections so that an essentially white and brittle iron will be produced at this point. This makes the sections of a nature which will undergo the annealing process.

E. R. Jackson stated that the malleable iron parts furnished for certain trucks and tractors in the army had not given satisfactory service. Of twenty motor trucks fitted with malleable valve tappets, not a single one crossed the continent without the replacement of the tappets. The author stated that malleable iron could be used for such purposes. What experience had other members with malleable iron for such purposes?

Richard Moldenke. Captain Hughes is correct in his criticism of malleable iron being porous. The reason is that a white iron casting has a volumnar contraction from the molten to the solid state of over 12 per cent, and unless the casting is poured with care the outside will be sound and the inside porous.

Commenting upon the discussion of Professor Hibbard, Dr. Moldenke said that the malleable casting must be of white iron in order to be successfully annealed. For that reason, chills are used to cool quickly the thicker sections. If the iron set as gray iron, the annealing process would only ruin it. Malleable iron was a superior material for purposes for which it was adapted. Compared with cast steel it was superior only in withstanding shock.

The malleable parts mentioned by E. R. Jackson, which had failed as valve tappets, had been machined. The change in the strength of the iron in malleable castings as made in this country was confined to the region of the skin of the casting. Hence if this is removed in machining, the better portion of the material is taken away. The malleable castings made in Europe by another process are more nearly a steel throughout and are more readily used for machine parts.

In answer to a question about the use of permanent molds, Dr. Moldenke replied that future development should be not in the direction of permanent iron molds but rather "long-life" molds which might be used many times. He recited experience to show the impracticability and expense of permanent molds and spoke of the possibility of using a mold made of gas carbon and binder pitch and others in which fire clay formed important portions of the mold.

The Author has read the very interesting discussions of his paper. The author understands on the best authority that the statement concerning the Ordnance Department was acknowledged by them to be the case on numerous occasions when an attempt was made to have them adopt the malleable iron casting for many parts for which it was thought to be well suited. The drop bomb is a casting very difficult to make sound when made of any metal. During the war the American Malleable Castings Association placed its various foundries and its research laboratory at the disposal of the Ordnance Department to conduct any research or experimental work that this or any other governmental department desired should be made. The author is absolutely positive of this statement as the offer was made through him in person and in public. Since the war the malleable iron foundries have been so loaded with work that they have been obliged to refuse orders of even the most inviting character. This situation has not been occasioned by lack of foundry capacity at all, but wholly a lack of molders. Under these circumstances it could hardly be expected that any foundry would desire to experiment at the present time with the drop bomb. The contention that malleable cast iron is not sufficiently sound to make a drop bomb can very well be questioned. The author is consulting engineer of a concern that manufactures a very large tonnage of ammonia fittings, all of which are tested hydrostatically. Taking for illustration a very recent case only, that of a flange, a change in the molding practice resulted in reducing the loss due to leakage from over 50 per cent to one tenth of one per cent, as there was but one leaky flange in the one thousand made. The cause for unsoundness is often placed on the metal instead of where it properly belongs. A metal that will average 0.90 per cent in silicon, 2.35 per cent in carbon, and whose temperature when tapped from the furnace will average about 2700 deg. fahr., must be acknowledged to have properties that make for soundness. Is there any ferrous metal from which more minute castings can be made? If malleable iron possesses such a high degree of fluidity as is implied in this question, is it likely to be porous provided molding conditions are correct?

Concerning the discussion by Mr. Jackson, the author certainly did not intend to convey the idea that malleable cast iron was suitable for use in any part of a motor vehicle. Ignorance as to whether the tappets broke in service or were replaced due to wear makes a reply impractical. The statement made was that "The wearing properties of this metal correspond to what can be expected of wrought iron or dead soft steel." It is possible that a harder metal should have been used for the purpose and it is possible in case the tappets fractured that the castings were inferior.

DIE CASTINGS

Robert J. Anderson.[1] Probably the major part of the technical literature dealing with the production of die castings is owing to Mr. Pack. In regard to light aluminum alloys for die-casting purposes, it is a matter of considerable conjecture as to what is a suitable alloy, i.e., what may be regarded as the best composition. The properties demanded in an aluminum alloy for die casting are essentially different than for sand casting and the question of strength at elevated temperatures is a most important one. At the same time, the elongation should be high.

The light aluminum-copper alloys, exemplified by ordinary No. 12 alloy as well as alloys higher in copper, are relatively strong at elevated temperatures, and their strength is considerably increased by iron additions in moderate amounts — say up to 3 per cent iron. On the other hand, the light aluminum-zinc alloys possess great elongation at high temperatures, and I may cite the alloy containing 80 per cent aluminum and 20 per cent zinc, which has the enormous elongation of 130 per cent at 595 deg. C., as determined by Rosenhain and Archbutt. High elongation at elevated temperatures together with relatively high strength would appear to be prerequisites of an aluminum alloy for die casting owing to the nature of the process. Unfortunately, the aluminum-zinc alloys are quite weak as to strength at high temperatures, and they are also extremely susceptible to shock stresses above 400 deg. C. It would appear, however, that systematic investigations of some ternary or even more complex light aluminum alloys might lead to the development of better alloys for the die-casting process than those which are applicable at present.

STEEL CASTINGS

H. Wade Hibbard. In Par. 17 the author says: "When, however, the service demands all the strength, toughness and resistance to wear and fatigue that are obtainable in a cast metal, a steel casting should be specified." The speaker is perhaps somewhat unduly prejudiced against the suitability of cast steel for a rubbing surface. Those who are more familiar with the subject may be able to state under what condition a steel casting is suitable for a rubbing surface.

E. R. Jackson endorsed Professor Hibbard's remarks about the wearing qualities of cast steel when used for rubbing surfaces.

[1] Metallurgist, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

He said that when the Ordnance Department began the manufacture of military trucks, some of the parts which were to have been made of forgings were made of cast steel to facilitate production. The cast steel parts furnished in the first few hundred vehicles were so satisfactory that forgings have not been made since and it is expected that the specifications will be changed to require cast steel.

JOHN McGEORGE spoke of the necessity of making cores for steel castings so that they will crush under the shrinkage of the metal. Otherwise internal strains are set up in the casting which can be removed only by annealing.

RICHARD MOLDENKE said that while he was not as familiar with steel casting practice as with gray iron, he did know that in the case of the latter the metal would give or crack if the core were too hard. For this reason it was their practice to make test castings from patterns before regular molding proceeded to see if the pattern was properly constructed to allow for variation in shrinkage.

GRAY-IRON CASTINGS

H. WADE HIBBARD asked what was meant by the term "locally chilled" as used by the author in Par. 9. He was under the impression that a special grade of cast iron was necessary for chilled surfaces.

Dr. Moldenke explained that the word "chilled" had two meanings in foundry practice. It might mean that the iron was turned perfectly white, or, as in the case in question, that the surface was rapidly cooled, making the structure of the casting locally harder.

E. R. JACKSON stated that during the war he had made castings with a high percentage of scrap, transferring the molten metal from the cupola to the electric furnace for refining as mentioned by the author and that the results were satisfactory.

In making synthetic pig iron by charcoal for making steel castings, due to failure to obtain transportation for pig iron from regular sources, he had been able to obtain only 2.30 in carbon.

DR. MOLDENKE replied that in Canada castings had been obtained in this manner as high as 3.45 carbon. Putting in charcoal, silicon and high carbon steel, 3.25, 3.30 and 3.35 carbon were obtained. This cannot be done in the cupola but requires the high temperature of the electric furnace.

JOHN McGEORGE asked if the basic lining of the electric furnace mentioned by the author was successful in eliminating phosphorous from the iron. The author replied that it was not.

BRASS AND BRONZE CASTINGS

ROBERT J. ANDERSON. The paper by Mr. Bierbaum mentions the fact that in the case of manganese bronze, sprue returns are exceptionally large — so much so that in the manufacture of ordinary automotive castings only about 17 to 39 per cent of the entire metal melted finally appears in the form of finished castings. In the other papers comprising the symposium on castings, the writer does not find any references to sprue returns, casting losses, or mèlting losses. These considerations are of special importance in the selection of an alloy for the production of castings, and in fact they might properly be regarded as governing factors in a sense.

Mr. Bierbaum does not give any data on casting losses in brass and bronze, and the writer wishes to inquire whether he has any figures. In the founding of light aluminum-alloy castings, about 10 to 12 per cent may be taken as an average figure, on the basis of an investigation recently made by the Bureau of Mines. Average figures probably do not mean much, since casting losses necessarily vary a great deal, depending upon the type of castings made, and upon other determining factors. However, it would be useful for comparative purposes to know what the average casting losses in brass and bronze are. The writer does not refer to the melting losses, but has specific reference to actual scrap (casting) losses incurred because of rejections of castings in which foundry defects exist.

C. H. BIERBAUM. The writer can give the desired information for only one plant, one with which he is connected, namely, the foundry department of the Buffalo plant of the Lumen Bearing Company. The data for the year 1919 are as follows:

Total pounds of castings made, lb.	5,025,500
Average weight of castings, lb.	2.2
Cored castings, per cent	94

Imperfect castings, by weight, per cent	9.81
Castings returned from customers, per cent	0.83
14 Standard alloys were made, together with special compositions specified by customers.	

In considering the foregoing data it should be borne in mind that none of the work done in the plant can be regarded as a specialty, it is strictly a jobbing foundry, and that a very large percentage of the patterns are not even gated.

It can readily be appreciated that the foregoing results can only be obtained in a highly organized foundry, one with trained and skilled workmen using the most approved up-to-date foundry methods, with the entire work under the direction of a well-trained chemical, metallurgical and engineering staff.

No. 1744

AN IMPROVED FORM OF WEIR FOR GAGING IN OPEN CHANNELS

By Clemens Herschel,[1] New York, N. Y.
Non-Member

In the following paper are presented the results of a series of investigations on weirs which should mark an epoch in the measurement of water by that means. Asked by the Power Test Code Committee of The American Society of Mechanical Engineers to advise it with regard to weir measurements, and supplied by Engineering Foundation with the necessary funds to conduct the experiments, the author devised an entirely new weir which was tested in the Hydraulic Laboratory of the Massachusetts Institute of Technology. The tests revealed that for discharges of from 0 to 9.55 cu. ft. per sec. per ft. of weir length, the limits covered by the experiments, the quantity of water flowing, Q, *was directly proportional to the difference,* d, *in two pressures, one measured just upstream of the weir, corrected automatically for the velocity of approach, and the other measured at the crest, the formula being* $Q = 5.50\ d$. *The weir, which bears no resemblance to the sharp-edged weir now in use, has a 2 : 1 slope of approach to the crest. The crest is made in the form of an arc of a circle and is hollow for observing the pressure at that point. The nappe of the stream is supported for a short distance beyond the crest by another 2 : 1 slope. The paper opens with a résumé of the history of weir measurements and contains descriptions and diagrams of the apparatus and tests of the new weir, with tabular and graphic reports of the results obtained.*

> "Daniel Bernoulli often said to me to eschew all those complicated formulae; he believing that nature works according to laws that are too simple to lead to them; and if one finds such, that it is because one's computations have been based on false preliminary assumptions."—
> R. Wolf, Biographies Relating to the History of Civilization in Switzerland,[2] Vol. 3 (1860), p. 176.

[1] Hydraulic Engineer, 2 Wall Street.

[2] "M. D. Bernoulli m'a souvent dit de me défier de toutes ces formules compliquées; il croit que la nature est trop simple pour y mener; et que si on en trouve, c'est qu'on a fondé ses calculs sur des fausses hypothèses." — R. Wolf, Biographien zur Kulturgeschichte der Schweiz, Band 3 (1860), p. 176.

Presented at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers.

TO any one whose memory of hydraulic engineering in the United States, more especially in New England, runs back 60, or only 50, years, the standing at that time of the "Francis formula" for weir discharge will appear as little short of infallability, or of perfection. Courts of justice recognized it as decisive; lawyers of eminence played with it as though it were one of the choicest morsels from the tree of knowledge; it was the last word in gaging the flow of water. And there was reason for such an estimate of its value at the time.

2 James Bicheno Francis was then easily the leading hydraulic engineer in the United States, with perhaps not over half a dozen of the species in the whole country. Grappling as a young man with the problem of insuring their lawful quantities, day by day, to each of the seven "Proprietors of the Locks and Canals on Merrimack River," at Lowell, Massachusetts, out of the common trough of canals which fed them collectively, he may be said to have been the first one who thus brought order out of the disorder which that communistic distribution of water — naturally accompanying the American system of power distribution canals for manufacturing purposes, and now passing away in favor of the electrical distribution of power — always brings with it. For the purpose named he instituted, about 1850, the series of hydraulic experiments described in his book entitled, Lowell Hydraulic Experiments; and he did this with all the accuracy obtainable at the time and place that have been named.

3 From another point of view than just described were his experiments of a new order of importance. If any one will consult Weisbach's Experimental Hydraulik, or the later editions (1855-60), of Weisbach's Mechanics, he will be surprised at the Lilliputian dimensions of the apparatus then in vogue as the basis of engineering knowledge for use in the practice of hydraulics. Thus Weisbach's experimental reservoir had an area of 1.35 sq. ft.; and his ajutages were ordinarily about three hundredths of a foot in width, or diameter. Of course, we know that Atwood's machine, of extremely moderate dimensions, will measure the force of gravity exercised by the globe which we inhabit; but it requires more confidence in the "horse sense," no less than the technical ability of hydraulic engineers, than is possessed by the average mill owner or his agent to give him confidence in the work of the engineer he employs, should that engineer base it on experiments such as inspired Julius Weisbach or his immediate disciples. So that the experi-

ments of Francis, conducted out of doors on a scale commensurate with their importance, at the Lower Locks near the Concord River, came to the interested "Proprietors" and others as a refreshing and new conquest of mind over matter. They were much more reliable because made on a natural scale; exact, as every one could appreciate, from the care taken in making the experiments and the computations following the experiments, and thoroughly reliable.

4 At that time arose the expression, "made with a standard weir," which has remained alive to the present day, although, as will presently be seen, there was no propriety whatever for calling any weir a "standard." The only standard water-gaging apparatus is a tank or reservoir. Weirs, orifices, venturi tubes and meters, and other water-gaging apparatus and methods can become truly rated only by comparison with tank measurements, and thus thereafter competent to give reliable service. Such comparisons, moreover, establish the *degree* of reliability of the several methods that have been subjected to comparison, and from this point of view the weir has for many years been falling into deserved disrepute.

5 To treat the matter somewhat chronologically, there is the article in *Engineering News* for November 10, 1898, by the present writer, protesting against the multiplicity of formulæ that sharp-edged weirs and orifices, and present methods of observing heads upon them, bring in their train; and suggesting that observations be taken *at* the weir or orifice; and that weirs and orifices cut "in a thin plate" be abandoned.

6 In the Transactions of the American Society of Civil Engineers for 1914, the discussion on the paper of R. R. Lyman, M.Am. Soc.C.E., brought out clearly the number of methods there were in use to measure "head upon the weir," and their idiosyncracies. These several methods go far beyond any forethought of the pioneer experimenters in that sort of work, and definitely confirm the rule that when any one formula is to be used to compute weir discharge, the weir and methods of taking observations of flow over it must imitate with slavish exactness the construction of the weir and methods of taking observations that were followed in constructing the selected weir formula; a matter of annoyance or difficulty as a rule.

7 Fteley and Stearns, for example, Trans.Am.Soc.C.E. for 1883, p. 5, say: "It is, however, almost impossible to cover all cases that may occur in practice."

8 Bazin, in his 1890 article, Part I, for that year, of *Annales des Ponts et Chaussées*, p. 4, and elsewhere, says that to attempt to experiment on all cases possibly to be met with in practice (instead of setting up a standard form to be used in such work) must be pronounced impracticable. And in his 1898 article, Part II for that year, p. 161, he says: "Each type of weir would therefore require a special study; and face to face with a complexity like this, one must abandon the idea of establishing formulæ for general application."

9 R. E. Horton, Trans.Am.Soc.C.E., 1914, p. 1300, tells us that "of the making of many formulæ there is no end." Also, p. 1326, "If Francis had used higher and lower heads, he would have discovered that no one equation of the form he uses, $Q = mh^n$, can fit accurately a series of experiments on weirs of the type herein considered."

10 Hamilton Smith[1] saw this, and very likely Francis did, for Francis was very particular to state that the formulæ he gives must be used only between stated limits of head, and only upon fulfillment of other stated conditions of weir construction and operation.

11 Finally, in the discussion on the paper by Floyd A. Nagler, Trans.Am.Soc.C.E. for 1918, the subject now under consideration was again debated by interested hydraulic engineers, the present writer among the number; and the status of the whole question was summed up by F. P. Stearns, M.Am.Soc.C.E., himself an experimenter on weirs of more than 40 years' experience, when he said with startling, yet righteous, precision that "in the present state of the art the weir is not an accurate instrument for the measurement of water."

AN ADVANCE IN THE STATE OF THE ART BELIEVED TO HAVE BEEN NOW ATTAINED

12 After this survey of the state of the art, and to give a résumé of what this article will endeavor to establish, it may not be improper to say at once that it treats of a weir whose discharge is represented by a straight-line formula from 0 up to 9.55 cu. ft. per sec. per ft. in length of weir, and thus continuing, presumably, indefinitely beyond that quantity. Its equation is $Q = 5.50\ d$; d

[1] Hydraulics, John Wiley & Sons, Inc.,1886.

being the difference of two observations of water height or pressure; the one taken just upstream from the weir (taken so as to be automatically corrected for "velocity of approach," if one pleases, or corrected by computation before being used in the above formula, if preferred), and the other taken at the crest. In both cases d is obtained by measuring the elevation of a water surface in a pail by means of a hook gage, or otherwise as desired. This *difference* of pressures may also be obtained by means of one of the many apparatus already invented and on the market for measuring differences of pressure, in which case the register may indicate mechanically the quantity passing at any moment, sum it up for any given period of time, portray it graphically on a chart, or simultaneously do all three of these recording operations.

13 It will be observed that the process just described is not open to the criticism of R. E. Horton, mentioned above, who said, "No one equation, of the form he (Francis) uses, $Q = mh^n$, can fit accurately a series of experiments on weirs of the type (sharp-edged weirs) herein considered." It does this, moreover, with the added simplicity of making the exponent n equal to 1.

14 And the reason it does all this is because it is carried out with a weir that is not a "sharp-edged weir," the type that has rested like the incubus of an evil spirit upon the minds of all hydraulicians for full 200 years, ever since the Marchese G. Poleni first wrote about weirs in De Motu Aquae Mixto, Patavii, 1717. Bazin and others have thought that to attempt to set up a standard form of weir for measuring water was visionary, in view of the many variations in weir discharge so easily brought about. But this very thing was one of the objects of the experiments now under consideration, and time alone will tell whether or no it has not after all been accomplished.

THE POWER TEST CODE COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, AND ENGINEERING FOUNDATION

15 After this much said in anticipation, we will go back now to a chronological order of events. The American Society of Mechanical Engineers, having appointed a Committee to draft a revised form of Power Test Code in 1917, and invited the three other Founder Societies to appoint advisory committees to the same end, the writer was appointed as one such advisory member. It thereupon very soon appeared that the matter of weir measurements and of their

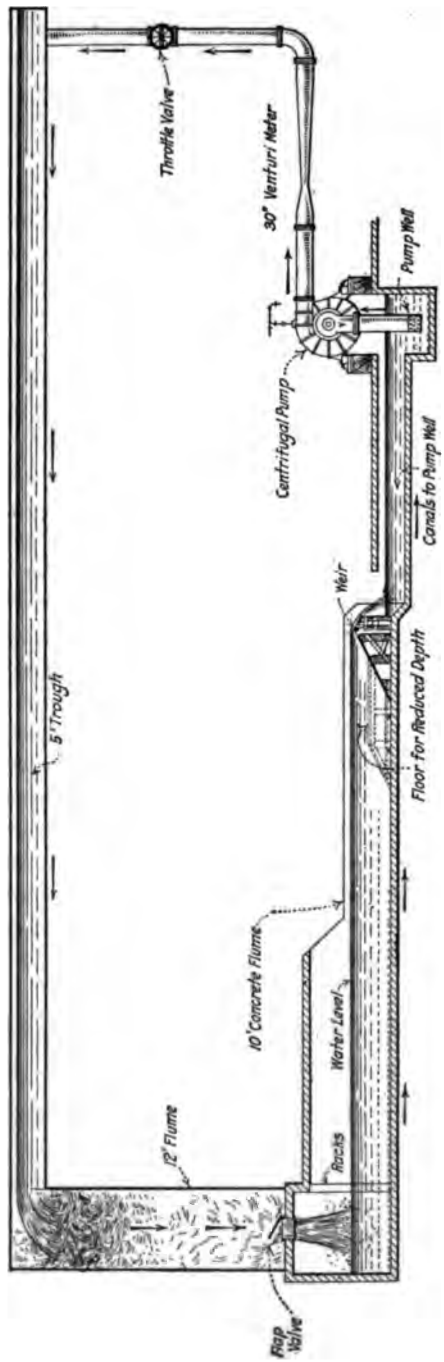

FIG. 1 DIAGRAMMATIC REPRESENTATION OF THE EXPERIMENTAL APPARATUS

accuracy was in a most deplorable state of confusion. On the proper representations having been made to Engineering Foundation that a new set of experiments on weir discharge were liable to lead to a considerable advance in knowledge in this branch of applied science, it made an appropriation for that purpose, to be expended by the writer.[1] The experiments about to be described have been the result of these preliminaries.

THE HYDRAULIC LABORATORY OF THE MASSACHUSETTS INSTITUTE OF TECHNOLOGY

16 The place selected for conducting the experiments was the Hydraulic Laboratory of the Massachusetts Institute of Technology. This laboratory had barely been completed when the United States entered the World War, and all the energies of the Institute thenceforth had been devoted to its notable war work to a degree that left improvement in and perfection of its school apparatus rightfully a secondary consideration. As a consequence, the hydraulic part of the laboratory on closer study was shown to have many remediable defects, which, no doubt, will be eliminated in course of time. These, again, affected both the difficulty of making the contemplated experiments and the precision at which results could be attained. But the choice had been made, many reasons existed for persevering in the place where the work had been planned, and thus the experiments were completed as they had been projected.

17 For a description of the Institute Hydraulic Laboratory reference is made to an account in the March 1918 number of the *Journal* of the Boston Society of Civil Engineers, by Prof. George E. Russell of the Institute. Briefly stated, the apparatus used in the present experiments and shown diagrammatically in Fig. 1 consisted of a 350-hp. Ball angle-compound engine, 240 r.p.m., direct-connected to a 30-in. Worthington centrifugal pump, having some 48 cu. ft.

[1] Acknowledgment is also here gratefully made to the President and Faculty of the Massachusetts Institute of Technology, especially to Prof. Edward F. Miller; to Wm. F. Uhl, B. S. Rose and H. L. Woodruff of the office of Chas. T. Main and to Mr. Main; to Chas. W. Sherman, H. P. Eddy, Jr., and others of the office of Metcalf & Eddy and to their principals, all of Boston; and to Builders Iron Foundry, of Providence, R. I.; for most efficient aid rendered in various ways at various stages in the conduct of the experiments herein described; and done in a way that amounted to a contribution to enable all the described work to be done.

per sec. maximum capacity, at 40 to 45 ft. lift. This pump takes its suction from a system of canals under the floor of the basement, which are or may be connected with the Charles River Basin.

18 Right here arose one great obstacle in the making of these hydraulic experiments. The canals that have been named at times took to "surging," causing oscillations of the pump-well surface, and these, in turn, were transmitted through the pump, and could plainly be seen at the upstream end of the discharge pipe. They were also indicated by the 30-in. venturi meter set in line of the discharge pipe, next downstream from the pump. Two whole afternoons' work was rendered useless from this cause. But by experience and careful work, any gross effect from this cause could be avoided.

19 The discharge pipe delivered into a riveted trough of steel, 5 ft. by 5 ft. and about 135 ft. long; at some 40 ft. higher elevation than the pump. In it was set the 16-in. by $5\frac{1}{4}$-in. venturi meter, used to measure quantities smaller than about 8 cu. ft. per sec. (September 25, 26, and 29), the larger quantities being measured by the 30-in. meter already spoken of.

20 Both these meters were read by means of mercury straight-tube manometers, as made by the builders of the meters. The scale of these manometers was based on a coefficient of discharge for the meters of 0.99. These were the facilities at hand, or readily procurable and installable; and though not so precise as a tank measurement would have been, were sufficiently so for the pioneer work to be done. It is not often, as yet, that one can get a reservoir to hold 46 cu. ft. per sec. for the period of an hydraulic experiment; and in the opinion of the writer, the venturi meter is the next best thing. It certainly was at the hydraulic laboratory that had been selected for making the present experiments.

21 That the venturi meter is of a high order of accuracy has been shown by a compilation by the writer for the Power Test Code Committee of all attainable relevant experiments made with it during its existence of 30-odd years. See Table 1. A very significant series of tests, made by F. H. Shaw, Mem.Am.Soc.C.E., of a meter at Lancaster, Pa., show an average variation of 0.036 of 1 per cent for 44 tests, covering a period of eight years. See Table 2.

22 The variations of this record illustrate the imperfections of human abilities rather than those of the meter. The meter, with its bronze-lined throat, surely did not vary in its action, as the 8-year

record indicates; nor did the laws of nature, as manifested in the science of hydraulics. Nor can the city water-supply reservoir that was used as a tank be supposed to have varied in capacity.

23 What, then, did vary? Plainly the accuracy of the human element brought into play.

24 In another place the present writer has compared the case with the familiar results found on repeatedly measuring, say, with a steel tape, a base line. Nothing is simpler in execution, yet, as every engineer knows, no two measurements are ever precisely alike except by accident. The end points have not moved; the steel tape has been compensated each time for stretch and temperature; and yet we take an average of measurements to represent the true distance apart of the end points. It may be argued that the percentage of variance in the two cases is different, and smaller in the case of linear measurements; but, on the other hand, feet are easier to measure than cubic feet per second. The one deals with a simple dimension, the other with four.

25 To return to our present meter observations. The scale of the manometers read tenths of cubic feet per second; and hundredths could readily be estimated, on both of the meters used.

26 From the 5-ft. trough the water fell into a vertical flume or wheel pit 12 ft. in diameter, made of riveted steel, and 22.5 ft. high; and through a wooden adjustable flap valve in the floor of this flume, fell into a concrete tail-race pit, which, through racks and gridiron-valve gates, discharged into the concrete channel about 72 ft. long that led to the weir to be tested. The discharge of the weir fell back into the system of canals which fed the centrifugal pump, thus completing the cycle actuated by the Ball engine and pump.

27 The free discharge of the 5-ft. trough into the 12-ft. vertical flume could be rendered somewhat less turbulent by keeping the flume-water elevation as high as possible without waste, this again being crudely controllable by means of the wooden flap valve in the floor of the vertical flume. But this whole arrangement and the free discharge through the flap valve into the tail-race pit below were imperfect arrangements, acknowledgedly only temporary in character, and were the cause, together with the "surging" of the pump-well water above referred to, of many of the difficulties experienced in making these experiments. They also were the principal cause of most of the lack of precision that remains and is inherent in the final results.

TABLE 1 VENTURI METER COEFFICIENTS[1]

(See Notes on page 197)

		Velocity through the Throat																Remarks
		5 ft. per sec.		10 ft. per sec.		20 ft. per sec.		30 ft. per sec.		40 ft. per sec.		50 ft. per sec.		60 ft. per sec.		Average		
		Meter	Tube	Meter	Tube	Meter	Tube	Meter	Tube	Meter	Tube	Meter	Tube	Meter	Tube	Meter	Tube	
1 ft. × 0.33 ft.	Coeff.		(0.940)		0.987		0.995		0.999		(1.000)							Made of hooped wood.
Tank Test, Herschel	Loss Hd.		0.06		0.26		0.97		2.10									
1 ft. × 0.5 ft.	Coeff.		0.984		0.987		0.991		0.987		0.983						0.987	Cast iron.
Tank Test, Cornell University	Loss Hd.																	
1 ft. × 0.43 ft.	Coeff.		0.983		0.990		0.997		1.000		0.998						0.993	Cast iron.
Tank Test, Cornell University	Loss Hd.		0.06		0.25		0.95		2.10		(3.30)							
1 ft. × 0.33 ft.	Coeff.		0.989		0.985		0.987		0.990		0.993		0.989		1.000		0.988	Cast iron.
Tank Test, Cornell University	Loss Hd.																	
2 ft. × 1 ft.	Coeff.	0.99	0.99	0.99	0.99													Cast iron.
Weir Comparison, Holyoke W. P.	Loss Hd.																	
2.5 ft. × 1.08 ft.	Coeff.			1.00		1.00[2]												Cast iron.
Tank Test, F. H. Shaw	Loss Hd.																	
4 ft. × 1.33 ft.	Coeff.		1.00		0.995		0.986		(0.978)									Wood, inside riveted plate.
Weir Comparison, Herschel	Loss Hd.		0.03		0.15		0.62		(1.44)									
8 ft. × 4 ft.	Coeff.			1.00														Riveted plate.
Tank Test, Pittsburg Water Works	Loss Hd.																	
9 ft. × 3 ft.	Coeff.		(0.994)		0.986		0.978		0.968									Wood, inside riveted plate.
Weir Comparison, Herschel	Loss Hd.		0.04		0.18		0.77		1.54									

[1] Compiled February 1918, by the author.

[2] The average of 44 tests during a period of 8 years; extremes were + 1.10 and − 1.89 per cent.

NOTES TO TABLE 1

Column 1 gives the size of the end and throat diameters of the meter tube; also the nature of the test or experiment, whether volumetric (in a tank) or by comparison with a weir; and the authority for the statements made.

Column 2 refers respectively to the coefficient to be applied to the theoretical discharge of the venturi tube, or of the complete venturi meter, to attain the discharge with which it has been compared, whether tank measurement, or weir. In the latter case, it does not follow that either the coefficient given, or the weir discharge from which this coefficient has been derived is correct.

There is also given, when it has been observed, the loss of head, in feet of water column, end to end of the meter tube.

Both coefficients, and losses of head, are entered as appurtenant to the velocities of water through the throat of the meter tube, in ft. per second, as given in the top main horizontal column.

Columns 3 to the column headed "Remarks," give, alternately, the coefficients and losses of head just spoken of, according as they have been found for the complete venturi meter or for the meter tube only; the latter requiring an observer, notebook and pencil, and computation or diagram, to result in knowledge of the quantity of water discharged.

Column headed "Remarks," gives the construction of the meter tube. It is to be understood, however, that the throat of the meter, and a short section at the upstream end, are invariably made of or lined with bronze.

Besides tank tests, and comparisons with weir gagings, we have comparisons of venturi master meters, with many venturi meters, conveying in sum the same quantity of water as the master meter.

Comparisons of this kind have been made for more than 25 years, always with entirely satisfactory results; and form today the basis of annual reports of water consumption in many cities and water districts.

Thus, the Metropolitan Water and Sewerage Board of Massachusetts, supplying Boston and some 20 adjacent municipalities, has 85 venturi meters in use; New York City has 64; St. Louis 68; Cleveland 57; and so on all over the world.

Like results have been found in the operation of the master meters and branch meters on the new Catskill Aqueduct supplying the city of New York, J. Waldo Smith, Chief Engineer. The master meters on this line can meter up to about 800 cu. ft. per sec.

28 It would have been gratifying to remove some or all of these crudities, and also to extend greatly the scope of the experiments, but the limits of the appropriation available forbade going beyond the work that was actually done.

29 To vary the discharge, the pump operating at a uniform speed as governed by a flywheel governor, and the boiler pressure also being fairly constant, a throttle valve in line of the pump discharge pipe was used. When the small 16-in. meter set in the 5-ft. trough was in action, the quantity and constancy of the flow leading to the meter and the efficient action of the pump, in spite of the small quantities lifted, could be regulated by opening a waste-valve out of the pump discharge pipe. To a minute extent, certainly sufficiently to remove all air from the pump discharge pipe before it reached the 30-in. meter, the same office was performed by keeping in action a 2-in. steam siphon, set at the extreme upstream end of the pump discharge pipe and ordinarily used to exhaust air from the pump before starting it.

30 The concrete channel above referred to was nominally and almost exactly 10 ft. wide, and had at its downstream end a sharp-

crested weir whose crest was about 5.35 ft. above the bottom of the channel. The new form of weir was so designed that it could be built over and around the old weir, and again removed without injury or displacement of what was found on the premises, and with its crest at nearly the same elevation above the bottom of the concrete flume.

TABLE 2 TESTS OF VENTURI METER SHOWING PERCENTAGE OF VARIATION[1]

Date	Over Per Cent	Under Per Cent	Date	Over Per Cent	Under Per Cent
July 1, 1910	1.035		Feb. 24, 1915		0.33
July 31,	0.87		Mar. —,	0.10	
Sept. 6,		0.28	Apr. 21,	0.002	
Sept. 29,		0.25	June —,	0.011	
Oct. 30,		0.25	July 18,	0.37	
Dec. 3,	0.77		Aug. 28,	0.49	
Dec. 31,		0.27	Oct. 16,	0.49	
July 8, 1911	0.93		Dec. 26,	0.23	
Aug. 3,		0.26	Jan. 30, 1916		0.47
Sept. 30,		0.49	Mar. 12,		0.57
Nov. 3,		1.19	Apr. 30,		0.40
Dec. 8,		0.73	June 11,	0.28	
Aug. 30, 1912		0.02	July 30,	0.33	
Oct. 4,		0.56	Sept. 3,		0.07
Nov. 5,		0.29	Dec. 3,		1.70
Dec. 6,		0.22	Dec. 30,	0.65	
Mar. 6, 1913	0.15		Jan. 1, 1917		1.15
Dec. 23,	0.55		Feb. 3,	0.07	
Mar. 7, 1914	0.57		June 3,	0.07	
July 10,	0.32		July 1,		1.89
Sept. 6,	0.32		Aug. 6,	0.13	
Jan. 24, 1915		0.04	Sept. 2,	1.10	
Total				9.838	11.430 9.838
Variation for 44 tests					1.592
Average, 0.036 of 1 per cent.					

[1] Tests made by F. H. Shaw, Cons. Engr., Lancaster, Pa., Water Works.

THE NEW WEIR

31 In its general outlines, the new weir, Fig. 2, was described in the *Engineering News* 1898 article above referred to. More particularly was it forecast in the Trans.Am.Soc.C.E. discussion of 1918 that has been named. So here was a case of an invention, or vision, if one pleases, in practical hydraulics, that had to wait

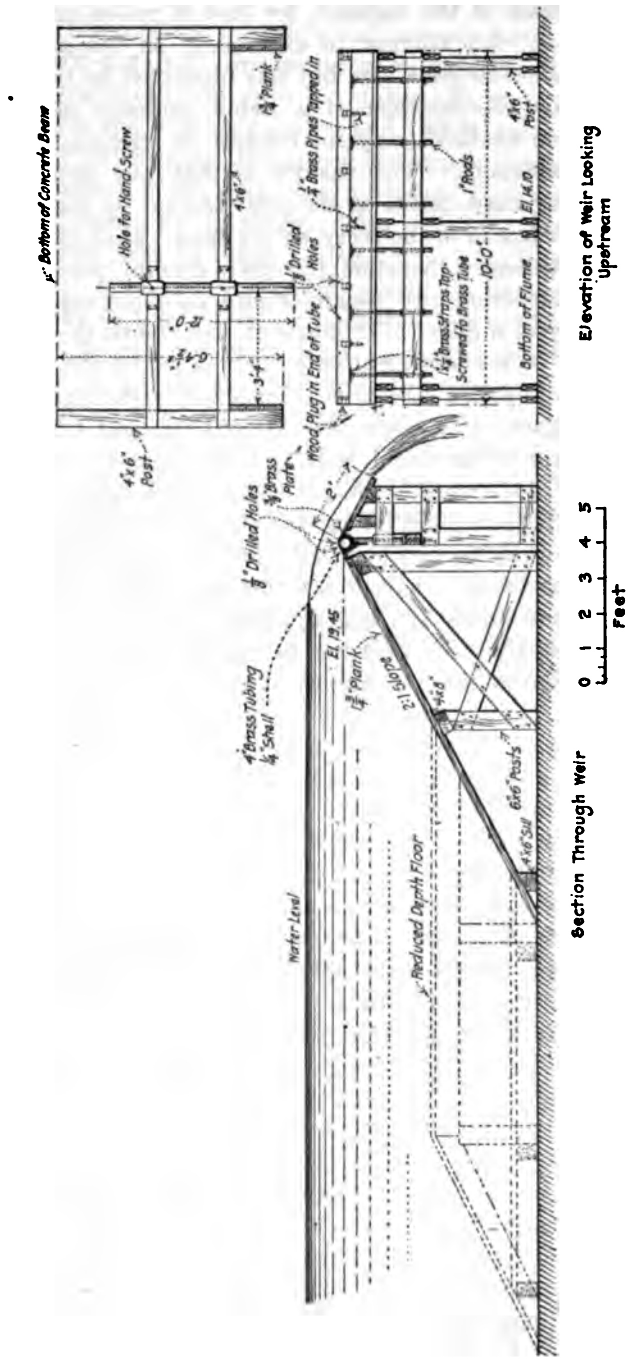

FIG. 2 THE NEW WEIR (THE SURFACE CURVE OF WATER DISCHARGING OVER THE WEIR IS DRAWN FROM MEASUREMENT.)

20 years in the brain of the engineer, for lack of public or private facilities to test it. An attempt to experiment as has now been done, was made some 10 years ago, but was frustrated by the writer being informed by the manager of a richly endowed institution that its funds were available only for research in sciences that had no practical applications. The wonder is that any progress at all is made in this most useful of the arts; treated by the general public, as was Cinderella in the story we all know; while the sisters of the family: astronomy, medicine, physics in general, pure science, even though it degenerate into pseudo-science, go about caparisoned in all the splendor of millions of endowment, and amidst the plaudits of the multitude. One endowment only exists as yet for the development specifically of applied science or engineering — the Engineering Foundation, — begun by Ambrose Swasey, of Cleveland, Ohio; and it is most gratifying to the writer to have been allowed to endeavor to illustrate some of the work of which such an endowment is capable.

32 The fundamental idea followed in the design of the new weir was to have the water to be measured conducted over the weir in a gentle manner, and so as to have it flow smoothly and regularly from the time it first encounters the weir construction, until it leaves it. Instead of allowing the body of water to impinge with more or less violence, according to the velocity with which it approaches the weir, against a perpendicular wall in its path (the upstream face of the ordinary weir) it is gently led to the crest by a 2 : 1 slope. Instead of striking or being torn over a sharp edge at the crest, the crest is made in the form of the arc of a circle; and instead of bothering about air under the nappe, the nappe is supported on another 2 : 1 slope downstream from the crest. Moreover, the crest is made hollow, so that observations of the pressure, or lack of full pressure, whichever the water may elect to exercise, can be taken *at* the crest, not at a distance upstream from the crest varying according to the fancy of the experimenter.

33 If the quantity passing the weir turn out to be a function of this observed pressure, well and good. If not, we will see what virtue there is in the *difference* of water elevations or pressures, the one taken upstream from the weir and the other taken by means of the hollow weir crest.

34 The difference referred to proved to be the sought-for solution of the problem at hand, and hereafter it alone will be closely considered. A United States patent has been applied for, covering the weir construction herein described.

35 The orifice in the side of the concrete flume conducting water to the hook-gage pail for measuring water elevations upstream from the weir was made flush with the flume side, at a point 13.9 ft. upstream from the weir crest center, and 1.68 ft. below it in elevation.

36 Furthermore, to connect the water passing over the weir with the hollow of the weir crest, a series of $\frac{1}{8}$-in. holes were carefully bored through into the latter. This was done by first tapping in a solid brass plug, and then boring a hole $\frac{1}{8}$ in. in diameter through this plug. A natural inclination would be to bore these holes at right angles to the surface of the water, but as this varies with the quantity passing the weir, such an inclination is impracticable. There are only two other natural directions for such piezometer holes, — either vertical, or at right angles to the upstream slope of the weir; and if the latter, naturally at the line of tangency of that upstream slope and the arc of the weir crest. The vertical holes are called "Orifices No. 1" and those perpendicular to the 2 : 1 slope "Orifices No. 2." The original intention of the writer had been to make the crest tube so that it could be revolved around its horizontal axis, and thus find the best position of the piezometer orifices, but a cumbersome and unduly expensive experiment like this was at once abandoned, and in view of the success already attained, may now be disregarded by engineers.

37 Both kinds of piezometric orifices were tried, and the last named proved much the superior for general use.

38 To vary the velocity of approach a false bottom was put into the channel for some of the experiments. These will be designated as having been made with "reduced depth," and others with "full depth."

39 The diagrammatic drawing, Fig. 1, will make clear the description of the weir test apparatus just given.

40 To increase the quantity per foot in length of weir, the available capacity of the pump was in many of the experiments made to pass over only about 3.3 ft. of the length of weir, as shown in Fig. 3, while other series of the experiments used a weir about 10 ft. long.

41 As these experiments had a practically useful end and aim, — the establishing of a commendable method of weir measurements, — end contractions were ruled out from the very beginning. They are of no use ordinarily and only complicate the situation. Even in the few cases of working with sharp-crested weirs, when side contractions are introduced so as to cause greater depths upon

the weir with a diminishing quantity of water or so as to cause a selected formula to apply, they could be avoided at the cost of a very

Burr A. Church, Photographer Newton Centre, Mass.

FIG. 3 SHORT WEIR IN ACTION

Leakage on each side could be measured either by the new formula; or by catching it in a pail and weighing the amount caught in an observed period of time; or by both these methods simultaneously.

little extra carpenter work. But with the straight-line formula about to be established and valid from zero up to the maximum found by the experiments herein described, there is no occasion whatever to change the length of the weir so as to accommodate the quantity of water about to pass it, and side contractions become either foolish, or pseudo-scientific. None of the following experiments includes them.

AUTOMATIC CORRECTION FOR VELOCITY OF APPROACH

42 In the photographic view of the new weir in action, Fig. 3, a vertical brass pipe dipping into the water will be noticed in the background, which is also shown in detail in Figs. 4 and 5. This was intended as a means of observing the water elevation upstream from any part of the weir, automatically corrected for velocity of approach by taking in the water pressure through a pitot orifice pointing upstream and transmitting it to a hook-gage pail in the usual manner from out of the downstream side of the vertical tube. A pointer above water, parallel to the pitot tubular orifice, which should point upstream, insures its correct position. Our experience leads us to advise that the vertical tube be not used as a reservoir, but only as a support for the pitot orifice.

43 The tube is held in its guides by two clamp screws, and may readily be set, so as to bring the pitot orifice in the locus of mean velocity as it may be judged or found. In these experiments it was set at $\frac{6}{10}$ depth of water, and $\frac{1}{3}$ the width of the channel from either side. A scale of feet and tenths on the tube, reading up from zero at the pitot orifice, and aided by another scale or gage painted on the side of the concrete flume, reading up from zero at the flume bottom, readily permitted this. The apparatus is mounted on two beams crossing the water channel and at a convenient height above it.

44 This apparatus exists as yet only in theory and should be experimented with at the first favorable opportunity, because on drawing the water out of the flume at the close of our experiments, a broken and leaky joint was discovered in the rubber pipe that carried the water pressure from the vertical tube to the hook-gage pail, so that the readings taken proved unreliable, or of no record value whatever.

NEED AND SCOPE OF FURTHER EXPERIMENTS

45 A crew of six men, including three observers, with a captain, had been working with the apparatus from September 10 to October 3, 1919, and it was with great regret that the writer gave the order to dismantle the weir because so much remained to be discovered.

46 But October 3 was Friday, the last day of the week for us. The Laboratory closed at 5 p.m.; and on Monday the regular school term commenced. The appropriation for the work was nigh exhausted and it was high time we quitted the premises.

47 There was, first of all requiring further investigation, the effect of the radius of the weir crest. Bazin had experimented on a weir composed of two 2 : 1 slopes meeting at a sharp angle and with a little pipe to indicate pressure "under the nappe," *let into*

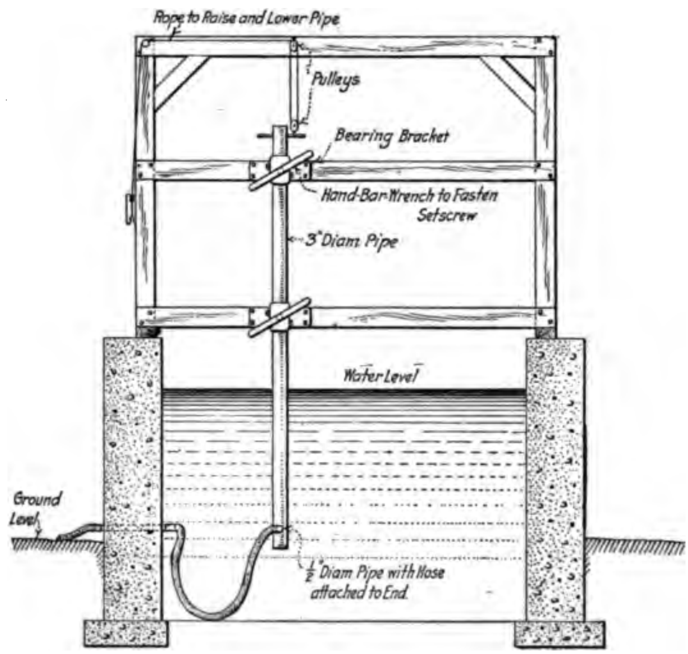

FIG. 4 APPARATUS TO OBSERVE THE WATER ELEVATION UPSTREAM FROM THE WEIR, AUTOMATICALLY CORRECTED FOR "VELOCITY OF APPROACH"

the body of the weir slope upstream from the crest. This cannot enlighten us very much, but is the nearest approach to our experiments known to the writer.

48 Our weir crest had a radius of 0.198 ft., and the outside surface was hard and smooth oil paint.

49 No doubt the radius of the weir crest is a factor in the application of the formula found for weirs of the construction shown in this article, but it remains for other sets of experiments to reveal the limits and detailed effects of this radius.

50 This illustrates the need there is for a hydraulic observatory, to be conducted on similar lines to those followed for so many centuries in astronomical observatories. There should be somewhere a hydraulic observatory whose work should go on continuously. It requires a trained and experienced crew and more time than we had to do work of full effect. The situation in the United States is much better than it was only 20 years ago when the Hydraulic Laboratory

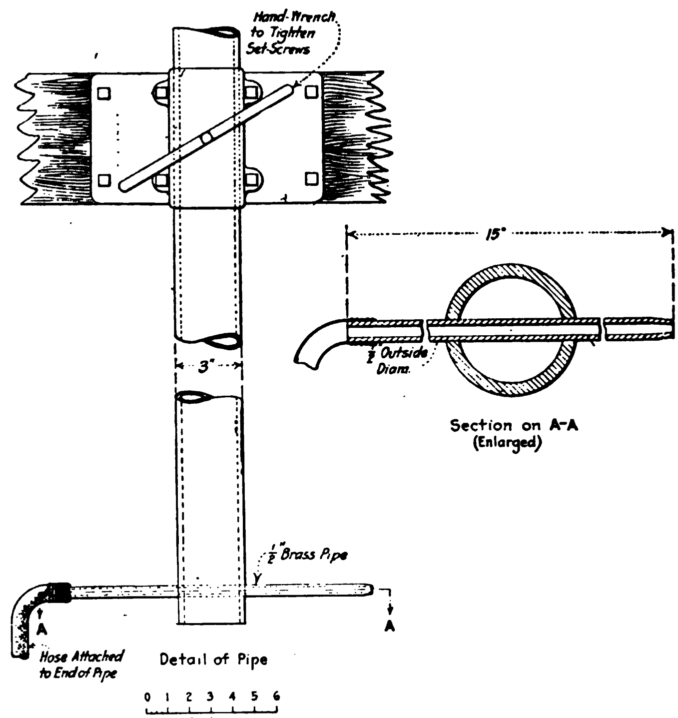

FIG. 5 DETAILS OF APPARATUS SHOWN IN FIG. 4

of Cornell University, the first one built in this country, was constructed. There are now at least half a dozen such laboratories, at Wisconsin, Illinois, Indiana, Iowa and no doubt other state universities, and others are projected. What is next needed is a hydraulic laboratory devoted to hydraulic experiments, and endowed sufficiently, so that it may bring certain aims of experiments to a conclusion and so that it may keep continuously employed the persistent efforts of trained men with a proper and permanent apparatus.

TABLE 3 EXPERIMENTS WITH THE HERSCHEL WEIR

Date 1919	Length of Run P. M.	Velocity of approach, ft. per sec.	d = Depression = Piezometer height (corrected for velocity of approach) less weir indication. As measured	Corrected for v. of a.	Q = Quantity per ft. of weir length, cu. ft. per sec.
Sept. 10[1]	2.45-2.50	0.70	1.332	1.340	4.53
	4.00-4.05	0.58	1.182	1.187	3.82
	4.30-4.35	0.48	0.991	0.995	3.06
	4.55-5.00	0.33	0.700	0.702	2.03
Sept. 11[1]	1.50-1.55	0.24	0.522	0.522	1.49
	2.20-2.25	....	0.347	0.347	0.95
	2.50-2.55	....	0.279	0.279	0.75
	3.20-3.25	0.24	0.508	0.508	1.47
	3.55-4.00	0.45	0.962	0.965	2.87
	4.45-4.50	0.62	1.240	1.246	4.07
Sept. 15[2]	2.15-2.20	1.24	1.319	1.349	4.48
	3.00-3.05	0.92	1.001	1.014	3.14
	3.50-3.55	0.72	0.790	0.798	2.34
	4.25-4.30	0.50	0.529	0.533	1.53
	4.55-5.00	0.30	0.305	0.307	0.85
Sept. 17[3]	2.30-2.35	1.62	2.683	2.723	12.53
	3.05-3.10	1.39	2.279	2.309	10.18
	3.40-3.45	1.00	1.725	1.741	6.87?
	4.20-4.25	0.63	1.229	1.237	4.15?
	4.50-4.55	0.40	0.864	0.868	2.57
Sept. 18[4]	1.55-2.00	2.87	2.603	2.733	13.38
	2.30-2.34	2.60	2.374	2.484	11.53
	3.05-3.10	2.01	1.915	1.977	8.31
	3.50-3.55	1.28	1.269	1.294	4.63
	4.35-4.40	0.75	0.773	0.782	2.45
Sept. 25[4]	3.10-3.15	0.40	0.478	0.482	1.33
Sept. 26[4]	1.43-1.45	0.47	0.491	0.495	1.42
	3.05-3.10	0.25	0.265	0.265	0.71
	4.10-4.15	0.13	0.120	0.120	0.36?
	4.40-4.45	....	0.101	0.101	0.24?
Sept. 29[5]	3.15-3.20	0.42	0.249	0.253	1.3
	3.55-4.00	0.29	0.194	0.195	0.87
	4.25-4.30	0.18	0.149	0.149	0.51
	4.45-4.50	0.18	0.042	0.042	0.49
	5.05-5.10	0.08	0.022	0.022	0.21
Oct. 1[5]	1.30-1.35	2.22	1.658	1.738	9.55
	2.10-2.15	1.90	1.189	1.249	6.91
	3.10-3.15	1.34	0.865	0.895	4.93
	3.45-3.50	0.92	0.581	0.593	3.14
	4.20-4.25	0.73	0.438	0.447	2.42

[1] Long weir, full depth.
[2] Long weir, reduced depth.
[3] Short weir, full depth.
[4] Short weir, reduced depth.
[5] Short weir, reduced depth, No. 2 orifices.

51 An example of the opposite method of proceeding is furnished by the experience of engineers with the venturi meter. Invented in 1886, more than 30 years ago, only the few experiments shown in Table 1 are as yet available to portray the laws that govern its action. It has accomplished much in the course of its life of 30 years, but it could have accomplished more, and that sooner, had there been a hydraulic observatory to teach its laws and qualities. During the first ten years of its life it was used only by its inventor, which is another illustration of the need of experiments which the working engineer cannot be expected to have opportunity to make in such cases.

THE EXPERIMENTS

52 With these preliminary statements, the record of the 40 experiments made is given in Table 3. Four of them have been marked doubtful on account of actual or supposedly defective observations and all those made September 9 and 30, also part of those on September 25, had to be rejected as worthless on account of the previously mentioned violent "surging" of the water discharged by the pump. Note that the strait line formula applies only to the experiments made Sept. 29 and Oct. 1. See also Fig. 6.

53 A separate set of gagings was made October 3 to compare the indications of the Institute's sharp-edged weir with the Institute's 30-in. venturi meter. Owing to the fact that this meter is set in line of the pump discharge pipe, only 12 ft. distant from the pump, it had been a question whether at that short range it would meter accurately, and the experiments of October 3 were made for the purpose of answering this question. The system available for observing head on the weir, was that known as the Fteley and Stearns method. A metal plate was set in the side of the concrete flume, flush with it, and an orifice in this plate, in the present instance, was situated 6.8 ft. upstream from the crest, and 1.68 ft. below it in elevation, connected with a hook-gage pail in the usual manner.

Record

Pump started about 12.30 p.m., discharging 30 cu. ft. per sec. at 1 p.m. All quantities corrected for velocity of approach.

	Weir	Meter
	Cu. Ft. per Sec.	
Oct. 3, 1919, 2.05 to 2.10 p.m., both inclusive	38.13	38.30
2.50 " 2.55 " " " "	30.33	31.25
3.50 " 3.55 " " " "	17.91	18.80
4.40 " 4.45 " " " "	11.73	11.00

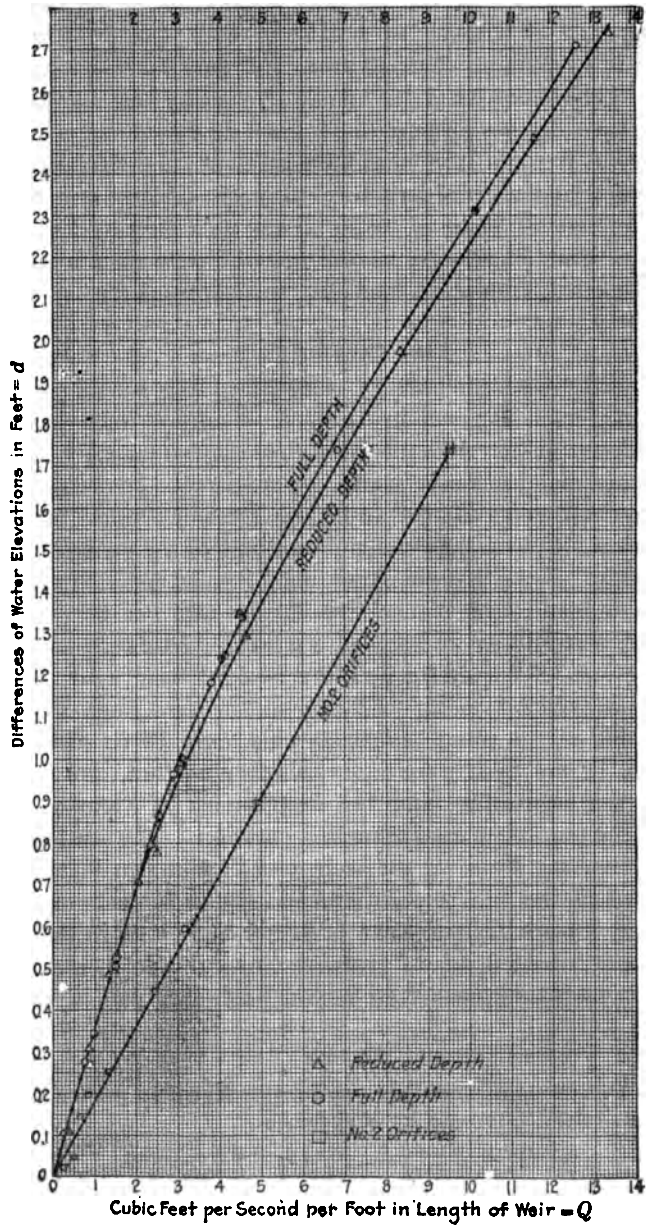

FIG. 6 DIAGRAM SHOWING THE RESULTS OF THE 40 EXPERIMENTS MADE

54 Having in mind the imperfections of the whole test apparatus from the canals in the basement, through the pump, riser, 5-ft. trough, 12-ft. vertical flume, wooden flap valve, into the 10-ft. concrete flume, and over the weir, as already mentioned, it is quite possible that 38.13, 30.33, 17.91, and 11.73 cu. ft. per sec. were passing the weir on the average for 5 min., while 38.30, 31.25, 18.80, and 11.00 cu. ft. per sec. were being discharged by the pump on the average, during the same 5 min. The difference could very well represent rise or fall in the mean elevations of the water in either the 5-ft. trough, the 12-ft. vertical flume, or in the 10-ft. concrete flume, or in two or more of these containers simultaneously.

55 The comparison which was made, taken by itself alone, does not argue for a superior exactness of gaging by either the weir or the meter — that must be determined by tank tests made with the two separately. It only showed that neither was materially inexact. And we know from data that have been given above, that the venturi meter gives water measurements closely approximating the exactness of tank gagings. And with the result placed in evidence, that No. 2 orifices act according to the straight-line equation —

$$Q = 5.50d \text{ in English units,}$$
$$Q = 1.675d \text{ in metric units,}$$

this paper might fitly close.

FORMULÆ

56 One word more, however, the writer allows himself in relation to formulæ for weir discharge. It had been his intention not to evolve any formula, but to present the result of his experiments in the form of a table, arranged somewhat as are tables of logarithms, so that quantities could always be read at a glance from the data observed. No one cares for the formulæ by means of which logarithms are computed; their use would not be furthered by a study of them. No more need a Power Test Code contain formulæ for weir discharge; a table would answer its purposes more fittingly.

57 But the exceeding simplicity of the weir formula appurtenant to the form of weir herein presented upset this plan. Such a weir does not need so much as a table to look up or to compute its discharge. All one has to do is to measure the length of the weir and multiply by 5.5 and the difference in head.

58 But there remains the question, why 5.5? Why not 20, or any other quantity? And for the mitigation of the stated laudable curiosity, further experiments will no doubt be undertaken in the course of time; and much ink of disquisition may yet be set adrift.

DISCUSSION

Arthur M. Greene, Jr., asked the author to supply a set of data so that he might check the computations and also asked if the author would object to experiments made with his weir by the colleges. If the formula arrived at by the author held, he considered that a great advance in the measurement of water had been made.

The author claimed that the location of hook gage in the standard form of weir is not definitely settled, yet he did not give reasons for selecting the position of the hook gages in his own weir.

R. L. Sackett. Condemnation ought not to be made of the old type of weir but rather of the lack of standardization in the location of the gages and of the type of gages.

One is surprised in looking over the results so far as they are given in the paper that the coefficient of the weir should remain constant for wide variations in head, as this is not true of any other horizontal weir.

This is not the only weir in which the discharge varies directly with the head. Laboratory tests have shown that certain sharp-crested, curved weirs in a vertical plane have constant coefficients of discharge. Such weirs as the author's are, perhaps, better for measuring large quantities of water, but in the measurement of wide variations of flow such as of the return water in a power plant the sharp-crested weir would be of great advantage.

C. G. Richardson. The Herschel Weir might be considered as a special conduit or flume constructed along the lines of a Venturi meter tube but not adhering to the usual inlet and outlet tapers. One would therefore expect that the discharge over such a weir would vary as the one-half power of the difference in head between the approach and the crest (corresponding to the inlet and throat portions of the Venturi tube), but it might be considered plausible that the departure from that established formula is due to the departure from the stream line of the true Venturi, as the water flows over the weir crest, and that this departure is just sufficient to change the exponents of the differential head from one-half to unity.

The Author. In reply to Professor Greene the author wishes to say that he will be only too glad to have others repeat, and also extend, his experiments with this particular form of hollow crest weir. Such permission has on their suggestion already been given to the State University of Iowa, and to the Société Hydraulique de France, located at Grenoble, France, and the author is glad to give it in addition public announcement. Arrangements have also been made for manufacturing such strait-line formula, hollow-crested weirs in the United States, in Great Britain and in France,[1] so that such weirs may now readily be constructed of wood or concrete. It may be noted that hook gages are not a necessary part of the outfit for gaging water with this form of weir. Two glass tubes, side by side, and measuring how much higher the water stands in one tube than it stands in the other, by means of a *uniformly* graduated movable, or even portable, pocket scale, is what the final metering part of the apparatus resolves itself into. The one tube is fed by the hollow crest of the weir; the other is fed by a tube connection from the main flume, naturally opening into that flume just upstream from the upstream toe of the 2-on-1 slope that runs from the flume bottom up to tangency with the weir crest.

Taking up Professor Sackett's remarks, attention should be called to the fact that the "crested, curved weirs" referred to are in fact not weirs in the ordinary acceptation of the term, but crescent-shaped, sharp-edged orifices.

The choice now presented is between a complex weir formula of discharge, with widely varying coefficients as they are affected by a numerous variety of circumstances, and a single coefficient strait-line formula, $Q = 5.5\ d$, in which only d is to be read by observation "in the field"; between weirs that call for a material loss of head, so as to "keep air under the nappe," and a weir needing a much less loss of head, and having no air whatever under the nappe; between a weir that lends itself readily to having its results

[1] By Builders Iron Foundry, Providence, R. I.; George Kent, Limited, Biscot Road Works, Luton, England; and Etablissements Bouchayer et Viallet, Grenoble, France, respectively.

mechanically or graphically registered and recorded because it operates with, and indicates, a *uniformly graduated* scale of quantities, and weirs such as they have hitherto been, which weirs may perhaps not need description to weir users, in this place, or elsewhere.

No. 1745

SIMPLIFICATION OF VENTURI-METER CALCULATIONS

By Glenn B. Warren,[1] Schenectady, N. Y.
Non-Member

In this paper the author describes a method which he has devised for the simplification of the calculations involved in the venturi-meter measurement of the flow of compressed air, and which reduces the work of one determination to a simple slide-rule computation requiring but a few settings. The method is based on the venturi-meter formula as given by Herbert B. Reynolds in Trans.Am.Soc.M.E., Vol. 38, p. 799, and can be easily applied to any venturi meter used for the measurement of a gas.

DURING the winters of 1918 and 1919 the writer had occasion to measure with considerable accuracy the amount of compressed air used in connection with some research work which was being carried on at the University of Wisconsin. The venturi meter presented itself as the logical method, but aside from the mechanical difficulties of building manometers to satisfactorily stand pressures of more than 100 lb. per sq. in. the difficulties presented by the calculations where several hundred volume determinations would have to be made were of considerable importance. Accordingly, a method of shortening the calculations was devised which reduced the entire work of one determination to a simple slide-rule computation, requiring but a few settings, and which at the same time did not affect the accuracy to any appreciable extent. Although the method of shortening these computations may seem somewhat laborious in itself, when it is remembered that with but a few changes it can be applied to any venturi meter used for the measurement of a gas, the result can be seen to have justified the means.

2 The venturi-meter formula is given by Dr. Lucke as follows:

[1] 25 North Ferry Street.

Presented at the Spring Meeting, St. Louis, May 1920, of The American Society of Mechanical Engineers.

$$W = CA_2\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{2g\frac{n}{n-1}\frac{P_1}{V_1}\left[\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}\right]} \quad . \; . \; . \; [1]$$

where W = weight of vapor or gas passing per second
C = coefficient
A_1 = area of entrance
A_2 = area of throat.

The units in which W is expressed are dependent, of course, upon the units used in all the other quantities of the formula.

3 Herbert B. Reynolds in Trans.Am.Soc.M.E. for 1916, p. 799, gives the following formula, which can be derived from Equation [1] by the substitution of the proper units:

$$Q = C\frac{3514\,A_2T_s}{P_s}\sqrt{\frac{n}{(n-1)G}}\left(\frac{P_1}{\sqrt{T_1}}\right)\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$\times\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [2]$$

where Q = cu. ft. per min. at 32 deg. fahr. and 14.7 lb. per sq. in.
C = coefficient (experimentally determined)
A_1 = area of entrance, sq. ft.
A_2 = area of throat, sq. ft.
T_s = absolute temperature of standard air = 492 deg. fahr.
P_s = absolute pressure of standard air = 14.7 lb. per sq. in.
n = constant = $\frac{C_p}{C_v}$ (1.402 for air)
G = specific gravity of gas (1.00 for air)
P_1 = pressure of air at entrance, lb. per sq. in. absolute
P_2 = pressure of air at throat, lb. per sq. in. absolute
T_1 = temperature of air at entrance, deg. fahr. absolute.
3514 = constant determined by above choice of units.

4 The *Sibley Journal of Engineering* (vol. 29, pp. 90–95) presents this formula in still another form, which has been derived by means of the infinite series, and is in better shape for calculation where the ratio (P_2/P_1) is very nearly unity or, in other words, for low rates of flow. This formula is:

$$W = CA_2\left(\frac{T_s 2gn}{V_s P_s(n-1)}\right)^{\frac{1}{2}} \frac{P_1}{\sqrt{T_1}}\left(x^{\frac{1}{2}}\left[\frac{n-1}{n(1-a)}\right]^{\frac{1}{2}}\right)$$
$$\times\left(1 - \frac{[3+a]x}{2n[1-a]}\right)^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where V_s = volume under standard conditions, 12.38 cu. ft. per lb. for air; $a = (A_2/A_1)^2$; and $x = \Delta P/P_1$, where $\Delta P = P_1 - P_2$; the other symbols being the same as before.

5 Any one of these three formulæ presents a rather difficult computation as far as solution is concerned, this being especially the case in formulæ [1] and [2] because of the last radical. The log-log slide rule enables the solution to be made quite readily, excepting for values of P_2/P_1 greater than 0.98, after which it is necessary to resort to very complete logarithmic tables, or else to the third formula.

METHOD USED TO SIMPLIFY COMPUTATIONS

6 The first step in the process of reducing these formulæ to forms which could easily be handled consisted in determining the values of A_1 and A_2 which would give a venturi meter having the desired range of capacity without making the difference between P_1 and P_2 either too great or too small for easy and accurate reading. In this series of tests it was necessary to have a range of capacity from 25 to 250 cu. ft. of free air per min. at a pressure in the venturi tube of about 100 lb. per sq. in. gage. Inasmuch as the flow is nearly proportional to the square root of the pressure difference, as can be seen from formula [3], a range of possible rates of flow of 10 to 1 means a range of readable pressure differences of nearly 100 to 1 in magnitude. This necessitated the combination of a mercury and a water manometer, which will be described later.

7 It was found by a trial and error method that a venturi meter having an entrance diameter of 1.6 in. and a throat diameter of 0.5 in. would answer the requirements. As actually constructed, the throat diameter was found to be 0.498 in. This gave a ratio of A_2/A_1 nearly equal to 0.10.

8 Upon substituting these numerical values for A_1, A_2, P_s, n, and G in formula [2], there is obtained the following expression:

$$Q = 297.6C\left(\frac{P_1}{\sqrt{T_1}}\right)\left[\left(\frac{P_2}{P_1}\right)^{0.714}\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{0.286}}{1-0.01\left(\frac{P_2}{P_1}\right)^{1.43}}}\right] \quad . \quad . \quad [4]$$

9 It can be seen that the part in brackets is independent of everything but the ratio P_2/P_1; in other words, it is a function of this ratio. Then if we let

$$f\left(\frac{P_2}{P_1}\right) = \left[\left(\frac{P_2}{P_1}\right)^{0.714}\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{0.286}}{1-0.01\left(\frac{P_2}{P_1}\right)^{1.43}}}\right] \quad . \quad . \quad . \quad . \quad [5]$$

and plot a curve of $f\,(P_2/P_1)$ as ordinates, against values of P_2/P_1, it follows that the finding of Q will merely amount to multiplying

TABLE 1 VALUES OF $f(P_2/P_1)$ FOR VALUES OF P_2/P_1 FROM 1.00 TO 0.90

$\frac{P_2}{P_1}$	$f\left(\frac{P_2}{P_1}\right)$	$\frac{P_2}{P_1}$	$f\left(\frac{P_2}{P_1}\right)$
0.90	0.1600	0.97	0.0917
0.91	0.1534	0.98	0.0753
0.92	0.1456	0.985	0.06535
0.93	0.1369	0.99	0.0535
0.94	0.1276	0.995	0.0370
0.95	0.1170	0.9975	0.02688
0.96	0.1057	1.0000	0.00000

several known quantities on the slide rule, since $f\,(P_2/P_1)$ can be found from the curve, thus:

$$Q = C\frac{297.6P_1}{\sqrt{T_1}}f\left(\frac{P_2}{P_1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

10 In practice it is the quantity $P_1 - P_2 = \triangle P$ which is measured by the manometers, and the most accurate method of finding P_2/P_1 from this is as follows:

$$\frac{P_2}{P_1} = \frac{P_1 - \triangle P}{P_1} = 1 - \frac{\triangle P}{P_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

It can easily be seen that this last expression can be evaluated much more accurately on the slide rule than can the second.

11 The values of $f\,(P_2/P_1)$ for values of P_2/P_1 from 1.00 to 0.90 are given in Table 1 and are shown plotted out in the form of a curve in Fig. 1. The values of $f\,(P_2/P_1)$ for values of P_2/P_1 between 0.98 and 1.00 were found by evaluating the last two parentheses in formula [3]. It is rarely advisable to use a gas venturi meter at values of P_2/P_1 less than 0.90, inasmuch as the total drop

in pressure across the meter becomes too great, and also because the reserve or overload capacity of the meter is so greatly restricted. This set of values is worked out for a ratio of $A_2/A_1 = 0.10$, but the effect of a slight change in this ratio upon the values of $f\,(P_2/P_1)$ is so slight that these values may be used for meters having a ratio differing considerably from 0.10. However, a set of values can be worked out for a tube having a ratio other than this.

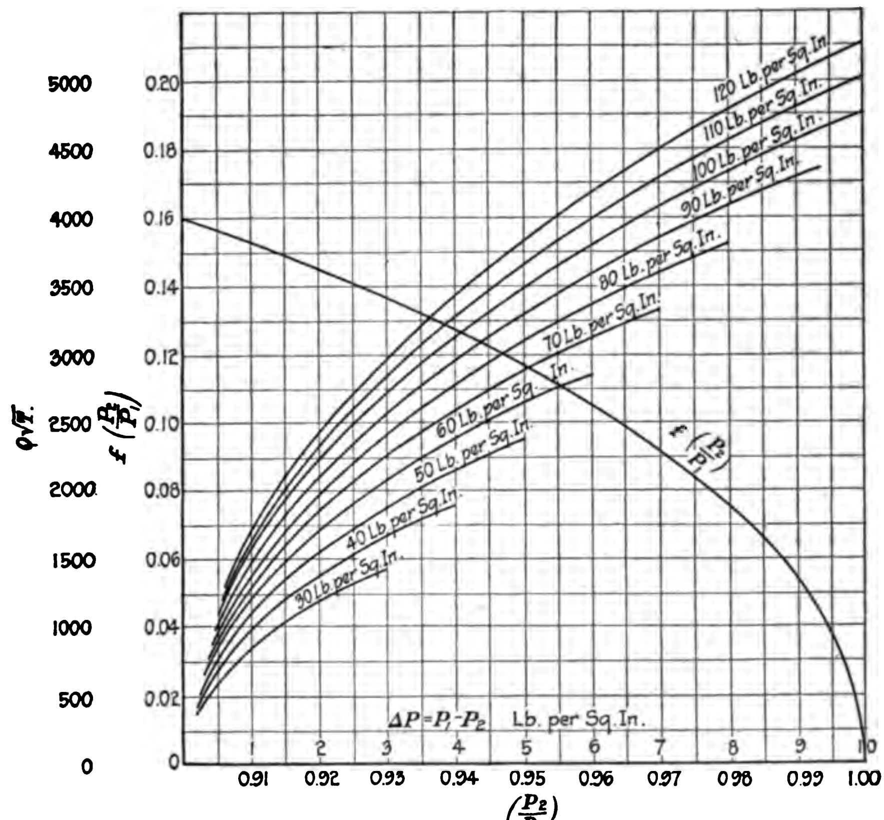

Fig. 1 Flow Curves for 1.6-in. × 0.498-in. Venturi Meter

12 The family of curves shown in Fig. 1 was worked out to facilitate rapid slide-rule calculation of the flow during a test. In order to plot these, the temperature in formula [6] was transposed to the left-hand member, thus:

$$Q\sqrt{T_1} = C297.6P_1 f\left(\frac{P_2}{P_1}\right) \quad . \; . \; . \; . \; . \; . \; . \quad [8]$$

and curves for numerous values of P_1 plotted against values of $P_1 - P_2$, that is, actual manometer readings. Interpolation could then be resorted to, and by dividing the value of $Q\sqrt{T_1}$ found in the ordinate by the square root of the inlet temperature the flow in cubic feet of free air per minute could be obtained. The accuracy would depend upon the care taken in the interpolation. It was generally within 2 per cent, however, which was sufficient to answer the purpose for which this set of curves was devised.

CALIBRATION

13 After consideration and trial of several methods of calibration, it was decided to check the meter against thin-plate orifices, using the empirical formula

$$Q = \frac{405A\ (P_1^2 - P_2^2)^{0.48}}{\sqrt{T}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

where Q and T are the same as before; A is the area of the orifice in sq. in.; P_1 is the pressure in lb. per sq. in. abs. in front of the orifice; and P_2 is the pressure in the same units beyond the orifice.

14 This formula, like formula [2], is due to Mr. Reynolds, and has been checked by him by means of a large number of very accurate experiments. Three orifices were used, having diameters of $\frac{3}{8}$ in., $\frac{1}{4}$ in., and 0.191 in., respectively. All the plates were $\frac{1}{16}$ in. in thickness and made of brass.

15 The results of this calibration are shown in Fig. 2. The coefficient seems to decrease with an increase in flow. This is not in accordance with the results obtained by several other investigators, but the orifices were later checked up and further runs made with substantially the same results. Another calibration was made after more than a year of operation of the meter, and, despite some slight corrosion and roughening of the walls, the coefficient was found to be very nearly the same. It is the writer's opinion that for most experimental work one would be very safe in assuming the coefficient of a carefully constructed venturi meter when metering compressed air to be practically unity, providing the ratio P_2/P_1 was not less than 0.90.

MANOMETERS

16 As stated before, it was necessary to use two manometers, one filled with water and the other with mercury. This was neces-

sary in order to read accurately values of ΔP from about 11 lb. per sq. in. to 0.10 lb. per sq. in. A valve and piping arrangement with which it was possible to quickly change from one manometer to the other while under pressure, and at the same time keep the liquid in the manometers from being blown out, is shown in Fig. 3. Its operation is apparent. It is necessary, however, to be sure that the valves connecting the two legs of each monometer are *open* when making any change of valves Nos. 1, 4, or 5, or whenever valves Nos. 4 and 5 are closed, and also that valve No. 2 remains *open* at all times when the water manometer is not in use.

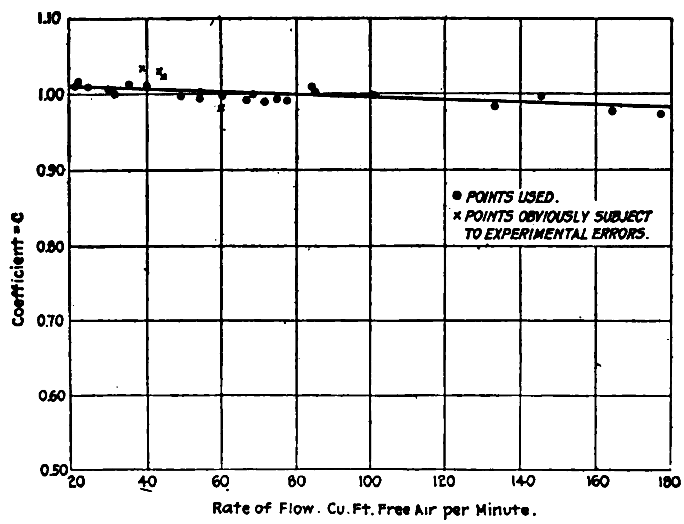

FIG. 2 CALIBRATION CURVE FOR 1.6-IN. × 0.498-IN. VENTURI METER

17 It was found that rubber washers cut from soft sheet rubber made the most satisfactory packing for the stuffing boxes on the air, water, and mercury ends of the manometers. Brass stuffing-boxes and U's were used, excepting on the lower end of the mercury manometer, where wrought iron was used. Glass tube 3 mm. outside and 1 mm. inside diameter was found to be satisfactory. At first an attempt was made to make the manometers from a single piece of glass bent into shape, but it was discovered that the internal strains set up by the heating were so great as to render the tubes unreliable when used at this pressure.

18 In the final set-up it was necessary to replace the ⅛-in. piping which connected the venturi meter with the manometers with

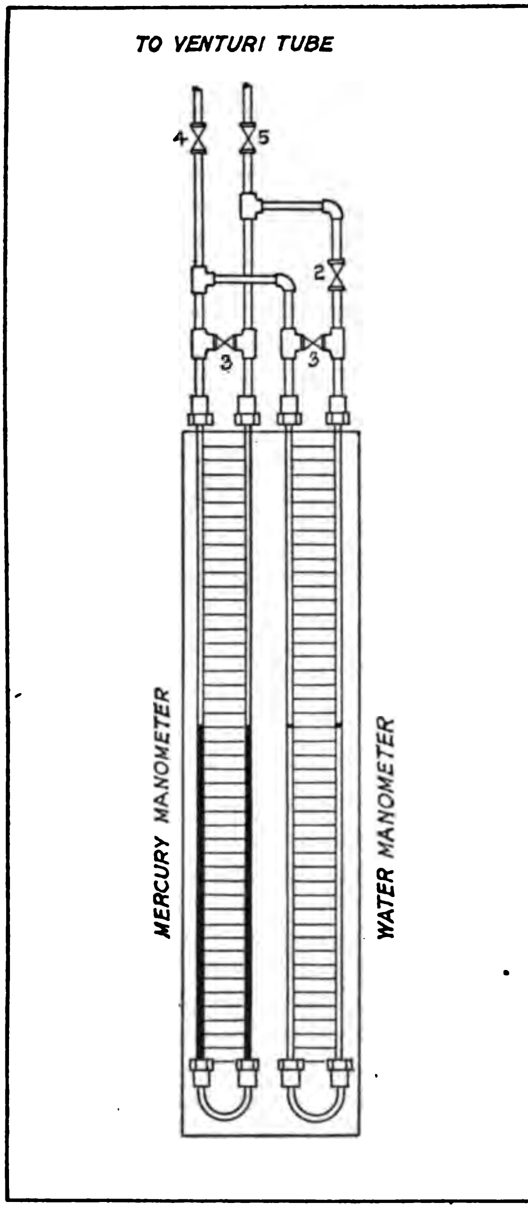

FIG. 3 ARRANGEMENT OF MANOMETERS, PIPING, AND VALVES

½-in. piping, and to provide small water traps or pockets at the point where these pipes were connected to the venturi meter. This was done in order to prevent water, which was frequently in the compressed air, from working its way over into the piping and even into the manometers, and thus rendering the readings inaccurate. A large number of experimental runs, otherwise perfect, were spoiled before this fact was discovered.

19 The total drop in pressure across this venturi tube for rates of flow under 100 cu. ft. per min. was only a fraction of a pound per square inch, while at 200 cu. ft. per min. it was only about 2.5 lb. per sq. in. when P_1 was 100 lb. per sq. in. gage. At rates of flow greater than this, however, the drop increased rapidly.

DISCUSSION

THOS. G. ESTEP. Equations [1] and [2] which the author credits to Dr. Lucke and Mr. Reynolds are simply derived from the fundamental equations for the frictionless, adiabatic flow of elastic fluids so that credit is really due to the older investigators.

The form which the fundamental equation takes for the velocity relation is as follows:

$$W_2^2 - W_1^2 = 2g\frac{n}{n-1}RT\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]$$

This equation applies to the Venturi meter as well as to orifices and short tubes, W_1 being the initial velocity and W_2 the velocity at the smallest section.

In the November 9, 1916 issue of *Iron Age* the writer presented an article on the simplification of the solution of the above equation by means of a curve. The abscissa of this curve was the ratio $\frac{P_2}{P_1}$ and the ordinate was the coefficient K, the latter being,

$$K = 2g\frac{n}{n-1}R\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]$$

Then for any given ratio of $\frac{P_2}{P_1}$, the value of K could be read directly. The original equation then becomes

$$W_2^2 - W_1^2 = KT$$

If this equation is applied to the Venturi meter, W_2 will be the throat velocity and W_1 the upstream velocity. Eliminating W_1 from the equation gives the following:

$$W_2 = \sqrt{\frac{KT}{1 - \left(\frac{A_2}{A_1}\right)^2 \left(\frac{V_1}{V_2}\right)^2}}$$

The curve sheet mentioned also included a curve giving the values of $\frac{V_2}{V_1}$ for various values of the pressure ratio. In the above equation, for any given pressure ratio, all of the values can be read from the curve except T and the area ratio, and in a given set of experiments, the area ratio will be constant and the inlet temperature nearly so. Knowing the throat velocity, the volume or weight flowing can be calculated. The entire solution is reduced to a simple slide-rule calculation.

It is suggested that if it is desired to simplify the calculation of the volume of gas flowing through a Venturi tube that instead of the upstream static opening, an impact tube be used, retaining the static opening at the throat. The differential pressure then represents the total velocity at the throat instead of simply the change in velocity as when the two static openings are used. (See The Impact Tube by Sanford Moss, Trans.Am.Soc.M.E.1916.)

The calculation can be further simplified if the size of the Venturi tube is such that the total differential pressure is small, say not over 4 in. of water. Then the change in the specific volume of the gas as it flows through the tube can be neglected and the throat velocity becomes

$$W_2 = \sqrt{2gh}$$

h being equivalent feet of gas corresponding to the differential pressure as measured.

It is a surprise to the writer that the author should select the throttle disk or thin plate orifice as a standard for the calibration of a Venturi meter.

The coefficient for a trottle disk is a combined coefficient which takes care of the contraction and the velocity of approach. It has not been proven by experiment but let us assume that for a given ratio of orifice diameter to pipe diameter that the contraction is constant for all rates of flow. Then as the flow increases the velocity of approach increases so that the total coefficient for the disk should be an increasing coefficient instead of a constant one. This explains why the author found that the coefficient of the Venturi meter decreased with the flow, a result contrary to other experiments.

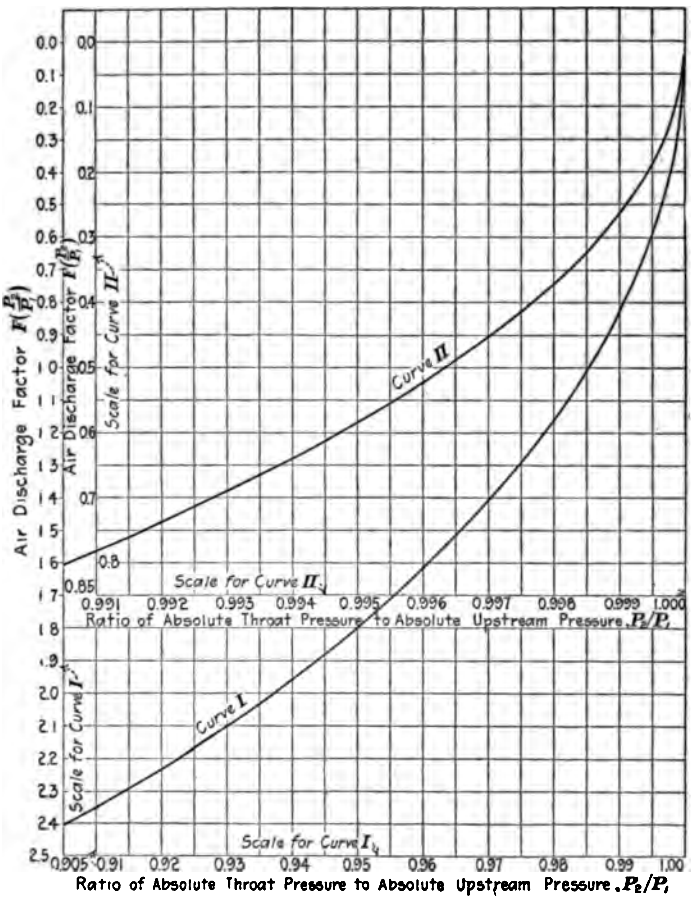

FIG. 4 CURVES SHOWING VALUES OF THE FUNCTION P_2/P_1.

It is the writer's opinion that the author's calibration shows that Mr. Reynolds' empirical equation for a thin plate orifice is a good one, rather than proving the accuracy of the Venturi meter.

M. C. STUART. Venturi meters have been used at the U. S. Naval Engineering Experiment Station, Annapolis, Md., for a number of years with much success in the measurement of air in fan tests.

Several years ago a method of simplified calculations almost identical with that presented in this paper was developed by the writer.

The principal calculations are shown on the enclosed tabulation. The values of P_2/P_1 raised to the various powers were worked out by the use of ten-place logarithms and a great degree of accuracy thus obtained. The function of P_2/P_1 in this case was made to include the term $\frac{2gn}{n-1}$ in addition to the portion of the formula given by the author.

Comparison of the values of the function of P_2/P_1 for values of $n = 1.40$ and $n = 1.41$ show only a minute difference; the error in assuming $n = 1.41$ being less than four one-hundredths of 1 per cent for a differential head of 40 in. of water, and decreasing with smaller differentials. The per cent of error in the value of the functions of P_2/P_1 for 1 per cent of error in the ratio of P_2/P_1 is about three one-hundredths of 1 per cent, thus showing that it is not necessary to determine the ratio of diameters very accurately, although the value of the throat diameter must be determined accurately. The function of P_2/P_1 is plotted in a large scale in Fig. 4. To obtain additional accuracy for similar differential heads, the values of the function P_2/P_1 from 0.99 to 0.9995 have been plotted on a separate curve. By means of these curves extreme accuracy may be obtained in computing theoretical flow from the exact formula.

Calibration of the Venturi meter made by comparing with the flow as measured with a standard orifice having parallel sides and a curve entrance, as described by Sanford A. Moss,[1] gives coefficients for the Venturi meter of 0.98 to 1.0.

Chas. G. Richardson. The author's method is based on the Venturi-meter formula as given by Herbert B. Reynolds, in Trans. Am.Soc.M.E., vol. 38, p. 799. In using this or any other modification or previously devised formulæ it is wise to check at least one point in the problem by a true formula, as some of the less complicated formulæ are limited in their application.

In 1898 F. G. Gasche, of the Illinois Steel Company, developed the thermodynamic formula for measuring gas and steam through the Venturi meter. Nine years later E. P. Coleman, in a paper entitled The Flow of Fluids in a Venturi Tube,[2] gave a formula equiva-

[1] The Impact Tube, Sanford A. Moss, Trans.Am.Soc.M.E., 1916, p. 761.

[2] Trans.Am.Soc.M.E., vol. 28, p. 483.

lent to the author's Equations [1] and [2]. The method which the author has independently worked out of handling what might be termed the most disagreeable factor of the equation is the same method which has been employed by several others. For instance, Mr. Gasche developed an elaborate series of curves and diagrams for calculating gas and steam flows through Venturi meters and among these were curves involving the ratios of the absolute pressures between throat and inlet, also specific heats, corresponding to the argument employed by the author in the solution of Equation [5]. In 1910 Beebe and Herrick in experiments on the measurement of air through a Venturi meter at the University of Wisconsin also reduced this factor to a curve.

It is probably the author's intention to confine his various conclusions to the results obtained with tests on a single small Venturi meter; for a more universal application, the following exceptions should be noted:

a There is no difficulty commercially in making manometers to withstand working pressures exceeding 100 lb., those on the market being adapted for pressures up to 250 lb. or even higher.

b A friction loss of 2.5 lb. through a 1.6 in. by 0.5-in. Venturi-meter tube under 100-lb. gage pressure and discharging air at the rate of 200 cu. ft. per min. is larger than would be obtained by a Venturi tube of standard proportions, which would show about 1.5 lb. at this rate. It would therefore seem that the Venturi tube employed in the test was of some special design, that the downstream piezometer was placed too near the outlet end, or that the loss in pressure was incorrectly indicated. Tests conducted by the Builders Iron Foundry of Providence, R. I., in 1911 on a 2-in. by 1-in. Venturi-meter tube measuring air showed the friction loss to be about 1/5 of the difference in pressure between the inlet and throat. Beebe and Herrick in their tests on a 2-in. by ½-in. Venturi tube also show the friction loss to be a straight-line function of the differential between inlet and throat and give the value as $F = H/5.467$ where F is the friction loss and H is the differential pressure. In a series of comparative tests made by W. F. Schell in July 1914 [1] between various types of meters measuring natural gas including 4-in., 10-in., and 12-in. Venturi meters, it was found that "the ratio of the differential pressure to the pressure loss through the venturi meter is approximately eight to one." Messrs. Bacon and Moulson, in a comprehensive series of calibrations at the Illinois

[1] Proc. Natural Gas Assn. of America, vol. 8.

Steel Co. on Venturi meters 24 in., 36 in. and 48 in. in diameter, found the drop in line pressure caused by the insertion of the meter tube to be $F = H/9$, the outlet pressure being observed about one and one-half diameters beyond the outlet end of the meter tube. Tabulating these results we have

Size of meter tube inlet and outlet diameter, in.	Friction loss equals differential between inlet and throat divided by
2	5
4 to 12	8
24 to 48	9

This shows conclusively that the author's determination of friction loss was not representative.

c For experimental work the reduction of the author's tabular arrangement of Equation [5] to curve form, using a coarse scale, would be valuable. A wider application to Venturi meters in general would involve a large family of curves due to the variations which occur in the values A_2/A_1, n, and P_2/P_1. For instance, for the measurement of gas each size of standard Venturi-meter tube has four throats, the diameters of which are approximately 3/12, 4/12, 5/12 and 6/12 of the inlet diameter. For n there would be a variation from 1.402 for air to 1.135 for dry saturated steam. P_2/P_1 might go much below the limit of 0.90 mentioned by the author; for instance this value becomes 0.8454, assuming 50-lb. gage pressure at the inlet of a Venturi meter tube connected to a mercury manometer indicating a differential of 10 lb. The wide range in the value of $f(P_2/P_1)$ for such changes in value of the terms under the radical sign is illustrated by the following table. Since this function enters as a factor in the final equation any variation in its value involves an equal variation in the calculated weight or quantity discharged through the Venturi meter.

$\frac{A_2}{A_1}$	n	P_1 lb. per sq. in.	P_2 lb. per sq. in.	$\frac{P_2}{P_1}$	$f\left(\frac{P_2}{P_1}\right)$
0.2500	1.402	114.7	104.7	0.9128	0.1548
0.1111	1.402	114.7	104.7	0.9128	0.1513
0.0625	1.402	114.7	104.7	0.9128	0.1507
0.2500	1.270	114.7	104.7	0.9128	0.1326
0.2500	1.402	89.7	79.7	0.8885	0.1721
0.2500	1.402	64.7	54.7	0.8454	0.1970

THE AUTHOR. Mr. Richardson is probably entirely right in his statement regarding the drop which should take place across a well-designed and constructed Venturi meter. As I was not interested particularly in this drop at the time of making the experiments referred to, I took no pains to measure it accurately and the 2½-in. drop mentioned was not intended to be more than an approximation.

Mr. Richardson's statement as to the permissible ratio between the pressure at the throat and the initial pressure being as low as 0.84 is of course correct, but I would not advise designing a Venturi meter having such a ratio of pressure at its rated capacity, inasmuch as any flow in excess of such rated capacity would increase this ratio to such an extent that it would probably go beyond the range of the pressure-measuring apparatus used in conjunction with the meter, whereas if a meter is designed for its rated capacity with the ratio nearer 0.90, it will be possible at times to exceed this flow without increasing the pressure differences very greatly out of proportion to the increase of the flow.

The tabular values of the function of P_2 and P_1 were plotted out to a very large scale for use during the experimental work and are shown in the paper only to a reduced scale.

No. 1746

THE DISSIPATION OF HEAT BY VARIOUS SURFACES

By T. S. Taylor,[1] Pittsburgh, Pa.
Non-Member

This paper is a discussion of the value of sheet asbestos as a covering for hot-air pipes. The author as a result of many experiments found that tin covered with 0.33*-mm.* (0.013 *in.*) *sheet asbestos will dissipate about* 37 *per cent more heat than a bare tin pipe; asbestos-covered tin having a layer of dust,* 32 *per cent more, and tin with a layer of dust only,* 7 *per cent more. Calculations based on a series of tests indicate that it would require a covering of about* 0.2 *in. of sheet asbestos to make the loss from the covered pipe equal to that from the bright uncovered pipe. A thickness of* 0.4 *in. of asbestos, however, would result in a saving of* 25 *per cent over that of bare pipe, and about seven layers of* 0.013*-in. sheet asbestos loosely applied on a bare pipe would effect a saving of about* 25 *per cent. Application of three layers of air cell* 0.25 *in. asbestos paper will net a saving of* 25 *per cent over that of a bare pipe.*

The author also presents figures, derived from tests, showing the effect of air velocity upon the dissipation of heat. Curves are given showing the watts dissipated per unit area and air velocity for definite surface temperature excesses of from 10 *to* 70 *deg. The effect of the angle of incidence of the air is also discussed in the light of experimental work and it is shown that the maximum dissipation of heat takes place when air is blown over the object at an angle of from* 40 *to* 45 *degrees.*

NOT long ago, while conducting a series of tests to determine the relative thermal insulating properties of asbestos, it was observed that warm water placed in a plain tin vessel cooled more slowly than when placed in a similar vessel covered with thin sheet asbestos. Since this observation was in direct contradiction to popular opinion, it seemed worth while to make some definite tests on the relative dissipation of heat by such surfaces in still air.[2]

[1] Mellon Institute, University of Pittsburgh, formerly of the Westinghouse Electric and Manufacturing Co., East Pittsburgh, in whose Research Laboratory the work herein described was done.

[2] Since compiling these results the writer has learned that somewhat similar results have been observed independently by V. S. Day at the University of Illinois and were noted at the meeting of the National Warm-Air Heating and

Presented at the Spring Meeting, St. Louis, Mo., May, 1920, of The American Society of Mechanical Engineers.

2 To conduct such tests tin vessels were accordingly constructed with a lid at one end and having a diameter of 10 cm. and length of 50 cm. A cylindrical heater was made by winding No. 21 constantan wire longitudinally on an asbestos-board framework so as to slip readily into the vessels. The heater was so constructed that the same amount of heat would be developed per unit area of surface of the vessel, both sides and ends. This made it possible

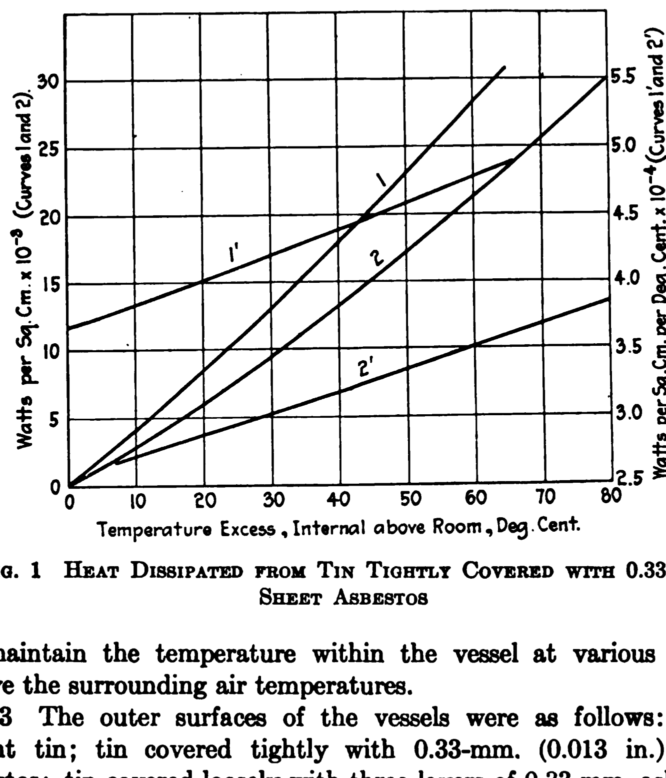

FIG. 1 HEAT DISSIPATED FROM TIN TIGHTLY COVERED WITH 0.33-MM. SHEET ASBESTOS

to maintain the temperature within the vessel at various values above the surrounding air temperatures.

3 The outer surfaces of the vessels were as follows: plain bright tin; tin covered tightly with 0.33-mm. (0.013 in.) sheet asbestos; tin covered loosely with three layers of 0.33-mm. asbestos; tin covered with three layers of air-cell asbestos; tin aluminum-painted; galvanized; and various dust-covered surfaces. A thermometer inserted through the side of each vessel provided means

Ventilating Association, Columbus, Ohio, June 11, 1919. Observations showing the results outlined in the earlier portions of the present paper were first taken about February 1, 1919. Nothing further was done at that time owing to the pressure of other work. Recently, however, occasion permitted further work and the results confirmed the original ones. A preliminary notice of the present results appeared in MECHANICAL ENGINEERING for January 1920, p. 69.

for measuring the temperature at the center of the vessel, and thermocouples of 0.005-in. copper-constantan wire were attached to the other surface of each vessel so as to measure their surface temperatures. The vessels were always placed horizontally in such positions in the room as to be free from unnecessary convection currents.

4 Observations were taken of the amounts of electrical energy, measured by ammeter and voltmeter, necessary to maintain various differences between the temperature within the vessel, as determined by the thermometer and the surrounding temperature. Surface and room temperatures were also taken at the same time. The room temperature was taken at points sufficiently distant from the vessel to be uninfluenced by it. In this manner results were obtained showing the watts dissipated per unit area for various temperature excesses, internal above room, for different surfaces.

TABLE 1 HEAT LOSSES FROM VARIOUS SURFACES

In watts per sq. cm.

Temperature excess, internal above room, deg. cent.	No. 3 — Bare tin	No. 1 — Tin covered tightly with 0.3-mm. asbestos	No. 1 — Covered with dust	No. 3 — Covered with dust	No. 5 — Galvanized sheet iron	No. 6 — Tin covered with 3 layers asbestos paper	No. 7 — Tin, aluminum-painted	No. 8 — Tin covered with 3 layers 0.25 in. air-cell asbestos
	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$	$\times 10^{-3}$
5	1.33	1.87	1.75	1.40	1.50	1.50	1.60	1.08
10	2.70	3.80	3.62	2.90	3.00	3.00	3.20	2.10
15	4.09	5.90	5.60	4.55	4.70	4.65	4.85	3.29
20	5.70	8.00	7.70	6.20	6.36	6.20	6.66	4.45
25	7 18	10.30	9.85	7.95	8.20	7.85	8.60	5.68
30	9.09	12.70	12.06	9.85	10.11	9.60	10.45	6.95
35	10.99	15.15	14.35	11.75	12.15	11.35	12.35	8.43
40	12.88	17.51	16.90	13.85	14.20	13.15	14.35	9.70
45	14.90	20.15	19.50	15.90	16.35	14.90	16.50	11.25
50	16.90	22.20	22.10	18.10	18.50	16.85	18.70	12.60
55	19.03	25.45	24.70	20.35	20.85	18.80	20.90	14.48
60	21.24	28.35	27.60	22.50	23.00	20.75	23.15	15.65
65	23.40	31.40	30.25	24.87	25.30	22.70	25.45	17.30
70	25.55	.....	.....	27.15	27.65	24.70	27.75	18.75
75	27.90	.....	.....	29.35	.....	26.70	30.20	20.20

5 Curves 1 and 2 in Fig. 1 show the results obtained for tin covered tightly with 0.33-mm. sheet asbestos and bare tin, respectively. Curves 1 and 2 give the relations, in watts per square centimeter per degree centigrade plotted against temperature for the

corresponding surfaces. It is seen that the dissipation of heat per unit area per degree of temperature excess increases almost uniformly with the temperature difference over the range of temperatures used. Thus the curves 1 and 2 can be represented approximately by the relation, $W = AT + BT^2$; where W is the watts dissipated per unit area, T the temperature excess (internal above surrounding air) and A and B constants for each surface.

6 Table 1 gives the values of the heat dissipated by the various

TABLE 2 HEAT DISSIPATED BY VARIOUS SURFACES AS COMPARED WITH THAT DISSIPATED BY BARE TIN NO. 3

Temperature excess, internal above room, deg. cent.	No. 1 — Tin covered with 0.33-mm. sheet asbestos	No. 1 — Covered with dust	No. 3 — Covered with dust	No. 5 — Galvanized sheet iron	No. 6 — Tin covered loosely with three layers asbestos	No. 5 — Covered with dust	No. 6 — Covered with dust	No. 7 — Tin, aluminum-painted	No. 8 — Tin covered with 3 layers 0.25 in. air-cell asbestos paper
5	1.41	1.32	1.03	1.13	1.13	1.18	1.04	1.20	0.810
10	1.41	1.34	1.07	1.11	1.11	1.19	1.02	1.18	0.800
15	1.43	1.37	1.11	1.14	1.13	1.18	1.05	1.18	0.805
20	1.40	1.34	1.09	1.11	1.09	1.17	1.03	1.17	0.807
25	1.40	1.33	1.08	1.11	1.06	1.18	1.02	1.17	0.790
30	1.40	1.33	1.08	1.11	1.05	1.18	1.02	1.15	0.772
35	1.38	1.30	1.07	1.10	1.03	1.17	1.02	1.12	0.767
40	1.37	1.31	1.04	1.10	1.02	1.19	1.01	1.12	0.761
45	1.35	1.32	1.07	1.10	1.01	1.20	1.01	1.11	0.755
50	1.34	1.31	1.08	1.10	1.00	1.20	1.00	1.11	0.755
55	1.33	1.30	1.07	1.10	0.99	1.20	1.00	1.10	0.750
60	1.34	1.30	1.06	1.08	0.98	1.20	0.99	1.09	0.747
65	1.34	1.28	1.06	1.08	0.97	1.19	0.99	1.09	0.740
70	....	....	1.06	1.08	0.97	....	0.98	1.09	0.731
75	....	....	1.05	....	0.96	....	0.97	1.08	0.725
Mean	1.37	1.32	1.07	1.10	1.03	1.19	1.01	1.13	0.768

surfaces at corresponding temperature excesses. In addition to the surfaces listed in Table 1, results were also obtained for galvanized surface and tin loosely covered with three layers of sheet asbestos when dust-covered. Table 2 gives the relative amounts of heat dissipated as compared with bare tin for corresponding differences in temperature. Tin covered tightly with one layer of 0.33-mm. sheet asbestos will dissipate 37 per cent more heat in still air than the bare tin. Even when both are covered with dust, such as that usually found on furnace pipes, the asbestos-covered surface will lose 23 per cent more heat than the bare one. The effect of dust on the surface is to increase the loss of the bare tin and decrease the loss of the asbestos-covered one. It requires at least three layers of 0.33-mm. sheet asbestos applied loosely on bare tin in order to dissipate no more heat than the uncovered bare tin. Three layers of air-cell asbestos on tin will permit a loss of but 75 per cent of the bare tin loss. These facts are readily shown by Table 2.

7 It will be observed from a study of Table 2 that the ratio for all surfaces with respect to the bare tin becomes somewhat smaller with increasing temperature excess. This indicates that if the temperature difference were high enough, very little difference would exist between the amounts of heat dissipated by each. This condition would not exist, however, until the surface temperatures were such that the heat would be lost chiefly through radiation, which for the present temperature range plays a small part.

TABLE 3 HEAT DISSIPATED FOR A TEMPERATURE EXCESS OF 20 DEG. CENT.

All values are expressed in watts per sq. cm.

No. 1	No. 3	No. 1d	No. 3d	No. 5	No. 6	No. 5d	No. 6d	No. 7
0.0106	0.0056	0.01085	0.00630	0.00680	0.00970	0.00840	0.01060	0.00625

8 In Table 3 will be found the values of the heat dissipated per unit area per degree surface temperature excess above surrounding temperature for the particular surface temperature excess of 20 deg. cent. These results are taken from curves similar to those in Fig. 2 where the ordinates are the values of watts per square centimeter per degree centigrade surface temperature excess and the abscissæ are the corresponding surface temperature excesses. No great exactness is attributed to the results of Table 3 for the reason that no particular pains were given to the accurate measurement of the surface temperatures.

HEAT LOSSES FROM HOT-AIR PIPES

9 One very interesting feature about these results is their application to the loss of heat by hot-air furnace pipes. From the results in Tables 1 and 2 it is quite evident that hot-air furnace pipes lose more heat when coated with the usual sheet asbestos than when

left bare. Furthermore, this difference is too great to be merely given a casual consideration, and the following brief discussion will emphasize the point, and at the same time indicate the methods of covering pipes which will result in a saving of 25 per cent over that of the bare pipe.

10 Let us consider in the first place what thickness of covering would be necessary in order to insure no more loss of heat by the covered pipe than by the uncovered one. For the same temperature excess, say, 40 deg. cent. internal above surrounding air, the covered pipe loses 17.52×10^{-3} watts per sq. cm. while the bare pipe loses

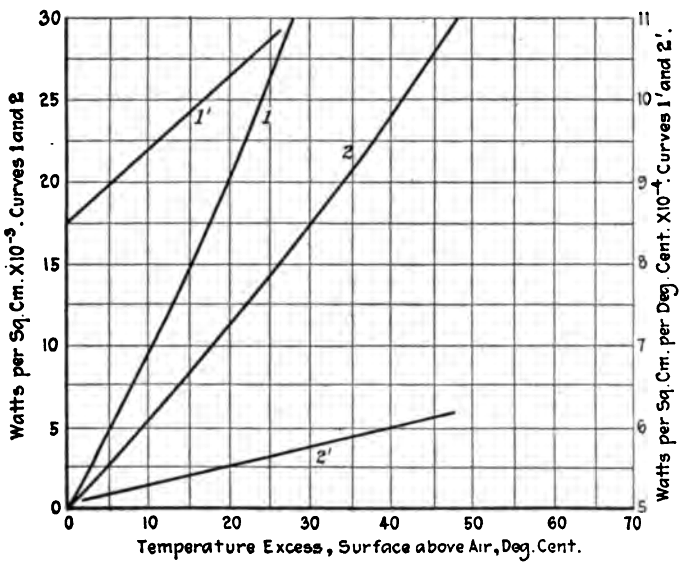

FIG. 2 HEAT DISSIPATED FROM TIN PIPE COVERED WITH ASBESTOS

12.70×10^{-3}. This is seen by reference to curves 1 and 2 in Fig. 1 which show that for a covered pipe (No. 1) to lose only 12.70×10^{-3} watts per sq. cm. requires an outer surface temperature excess of but 12.8 deg. cent. (see curve 1, Fig. 2), and that when losing 17.52×10^{-3} watts per sq. cm. its surface temperature excess is 17.2 deg. cent. (See curve 1, Fig. 2.) Hence sufficient insulation must be added to reduce the surface temperature to the lower value, or there must be enough asbestos added to produce a drop of 4.4 deg. (17.2 — 12.8). The thickness of asbestos necessary is given by the following equation:

$$12.7 \times 10 \times 0.239 = \frac{4.4 \times 0.00035}{d} \quad . \; . \; . \; . \; . \quad [1]$$

where 0.239 is the factor to reduce watts to calories and 0.00035 is the thermal conductivity of asbestos paper in calories per cm. per sec. Solving the equation for d gives a value of 0.51 cm., which is practically 0.2 in. While this is an approximate solution it shows that considerably more thickness of the usual insulation should be applied to hot-air pipes in order to make them as efficient as if they were left bare.

11 It is interesting to speculate as to the possible saving that would result by leaving the pipes bright and uncovered. Suppose there is a temperature excess, internal above surrounding air, of, say, 40 deg. cent. (72 deg. fahr.). As is shown above this corresponds to a loss of 17.52×10^{-3} watts per sq. cm. or 0.113 watt per sq. in. from the covered pipe. If we have 10 pipes 10 ft. long and 10 in. in diameter, that is, approximately 36,000 sq. in. of surface, the total loss would be $0.113 \times 36{,}000 = 4068$ watts. The total loss per day would be $4068 \times 24 \times 3600$, or 3.52×10^8 joules. One pound of coal has a heating value of approximately 12,500 B.t.u $= 1.32 \times 10^7$ joules. Consequently the loss in pounds of coal per day would be $3.52 \times 10^8 \div 1.32 \times 10^7$, or 26.6. This would be equivalent to about 75 bu. during the heating season. The loss through a bare pipe would be equivalent to $100 \div 137$ (see Table 2) of this value, or about 54 bu. These considerations indicate, therefore, that the pipe system in question covered with 0.33-mm. sheet asbestos will lose during a winter season a quantity of heat equivalent to that obtained from 20 bu. of coal more than the same system would lose if left uncovered.

12 Similar calculations to those just made lead to the following results. A layer of asbestos about 0.4 in. thick, seven layers of sheet asbestos (0.013 in.) applied loosely, or three layers of 0.25 in. air-cell asbestos, will reduce the loss through the bright tin surface thus covered to only 75 per cent of what it would be if the pipe were left bare. This means that in the ordinary dwelling having a hot-air heating system, the above thicknesses of insulation applied to the pipes would result in a saving of about one ton of coal per winter season over what would be lost through a bare-pipe system and two tons over what is lost through the pipes when they are covered with the usual thin layer of asbestos.

13 The explanation of the larger loss through a pipe when covered with a thin layer of asbestos is due to the fact that the asbestos surface is some three or four times as great as the plain tin surface so far as molecular dimensions are concerned. The loss

being due chiefly to air contact, it is readily seen that the greater the surface with which the molecules come in contact, the more heat will thus be liberated. The radiating power of the asbestos, also being larger than that of tin, will contribute an additional amount to the advantage of the asbestos as far as heat loss is concerned. The loss due to radiation, however, at these temperature differences is small compared with that lost by convection currents. Since the asbestos surface facilitates the loss of heat due to its increasing the effective molecular contacts and radiation, the surface of the asbestos and also the outer surface of the tin under the asbestos will thus have their temperatures considerably decreased and the result will be that more heat must pass through the tin and asbestos as a consequence of this condition. Therefore, when the surface of the pipe is thus changed and the heat losses increased for a given temperature gradient, it is necessary to overcome this by increasing the thickness of the asbestos layer to such an extent that the thermal resistance of the pipe and asbestos or insulation will cut down the heat flow to the desired amount. That is, the heat loss resulting from increasing the effective area must be counteracted by increasing the thermal resistance through the addition of a greater thickness of insulation.

INFLUENCE OF AIR VELOCITY ON DISSIPATION OF HEAT

14 The work described under this heading was primarily undertaken for the purpose of securing data useful to engineers in designing electrical apparatus. The results obtained are for the surface of a typical end coil of a turbo-generator, but they nevertheless are of value to anyone who is interested in the problem of air cooling.

15 The apparatus upon which the wrapper, composed of treated cloth and tape, was placed was the same as had been previously used in measuring the thermal conductivities of coil wrappers (see *Electrical World*, February 14, 1920, p. 369). An iron bar $1\frac{1}{2}$ in. by $1\frac{1}{2}$ in. by 24 in. was used as a core in order to secure rigidity. A layer of $\frac{1}{16}$-in. heater mica was pressed over the core and a heater wire of No. 21 constantan wound over the mica so as to have eight turns to the inch. The space between the turns was filled with asbestos cement and a second layer of $\frac{1}{16}$-in. heater mica plate was then pressed over the entire apparatus. After having been thoroughly dried out by sending a current through the heater wire while the

entire heater was held between clamps, the apparatus was wrapped with the insulation according to definite specifications. Two thermocouples were placed on the wrapper on each side of the heater, one at the center of each side and another at the edge at corresponding positions along the heater. The thermocouple wires (0.005 in. copper and constantan) were run out along the heater its entire length, the copper ones to one end and the constantan to the other. A sketch of the heater is shown n Fig. 3.

16 The heater after having been thus wrapped and arranged was placed about eight inches in front of the outlet of a blower and parallel to the opening so that the air stream fell at right angles upon it. The ends of the heater were covered with wool felt to prevent loss of heat therefrom. The outlet from the blower was of such dimensions ($2\frac{1}{4}$ in. by $2\frac{1}{4}$ in. by 24 in.) that the coil was com-

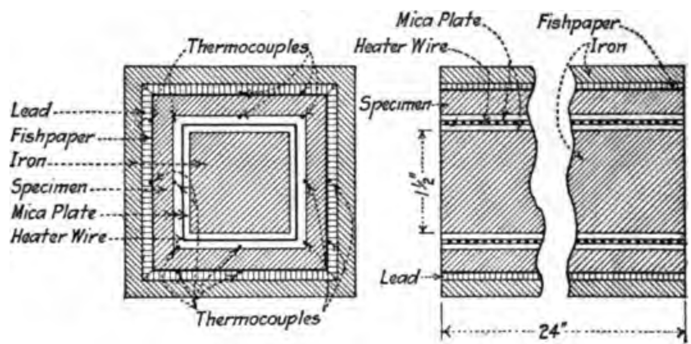

FIG. 3 APPARATUS USED IN MEASURING HEAT LOSSES IN MOVING AIR

pletely within the air stream. Baffle plates placed in the air channel made it possible to secure a symmetrical distribution of the air stream. A small pitot tube made from a hypodermic needle was used to measure the velocity. The differences of pressure were read on a differential draft gage.

17 Observations were also made of the air temperature and barometric pressure. The velocity of the air, V, was calculated for standard conditions, 760 mm. pressure and 25 deg. cent. temperature, by use of the formula.

$$V = \sqrt{\frac{2\,ghd}{d'}}$$

where g is the acceleration due to gravity, h the height of the liquid in the differential gage, d the density of this liquid, and d' the density of the air.

18 Measurements of the air velocity at the point in the air stream where the coil was situated showed but little variation from that for corresponding points at the opening. At least, whatever variation did exist was of the same order of magnitude as the experimental error. It was therefore assumed that the average of the velocity would be a fair value to take as the velocity of the air blowing over the coil.

19 The current in the heater was maintained constant and a constant number of watts were thus dissipated per unit area. Ob-

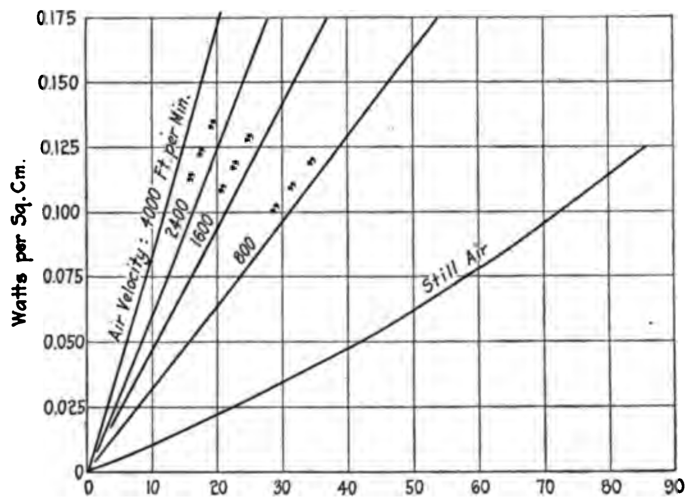

FIG. 4 EFFECT OF AIR VELOCITY ON DISSIPATION OF HEAT

servations were taken of the excess of the surface temperature of the coil (determined from the average of the eight thermocouples on its surface) above the temperature of the impinging air for various air velocities and was found that the heat liberated per degree excess increases approximately uniformly with the velocity over the range of velocities investigated.

20 In the above manner relations were determined for various amounts of heat liberated up to 0.186 watt per sq. cm. (1.2 watts per sq. in.). From these values relations were obtained between the watts dissipated per square centimeter and the corresponding temperature excess of the surface above air temperature for various air velocities.

The curves in Fig. 4 show the watts dissipated per square centimeter and corresponding temperature excesses for air velocities of 0, 800, 1600, 2400, and 4000 ft. per min. For still air it is seen that the amount of heat liberated per degree of temperature excess increases with increasing temperature excess. On the contrary, it is seen that the watts liberated per unit area per degree excess is practically constant for all air velocities other than natural convection currents. The watts dissipated per unit area varies uniformly with the temperature excess for constant air velocities. However, it is not safe to assume

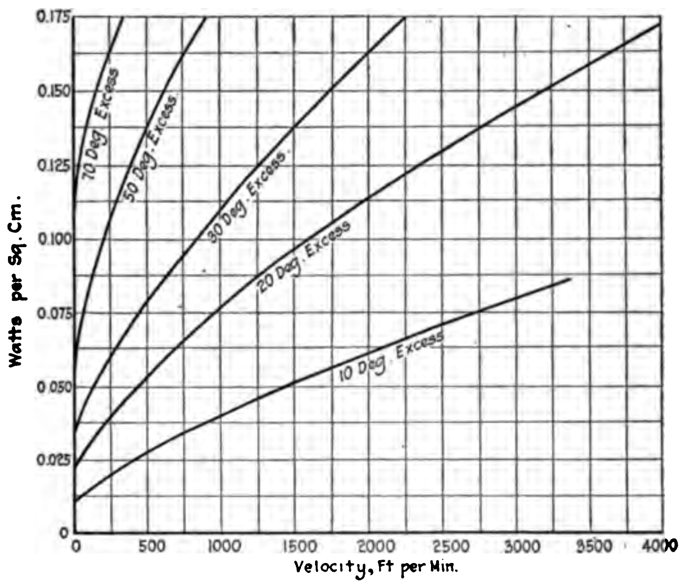

FIG. 5 EFFECT OF TEMPERATURE EXCESSES ON DISSIPATION OF HEAT

from these experiments that such linear relationships continue to hold indefinitely as the watts dissipated increases.

21 Relations were also obtained between the watts dissipated per unit area and the air velocity for various constant temperature excesses, of 10, 20, 30, 50, and 70 deg. respectively. Such relations are useful in determining the amounts of heat that can be liberated for definite temperature excesses and are shown in Fig. 5.

EFFECT OF ANGLE OF INCIDENCE OF AIR STREAM

22 All the foregoing results were obtained for perpendicular incidence or when the angle of incidence of the air stream was zero.

From results that had been obtained previously on the cooling of a very small coil of wire ($\frac{1}{8}$ in. in diameter) when placed in an air current, it was seen that the amount of heat dissipated for a given temperature excess was different for different angles of incidence of the air stream. It therefore seemed worth while to make some tests to determine the way in which the temperature excess of the surface of the coil wrapper varied with the angle of incidence for a definite amount of heat dissipated per unit area and constant air velocity.

23 This was done for a dissipation of 0.145 watt per sq. cm. (0.938 watt per sq. in.) and an air velocity of 3267 ft. per min. Curve

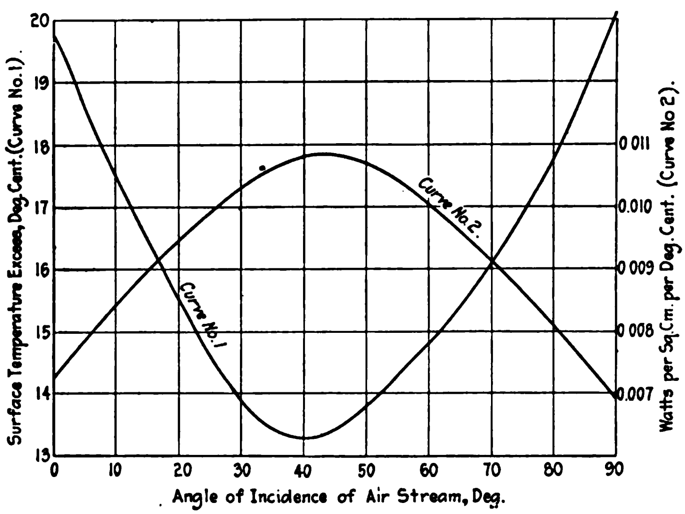

FIG. 6 EFFECT OF ANGLE OF INCIDENCE ON DISSIPATION OF HEAT

No. 1, Fig. 6, shows how the temperature excess changes under the above conditions as the angle of incidence of the air stream changes from zero, that is, perpendicularly, to 90 deg. or parallel to the coil. Curve No. 2 shows how the watts per square centimeter per degree centigrade change with the angle of incidence.

24 The curves are quite interesting in that they show the relative cooling effects of air at various angles of incidence. It is seen that the temperature excess for the particular air velocity of 3267 ft. per min. and a dissipation of 0.145 watt per sq. cm. at an angle of incidence of air stream of about 40 to 45 deg. is only 67 per cent of what it is for an angle of incidence of zero degree. Then, since

the watts dissipated varies directly with the temperature excess for a constant air velocity, it is seen that about 45 per cent more heat will be dissipated per degree under the above conditions of air velocity and heat supplied for an angle of incidence of about 42 deg. than for either perpendicular or parallel incidence. It is further seen that the least heat will be dissipated for 90 deg. incidence or when the coil is parallel to the air stream.

25 Values of the temperature excess were also obtained for 45 deg. and perpendicular incidence for various air velocities and a constant value of the heat dissipated of 0.130 watt per sq. cm. (0.840 watt per sq. in.). The curve in Fig. 7 represents the relative

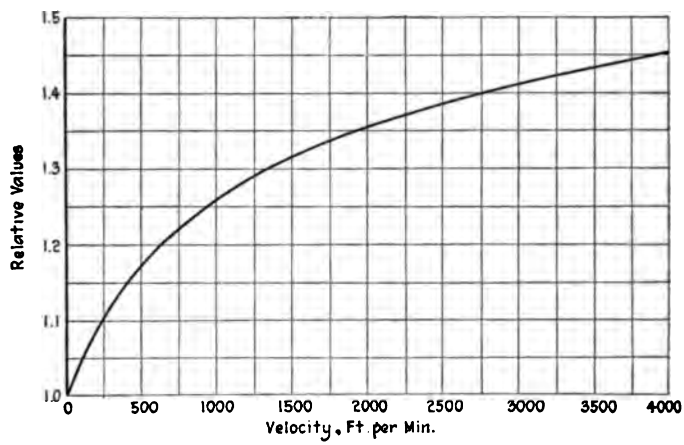

Fig. 7 Curve Showing Relative Values of Temperature Excess

values of the temperature excess for perpendicular and 45 deg. incidence. It is seen that the temperature excess for a velocity of 4000 ft. per min. is 45 per cent greater for perpendicular incidence than for 45 deg. This is entirely in agreement with the results shown in Fig. 6. While these results have been obtained for a particular surface as well as for a special shaped coil, it seems safe to assume that similar results should hold for all surfaces under the same conditions, the only difference being in the absolute order of magnitude and not relative orders of magnitude.

26 Tests were also made for perpendicular incidence of the air stream for various positions of the coil about its axis. No noticeable difference was observed other than what might well be ascribed to experimental error. This is quite significant since it shows the

factor of importance to be the relative position of the axis of the coil in the air stream and not the relative position of the coil about its axis. Such a condition is quite likely to hold for all objects of relatively small cross section completely within the air stream.

27 In conclusion the writer takes pleasure in acknowledging his indebtedness to the Westinghouse Electric and Manufacturing Company in whose research laboratory the work was done, for furnishing the facilities for carrying out the experiments, and in particular to Mr. C. E. Nolan for his able assistance throughout the work.

DISCUSSION

A. C. WILLARD and V. S. DAY.[1] This discussion deals only with the first part of the author's paper, and particularly with the so-called internal temperatures. In general, the author's results confirm the work in this field already completed at the University of Illinois and reported in Bulletin No. 117 of the Engineering Experiment Station as the Emissivity of Heat from Various Surfaces, by V. S. Day. In fact, when the results of the two investigators are compared on the basis of the excess of surface temperatures of the cylinders above the surrounding room temperatures, a satisfactory agreement results. The importance of knowing the surface temperature of the cylinders tested became so evident in the work done at the University of Illinois that it was decided to make the "internal temperature" of each cylinder the same as that of the metal surface itself by using dry saturated steam.

It appears that the internal temperatures used in determining the heat dissipated in the author's tests are based on mercury thermometer readings, taken by a thermometer inserted through the sides of the cylinders tested, with the bulb apparently at the center of the cylinder. If this is the case, the thermometer bulb is also surrounded by the heating coil, and indicates an "internal air temperature" which is of more or less indefinite significance. Unfortunately, the author's paper shows no sectional or other views of his cylinders.

The above statement is made advisedly. Attempts to determine the "air temperature" with small thermocouples No. 23 B. & S. gage in a 10-in.-diameter bright tin leader pipe at the Uni-

[1] University of Illinois, Urbana, Ill.

versity of Illinois showed a drop of about 40 deg. fahr. from the air near the center of the pipe to the air near (approximately 1/100 in.) the inside surface of the pipe. Moreover, in these same tests the temperature of the metal of the pipe was found to be about 60 deg. fahr. lower than the air temperature near the center of the pipe. This was true for air flowing at 6 to 7 ft. per sec. with a maximum temperature near the center of the pipe of about 180 deg. fahr.

The variation in the "internal air temperature" across a pipe or cylinder changes with a change in the rate of air flow, and even in a closed cylinder there will be a marked difference between the temperature of the air near the center and the air in close contact with the cylinder walls.

It, therefore, appears that the only definite basis for estimating the amount of heat dissipated by the metal surfaces which separate air or gas at high temperatures from air or gas at low temperatures is the difference in temperature between the surface itself and the temperature of the cooler surrounding air. In the above cases, the surface was so thin that the temperature drop through the metal is negligible as compared with the drop from the metal surface to the surrounding room temperatures. This was proved by actual test.

Hence, if we take the author's values for the heat given off from a bare tin drum with the surface temperature 40 deg. C. above room temperature from Curve 2 in Fig. 2, we find approximately 0.024 watts are dissipated per sq. cm. If reduced to a coefficient for practical use such as B.t.u. per sq. ft. per 1 deg. fahr. per hr. this becomes

$$K_1 = \frac{.024 \times 929 \times 3.415}{40 \times 1.8} = 1.05 \text{ B.t.u.}$$

when the temperature difference is 72 deg. fahr. For a temperature difference considerably greater than this (135 deg. fahr.) Mr. Day reports a value of

$$K_2 = 1.28 \text{ (p. 18 and 19, Bulletin 117)}$$

Allowing 0.2 per cent increase or decrease per degree variation in the value of K for a temperature difference from metal surface to surrounding air of 150 deg. fahr. we get for the author's emissivity value, reduced to 150 deg. fahr. difference metal surface to air,

$$1.05 = K_1 - 0.002 \times (150 - 72) \times K_1$$

$$K_1 = 1.24$$

and for Mr. Day's emissivity value, reduced to 150 deg fahr. difference metal surface to air,

$$1.28 = K_2 - 0.002 \times (150 - 135) \times K_2$$
$$K_2 = 1.32$$

The value 0.002 is taken from tests by J. R. Allen to show the variation in transmission of heat from a radiator for varying differences in temperature between heating and cooling media.

It will, therefore, be apparent that the author's values for the coefficient of emissivity when surface temperatures are used as a basis are only 5 to 6 per cent lower than Mr. Day's values for bright tin cylinders.

L. R. INGERSOLL. While somewhat striking at first sight, the results of the author's investigations will be by no means surprising to one familiar with the main laws of heat transference. The effect would be exaggerated if the pipe were of very small diameter, for in this case the addition of insulation would add appreciably to the radiating surface. Thus a current-carrying wire with insulation, of not too low conductivity, may run cooler than the same wire if bare.

From a practical standpoint, however, it seems that furnace pipes are so generally dirty and dusty that to leave off the customary asbestos paper covering in the expectation of saving heat would be a practice of doubtful wisdom.

L. B. McMILLAN. This matter of differences in losses from bright and dull surfaces is a striking example of the effect of surface resistance. The author's conclusion that a thin layer of asbestos paper applied closely to a bright surface increases the loss of heat from the surface is undoubtedly correct and is in accordance with former tests of the same general nature. Calling attention to this fact will do a great deal of good in dispelling the popular notion of the magical results which can be obtained with the thinnest layer of asbestos paper.

Asbestos is a wonderful material and by making proper use of its high heat-resisting properties, very desirable results may be obtained, but the thickness and kind of material must be proportioned to suit the conditions. However, the fact that a thin layer of asbestos increases rather than decreases the loss from a bright surface should not be construed as indicating that a bright surface needs no insulation.

The increase in heat loss is due almost entirely to the change in the character of the exposed surface. The resistance to heat flow from a bright surface is greater than that from a dull surface in the ratio of about 0.7 to 0.5. The addition of a thin layer of asbestos or other insulating material adds some resistance, but not enough to offset the decrease in surface resistance, due to the change from bright to dull surface. If sufficient insulation is used it is possible to bring the loss to just as low a point in case the insulation is applied to a bright surface as could be done by the same amount of insulation applied to a black or dull surface.

In the case of furnace pipes, the losses from the bright surfaces are so great that it would be wasteful to leave them bare. Therefore, if the bright surface is to be covered anyway, what it would lose if exposed is of litt'e importance. The important thing is what one thickness of insulation will save compared with another.

It should be noted that Eq. [1] and the computations of the following paragraphs are based on the conductivity of asbestos paper wrapped so closely as to exclude all air from between the layers. If plain asbestos paper were used it would not in practice be applied in that manner. A better type of material would be of cellular construction, and actual tests at the University of Illinois on pipes covered with three layers of air cell asbestos showed a saving just double that shown by the same material by the author's calculation.

The author explains the increased heat loss from the papered surface on the basis of molecular area rather than on account of the brightness of the surface. Painting the surface with a smooth coat of paint, smoother even than galvanized iron, will increase the loss even more than asbestos paper covering does. Therefore, it would seem that molecular area had very little effect and that the difference was due to difference between radiation coefficients.

The radiation even at the temperatures considered amounts to almost one-half of the loss from dull surfaces and considerably less from bright surfaces, so that the change in radiating power is sufficient to account for the difference noted.

It has been established by experiments that the radiation from a galvanized iron surface increases very materially as the surface becomes tarnished. Tests reported in *Engineering* (London), October 19, 1917, showed that galvanized iron exposed to the weather for one year offers practically no more resistance to radiation than black iron. Even a few months' exposure decreases the radiation resistance greatly.

This has no direct bearing on furnace pipes, as they are not similarly exposed. However, even they do not retain their bright condition permanently and the results of tests on bright surfaces cannot be considered as fairly representing the average losses from these surfaces.

E. R. Hedrick. The laws that are presented in this paper are for entirely different conditions from those for dissipation of heat from a gas that passes down a long pipe, for the experiments given were conducted in a closed receptacle whose volume remained sensibly constant. In the case of a very hot gas passing through a pipe, as in the case of the flue gasses passing through a boiler tube, the volume occupied by the same quantity of gas changes very decidedly as the gas passes down the pipe, and hence the exposed surface through which cooling takes place diminishes. For this reason no attempt was made to compare these formulæ with those derived by others for flow of gases in pipes.

The formulæ given by the author seem to give a sufficiently close approximation to the experimental data, and may well be accepted for the purposes of this paper. It is believed, however, that these formulæ would be subject to change if the total drop in temperature were greater. The drop in temperature in these experiments is about 70 deg. C., which would correspond to a rather short arc of the curve for many cases.

Attention is called to the fact that the Curve 2 in Fig. 1 is apparently concave upward, whereas the data in Table 1 seem to give a curve that is concave downward. This would be of importance to any one who was attempting to derive an empirical formula.

J. D. Hoffman said that he was glad that the work done by the author and that done at the University of Illinois confirmed one another, as the conclusions were so strikingly opposite to accepted practice.

The Author. In their discussion, Messrs. Willard and Day state that since the internal temperature of the vessels was determined by means of a mercury thermometer inserted through the sides of the cylinders, this internal air temperature has a more or less indefinite significance. Since the heater was of cylindrical form and of such dimensions, as is pointed out in the paper, to permit its being readily slipped into the vessels without making electrical contact with their sides, the entire inner part within the heater would attain a very definite constant temperature for a given input. The only drop in temperature within the vessel would be from the heater to the walls, ends and sides, of the vessel. This distance was of the order of $\frac{3}{32}$ of an inch and consequently the drop from the heater to the vessel walls would attain a definite value for each input. Thus the temperature indicated by the thermometer would represent the temperature of the entire space within the heater, which was by far the greater part of the inclosed space within the vessel. There is little question but that such an arrangement would give an internal temperature as determined much more definite than could possibly be obtained by allowing dry saturated steam to flow into the vessel and be condensed on its walls. This latter method is certain to give considerable temperature drop from the center of the vessel to its walls and would not give a large volume of a definite constant temperature such as is obtained by the author's arrangement. Furthermore, it is by no means safe to assume that the walls of the vessel, when dry saturated steam is passed into it and condensed on its walls, are at the same temperature as that of the enclosure. The steam if saturated will become cooled as it advances towards the walls and a part will be condensed before reaching the walls.

Experiments show that an object which has a surface temperature excess above surrounding air temperature of 18 deg. fahr. in still air will have less than 90 deg. fahr. temperature excess when placed in an air stream of the same temperature and having a velocity of 500 ft. per min. Thus the temperature of a surface is greatly dependent upon the velocity and temperature of the air stream passing over it. In a similar manner there will be a considerable drop in temperature between the center of a pipe and the pipe itself even at low velocities. Even though steam may be used as pointed out above there will be a drop in temperature between the center of the tube and the walls due to condensation or cooling of the steam as it approaches the walls of the vessel. To the author, the internal temperature of the vessels as measured by the thermometer would have a very much more definite value than it would have in the arrangement suggested by Messrs. Willard and Day.

It is quite comforting to note that the so-called coefficient of emission as obtained from the author's results and those obtained by Mr. Day has practically the same value. This is particularly true when one considers that such a coefficient of emissivity depends upon so many different factors — the nature of the surface, temperature of surface and surrounding objects, etc., and is dependent upon the loss of heat by radiation and free convection as well as the relative distribution of the loss between these two methods of transfer of heat.

If Mr. Ingersoll will observe the data in Tables 1 and 2, it will readily be seen that a dust-covered bare pipe still loses about 25 per cent less heat than the same pipe covered with the usual layer of asbestos paper when dust coated. Consequently there is no doubt but that so long as one can keep the pipes free from rust, there is no advantage to be gained by putting on a thin layer of asbestos in the usual manner. On the contrary it is a decided disadvantage.

As is indicated by Mr. Ingersoll, it is a well-known fact that of two cables of the same carrying capacity but having different thicknesses of insulation, the one with the thinner insulation will have the higher conductor temperature under similar conditions. Furthermore, each will be cooler than the bare conductor. This particular fact cannot be interpreted to apply to all cases of insulated bodies. If one knows the constants of the insulation and of the surface one can calculate what thickness of insulation would give a minimum temperature of the conductor or pipe to be insulated. In the case of steam pipes the so-called coefficient of emission is too high to give such a result as is indicated above for cables. In other words the diameter of the pipe would have to be very small indeed to have a thickness of insulation which would give a minimum temperature of the pipe. The proper radius of the pipe including insulation can be shown to be $r = \frac{k}{E}$ for a minimum temperature where k is the conductivity of the material and E the coefficient of dissipation of the surface of the insulation. So that for all steam pipes in use the radius is always greater than this radius indicated above and consequently the application of insulation will always conserve heat.

When considering the influence of a layer of asbestos on a hot-air pipe, one readily sees that the asbestos layer does not increase the effective radius of the pipe, but it does increase very materially the effective surface from which heat can be dissipated both by radiation and convection. This is due to the fact that a very uneven surface as far as molecular dimensions and motions are concerned is substituted for a very smooth surface and one which has a low radiating power.

The author would not have Mr. McMillan think it was his intention to advocate that, in general, a bright surface needs no insulation. However, scientific facts are usually interpreted to indicate the most satisfactory and economical procedure. Since it is the popular belief, not only among the non-scientific people but even among those of scientific training, that the usual method of covering hot air pipes with a thin layer of asbestos paper is a great economy, it is the intention of the author to call attention to the facts in the case as obtained from trustworthy tests and point out that the present methods of insulating hot-air pipes in a useless expense, instead of an economy.

According to Mr. McMillan's statement the losses from a bright hot-air pipe are so great that it would be wasteful to have them bare. The results shown here and those obtained by Messrs. Willard and Day are by no means in accordance with such a statement. From a consideration of the results as shown in Par. 12, it is quite doubtful whether it is economy to apply sufficient thickness of insulation to reduce the losses to say 75 per cent of that from a bare pipe. This would require about 7 layers of thin-sheet asbestos applied loosely or about 3 layers of the usual $\frac{1}{4}$-in. air cell. The expense and trouble to do so is considerable and it is very questionable whether it is at all economical.

Mr. McMillan is very much in error when he says that painting a bright surface with a smooth coat of paint increases the losses even more than a layer of asbestos applied in the customary manner. By reference to Table 2 it is readily seen that experiments show just the opposite to be the case. An asbestos layer on tin increases the losses 37 per cent, while a coat of aluminum paint increases the losses but 13 per cent. Mr. McMillan certainly did not interpret the data given in this paper correctly if he obtained such an idea from the results contained therein. As is pointed out in the main text of the paper, molecular area does have a very important part in determining the losses from the surface. By introducing a rough surface such as asbestos instead of a bright surface, the effective molecular area is very considerably increased. This furnishes more points and surface from which radiation can take place and also with which the molecules can come into intimate contact. Consequently more heat must be carried away.

That the losses due to radiation alone at the temperature of steam pipes are about $\frac{1}{2}$ the total losses is quite doubtful. By using the accepted value of the coefficient of radiation for a black body at such temperatures, it is easily shown from Stefan's Law that not nearly ½ the amount of heat should be expected to be lost by radiation. Those experiments which indicate such to be the case were not performed in a sufficiently good vacuum to be rid of convection losses. In fact the pressure was still sufficient to permit of very considerable losses by convection on account of the large magnitude of the mean free path of the molecules at such pressures. The fact that the so-called radiation resistance of galvanized iron decreases as the surface becomes tarnished confirms the statement that molecular area has considerable to do with the heat lost. The tarnishing of the surface results chiefly in increasing the molecular area by the forming of the oxide.

As indicated above, while these tests show the uselessness of covering hot air furnace pipes with the usual thin-sheet asbestos, the same conclusions must not be drawn with respect to steam pipes. In the latter case we are dealing with very good radiators, having uneven surfaces and thus large molecular area, and where it has been the practice to use sufficient insulation to cut down very noticeably the heat losses.

There seems to the writer to be no reason why the laws governing the loss of heat from the outer surface of a covered or uncovered hot-air pipe should be any different for a pipe in which the air was moving and when the air was stationary. To be sure the average value of the internal temperature will have a different significance. The manner in which the surface obtains its heat from within, from still air, moving air, steam or water will have no effect upon the laws governing the loss of heat from the outer surface. These factors will only determine the relative temperature of the outside with respect to the surroundings. The formula in Par. 5 was found to fit the curves such as 1 and 2 in Fig. 1 very nicely for several cases. To be sure if the temperature increased sufficiently high a third term involving the fourth power of the temperature excess was useful in getting a more accurate agreement between the equation and the observed curve. This extra term is useful in taking account of the extra heat lost at the higher temperatures due to radiation. At the ordinary temperatures found in hot-air pipes and in particular those found in the outer surfaces of such, the heat lost by radiation is quite small indeed.

No. 1747

PULVERIZED COAL IN METALLURGICAL FURNACES AT HIGH ALTITUDES

By Otis L. McIntyre, New York, N. Y.
Member of the Society

The following paper describes experiments with pulverized coal which led to the installation of apparatus for its use in the blast furnaces, reverberatories, and sintering machines of the Cerro de Pasco Copper Corporation at La Fundicion, Peru. Information is given of the method of determining the firing performance of pulverized coal; the behavior of the various mixtures; experiences with different air pressures; and the effects of ash. The experiments were conducted at an elevation of 14,000 ft. and this fact greatly adds to the value of the data presented, for heretofore but little attention has been paid to the effects of altitude upon the burning of pulverized coal.

THE Cerro de Pasco Copper Corporation at La Fundicion, Peru, uses about 65,000 tons of coke per year, of which about 85 per cent is local coke made at the smelter, and 15 per cent is imported. This latter is very expensive, and of course both classes of coke enter largely into the smelting costs; consequently, about two years ago it was decided to determine what could be done in the way of using pulverized coal in the various departments of the smelter. The preliminary work consisted in determining the general combustibility of the local coals in pulverized form. These coals are obtained from two mines operated by the company and have the following general analysis:

Ash	Volatile Material	Fixed Carbon
26.8	40.05	33.15

2 This coal was dried by hand on steam hot pans to less than 1 per cent moisture, and then ground in a 4-ft. by 4-ft. Marcy mill, the product being stored in barrels until a sufficient quantity had

Presented at the Spring Meeting, St. Louis, May 1920, of The American Society of Mechanical Engineers.

been pulverized to run a test. An average screen test of this pulverized coal was about as follows:

+ 60 mesh, 8.00 per cent
+ 100 mesh, 8.00 per cent
+ 200 mesh, 14.00 per cent
− 200 mesh, 70.00 per cent

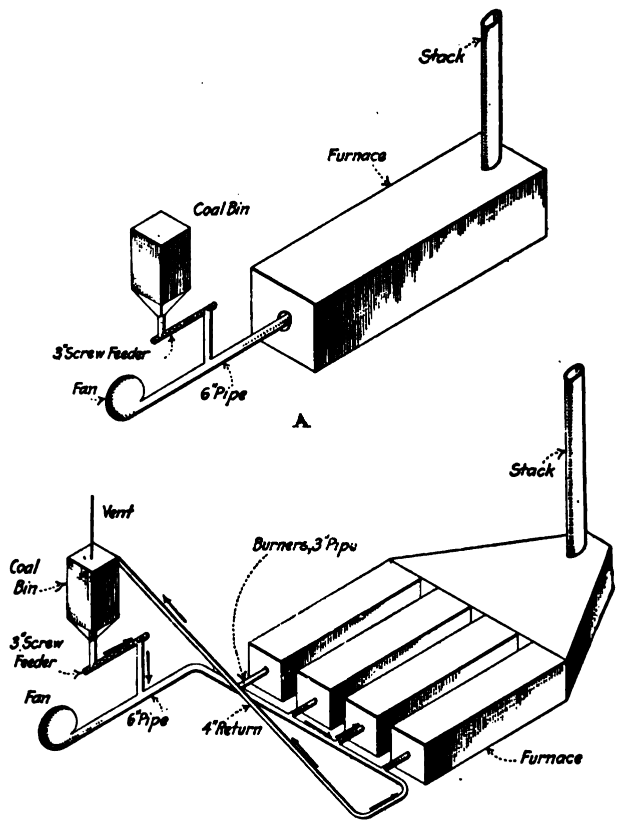

FIG. 1 LAYOUTS OF EQUIPMENT USED IN FURNACE TESTS OF PULVERIZED COAL

3 The equipment used in the test is shown by Fig. 1-A and consists of coal hopper, a 3-in. feed screw driven by variable-speed motor, and a No. 2 Sturtevant blower supplying the air. The burner was a standard 6-in. pipe projecting about 12 in. into the furnace, which was approximately 4 ft. by 4 ft. by 16 ft. and con-

structed of firebrick. A number of tests were run with this equipment and though no pyrometric measurements were taken, observation of the furnace showed the results to be satisfactory.

4 The tests were first made with pure pulverized coal, and then with mixtures of coal and coke breeze, varying from 10 to 35 per cent of coke breeze, which gave practically the same results as the pure coal. The layout was then changed, Fig. 1-B, to test the practicability of using more than one burner with a single feeder. This test was run with the 4-in. return pipe both open and closed, the results indicating that satisfactory operation could be obtained by either method with a properly proportioned pipe system.

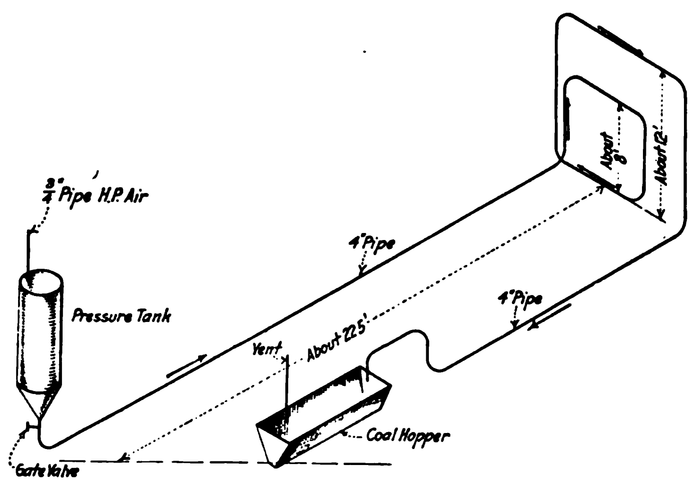

FIG. 2 LAYOUT USED TO TEST FEASIBILITY OF CONVEYING PULVERIZED COAL UNDER DIRECT-AIR PRESSURE

5 The next test made was in the sintering of fine ores on a standard Dwight-Lloyd sintering machine. These machines are oil-fired, and if coal could be substituted it would show a considerable saving. The equipment used in this test was the same as shown in Fig. 1-A, except that a 1-in. screw feeder, a smaller fan, and a 2-in. pipe burner were used. This test produced a satisfactory sinter, though some trouble was encountered in the primary ignition of the coal, and the standard oil muffle proved to be too small.

6 The next experiment was to test the feasibility of conveying pulverized coal under direct-air pressure. The layout used is shown

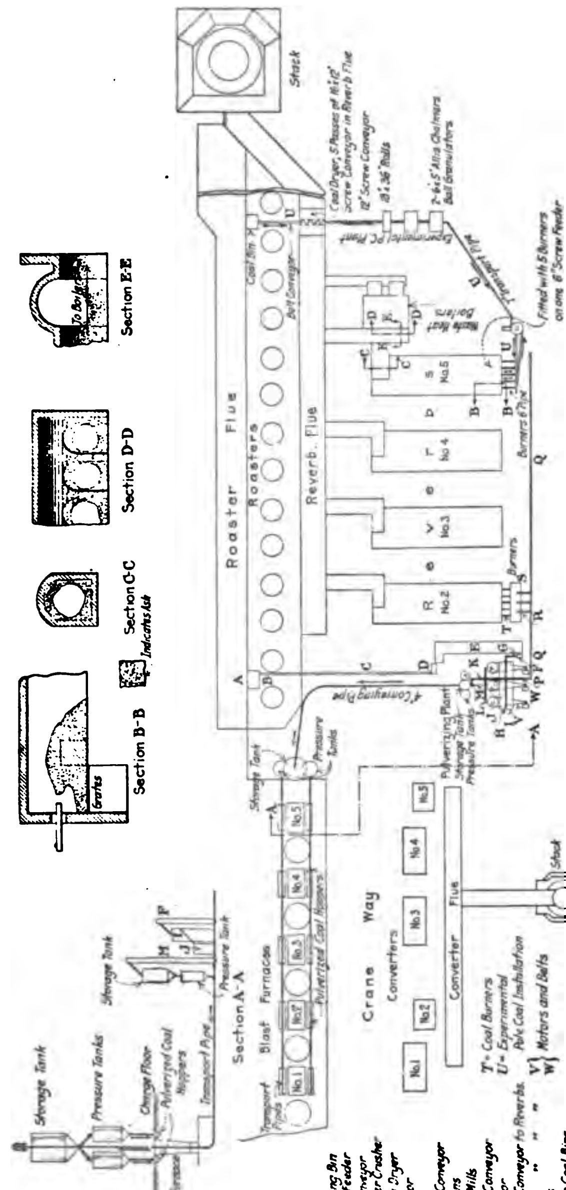

FIG. 3 LAYOUT OF PRESENT CERRO DE PASCO COPPER CORPORATION'S SMELTER, SHOWING LOCATION OF COAL-PULVERIZING PLANT

in Fig. 2. Pulverized coal was placed in the pressure tank and air at 20 to 25 lb. was then admitted through the $\frac{3}{4}$-in. pipe at the top of the tank. The 4-in. valve at the bottom was then opened and the coal passed through the 4-in. piping system to the coal hopper. In this way 4000 lb. of coal was transported in $1\frac{1}{2}$ to 2 min. The loss through the vent pipe varied from 100 to 200 lb. This can be taken care of by using dust collectors on the hopper, or an exhaust system which would return this waste coal to the main hopper.

7 The foregoing tests showing up so well, it was decided to erect a larger experimental pulverizing plant. There were available for this purpose one set of 18-in. by 36-in. rolls, one 4-ft. by 4-ft. Marcy mill and two 6-ft. by 4-ft. Allis-Chalmers ball granulators. The drier consisted of five passes of 16-in. by 12-ft. screw conveyor, mounted in a brick housing on top of the reverberatory flue, through which part of the flue gases were by-passed. The layout in plan is shown in Fig. 3.

8 After completing this plant it was decided to make the first experiment on the blast furnaces, so No. 5 blast furnace was selected for the purpose and was equipped on one side only, as shown by Fig. 4. The coal was ground at the experimental plant and transferred to the No. 5 furnace in a hopper car, being weighed in transit. A number of tests varying from 8 to 12 hr. were run with this equipment. The furnace air pressure averaged 34 oz., and auxiliary air for injecting coal about 22 lb. The coke charge was reduced first 25 per cent, and then 50 per cent. These tests were so encouraging that it was decided to equip the other side of the furnace with coal feeders and run a test of several days' duration. This was done and the results were entirely satisfactory. During these tests the auxiliary air was taken from the converter air line, which varied from 12 to 16 lb. pressure.

9 The following quantities will give an idea of the proportion of coke and pulverized coal used:

Length of run, hours	14	50
Normal charge of coke, lb.	31,000	114,000
Actual charge of coke, lb.	17,000	61,800
Pulverized coal fed to furnace, lb.	8,900	41,000

10 The analysis and screen tests of the coal used were practically the same as noted above. The furnace performance through all tests was carefully observed and was found to be fully equal to that when operating on the normal coke charge. Two difficulties

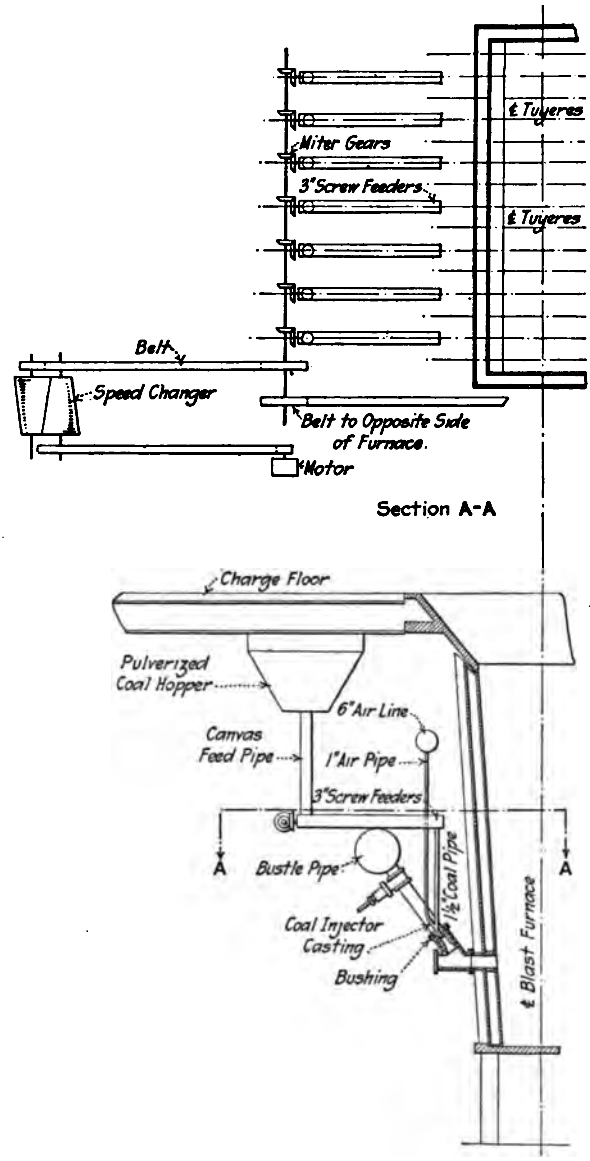

Fig. 4 Experimental Pulverized-Coal Equipment of No. 5 Blast Furnace

were encountered on the blast-furnace test: namely, keeping some to the feeders in operation and keeping the tuyeres open. It was observed that in some of the feeders there was a slight back pressure, due probably to partially blocked tuyeres. This did not affect materially the feeding, but forced some coal dust into the feeder bearings which mixed with the oil and finally bound the bearings so that it became necessary to shut down that particular feeder and clean the bearings. This was easily done without shutting down the other feeders, as the gears on the main shaft were mounted on feathers and provided with shifters. By using dust-proof bearings and a better-designed injector, we expect to eliminate this trouble.

11 Keeping the tuyeres open is absolutely essential to the safe and efficient operation of this process, and as it is a manual operation it must be handled by the operators. During these tests, tuyeres were "punched" every fifteen to twenty minutes on signal. On one occasion a tuyere became badly blocked, the feed was cut off and the tuyere cap opened. The blast from the furnace blew out a dense cloud of coal dust and molten material. The dust was ignited and burned on the outside of the furnace for 20 to 30 sec. with an intense flame about six feet long, the tuyere acting as an ordinary coal burner.

12 In view of the difficulty of keeping the tuyeres open and the connections air-tight, it is probable that the most satisfactory place to inject the coal into the furnace would be through a separate opening in the jackets, between and preferably somewhat above the tuyeres.

13 The No. 5 reverberatory was selected for the final test. All four reverberatory furnaces in use are identical: they are old-style, designed for hand-firing and about 18 ft. by 58 ft. inside the bridge wall. No. 5 was fitted up as shown at *U*, Fig. 3. The coal was discharged from the last mill into a hopper and dropped into a 7-in. pipe where it was picked up by an air jet and conveyed to the coal hopper, a distance of about 80 ft. and rise of about 30 ft.; the top of the hopper was constructed similar to a cyclone dust collector. From the hopper a 6-in. variable-speed screw feeder fed the coal into the suction side of a No. 9 Sturtevant Monogram blower; this in turn discharged the mixture of coal and air into the feed piping from which branched five 6-in. pipe burners into the furnace, the excess air and coal returning to the hopper.

14 This test was disappointing from the actual results, but when the following difficulties which were encountered are corrected,

the furnace will, beyond question, show a much higher efficiency than the hand-fired furnace. First, the coal could not be dried sufficiently, the average moisture being in excess of 1.5 per cent. This introduced handling troubles. The plant would not grind sufficient coal to the required fineness, the average screen analysis being:

+ 65 mesh, 22.8 per cent
– 65 mesh, 8.5 per cent
– 100 mesh, 25.6 per cent
– 200 mesh, 42.4 per cent

15 The discharge from hopper to feeder was too small, and the coal continually caked and bridged. The screw feeder was too short so that the coal flushed badly at times; also the discharge from the feeder was too far from the fan so that the coal accumulated in the suction pipe and had to be removed with an air jet. Under these conditions it was obvious that uniform feeding, which is essential to efficient operation, was impossible.

16 This test covered about nine days, and was run for about two days with the return pipe open. Sometime during the second day the return pipe was blocked, due to overfeeding, and it was decided to continue the test without opening the return pipe, the only difference being an apparently heavier feed at the burner farthest from the fan. With a properly designed piping system there seems to be no reason why a series of burners cannot be operated from a single feeder with or without a return. The last day's run of this test was made with a mixture of 75 per cent coal and 25 per cent coke breeze, which gave results equal to straight coal.

17 The following table shows a comparison between the average performance of reverberatories Nos. 2, 3, 4 (which were hand fired) and 5 over the same period:

No.	Charge Smelted Per Hr. Tons	Coal Used per Hr. Tons	Smelting Ratio	Hr. Run	Time Lost
Aver. of 2-3-4	5.35	2.00	2.67	262	....
5	4.63	1.99	2.33	225	37

18 These results are really not so bad when the troubles encountered are considered and it is remembered that this furnace was

not designed for pulverized coal and that it cools very rapidly during any shutdown, and considerable time is required to bring it up to the smelting temperature again.

19 As ash accumulations are an important factor in reverberatory smelting with pulverized coal, close observation was made of these accumulations, and the following samples taken:

1 Ash and slag float on the bath: Comes out when skimming in small and large pieces, sometimes has to be broken to pass the skimming door, is easily handled when furnace is hot, but is tough and sticky when furnace is cool.
2 Ash in boiler cross-flue: Spongy mass of ash and some slag accumulates in fairly large quantities in cross-flue between furnace and waste-heat boilers; is soft and easy to remove when first deposited, but if allowed to remain, is very difficult to remove. (See sections E-E and D-D, Fig. 3.)
3 Ash on sides and roof of furnace: Almost pure ash, lightweight and brittle when cold, appears to accumulate on sides and roof of furnace until too heavy to stick, when it drops and floats on the bath.
4 Ash in reverberatory flue, very similar to No. 2 (see section C-C, Fig. 3).

20 Section B-B, Fig. 3, is taken at right angles to the bridge wall of the furnace and shows the large mass of ash, slag, unburned coal and coke which accumulated here and materially affected the operation of the furnace. This was deposited on the bridge wall and gradually built up in the shape of large horns in front of each burner, sometimes reaching within 8 or 12 in. of the burner.

21 Quite a large quantity of ash was deposited each shift on the boiler tubes, but was easily blown off by compressed air once or twice a shift. It was estimated that at least 50 per cent of the total ash was disposed of in the manner described, while the remainder was deposited in the main flue and went up the stack. The average ash analysis was as follows:

SiO_2	Al_2O_3	FeO	CaO	MgO	Cu
49.9	28.4	11.0	2.5	0.26	1.24

22 The high copper content is probably due to calcines lodging on the ash and slag float. Two samples showed Cu 2.92 and 1.91, respectively.

23 As a result of these experiments a modern 250-ton coal-pulverizing plant was designed and is now in course of erection. Blast furnaces, reverberatories, and sinter plant will be equipped for pulverized coal, and the experiments will be continued to ascertain the equipment most suitable for local conditions, which will then be used at the new smelter now being constructed.

24 Fig. 3 shows the layout of the present smelter in plan, with various elevations, and the location of the coal-pulverizing plant relative to the general smelter layout. The flow through the plant is indicated by the legend at the left of Fig. 3.

25 In conclusion, it may be of interest to note that these experiments and tests were carried out at an elevation of 14,200 ft.

DISCUSSION

W. N. Best. The writer's only criticisms of the paper are of the number of burners used and the lack of combustion space.

It is the writer's belief that with pulverized coal as few burners as possible should be used. One burner can always give better results than two or four, owing to the fact that the flame and temperature can be controlled at the will of the operator. Of course, the burner should be of such proportions as to insure the capacity required for the furnace and the maximum output from the furnace.

The writer also believes that a combustion chamber is necessary in all furnace work in order to insure the reduction of the pulverized or liquid fuel to heat before it reaches the furnace proper; that is, the combustion chamber should provide sufficient space to consume the fuel before it reaches the furnace proper. In this space, of course, the pulverized fuel combines with the air required for its combustion. The writer believes that the use of the combustion chamber will eliminate the piling up of unconsumed fuel before reaching the bath.

The Author, commenting on the discussions by W. N. Best, wrote as follows: "I wish to note that the size of burner is limited, because as the furnace capacity increases, the burner naturally has to be larger and in the larger sizes it is very difficult to get the required mixture of coal and air for complete combustion. Therefore, with large furnaces and high rates of combustion, it is more desirable to use multiple burners.

"Regarding combustion space, it is well known that especially in pulverized coal, this space has to be designed to meet the individual furnace conditions.

"Referring especially to my paper, it must be noted that the experiments were made on old style, copper reverberatory furnaces on which it was undesirable at the time to change the combustion chamber to the required size."

No. 1748

THE SEPARATION OF DISSOLVED GASES FROM WATER

By J. R. McDermet, Pittsburgh, Pa.
Associate-Member of the Society

This paper covers the results of a research on the separation of dissolved gases from boiler feedwater, both on a laboratory and a commercial-plant scale. Fundamentally the process consists of the rapid injection of heated water into a region of vacuo, and an explosive boiling of it at the expense of the heat of the liquid available to the vacuum, with a simultaneous recovery of the heat liberated to the vacuum by a heat exchanger or condenser cooled by the incoming water, preliminary to its heating. It is possible to deduce from a résumé of existing knowledge the results to be expected in the elimination of corrosion from boilers and economizers fed with oxygen-free water. Chemical analyses suggest the elimination of carbonates dissolved in water as bicarbonates in excess of their true solubility as carbonates. Test data are submitted to show the economies to be gained from the surface condensation of steam free from non-condensible gases both in increased conductivities and higher vacua.

THE idea upon which the method of separating dissolved gases from water described in the present paper is based was originated by Mr. W. S. Elliott of Pittsburgh and sponsored by him for development under the industrial fellowship system of the Mellon Institute of Industrial Research, Pittsburgh, Pa. It appeared at the conclusion of the laboratory investigation that the process evolved from the idea successfully fulfilled three requirements which were regarded as necessary for commercial utility: It was able to handle water in quantities sufficient for the feeding of power-plant boilers with a minimal operating expense and was particularly applicable to this service; it was capable of removing all of the dissolved gases in a commercial sense; and it involved no methods or apparatus with which the average operating engineer was not familiar. The scale of experimentation was therefore enlarged, but the basic idea underwent no change. This work was undertaken as an industrial-research problem with the expectation that it would

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have two distinct fields of usefulness; the elimination of boiler and economizer corrosion in boiler plants, and of the entire distribution system in central station heating; and the improvement of conductivity in surface condensation of steam.

2 The solution of gases in water obeys Henry's law, which states that a solvent in contact with a gaseous mixture dissolves the component gases in proportion to the partial pressures which they exert, provided they do not unite chemically with the solvent. Carbon dioxide which unites chemically with water deviates from this law, but is itself of variable content in atmospheric air. With this reservation in mind, it is possible to say that if extraction is complete, the gas mixture extracted from water is roughly of the same composition as air from which solution was attained. At incomplete extractions proportionality factors between the components appear due to different degrees of solubility. The process with which this paper deals might therefore with perfect propriety be called "air separation."

LABORATORY INSTALLATION

3 The initial laboratory installation consisted of an instantaneous water heater drawing its water under pressure from a supply main, and delivering heated water under thermostatic control to the spray head of a separator. The separator consisted of an airtight steel tank with a series of copper return-cascade pans supported by a central steel column, over which the spray head delivered water uniformly in a thin sheet. The separator tank was connected with a barometric tail-pipe column at the bottom to remove the treated water and with an air-pump connection at the top. Vapors and gases were withdrawn from the air-pump connection through a small condenser cooled by the supply water to the heater, and equipped with a trap for retaining condensate. Non-condensible gases were withdrawn from the condenser by a variable-speed motor-driven air pump and passed to a displacement air bell, adapted from a standard gas-meter prover, for collection and measurement. Thermometer wells were installed to indicate all significant temperatures and weighing tanks to collect and measure treated water. The condensate trap was so valved that it could be cut in and out of the system for the purpose of collecting the condensate over stated periods, and the air bell collected all non-condensible gases. While the provisions for determining all data but volume of non-condensible gases proved adequate and satisfactory, it was immediately

discovered that the air-bell data were not of consistent accuracy in spite of the laboratory tightness of the complete system. It was accordingly necessary to turn to direct testing of the water which had passed through the process. The laboratory arrangement is indicated in the diagram of Fig. 1.

METHODS USED IN LABORATORY TESTS

4 Three methods, developed previously by experimenters in other fields, were utilized in testing for the dissolved gas content,

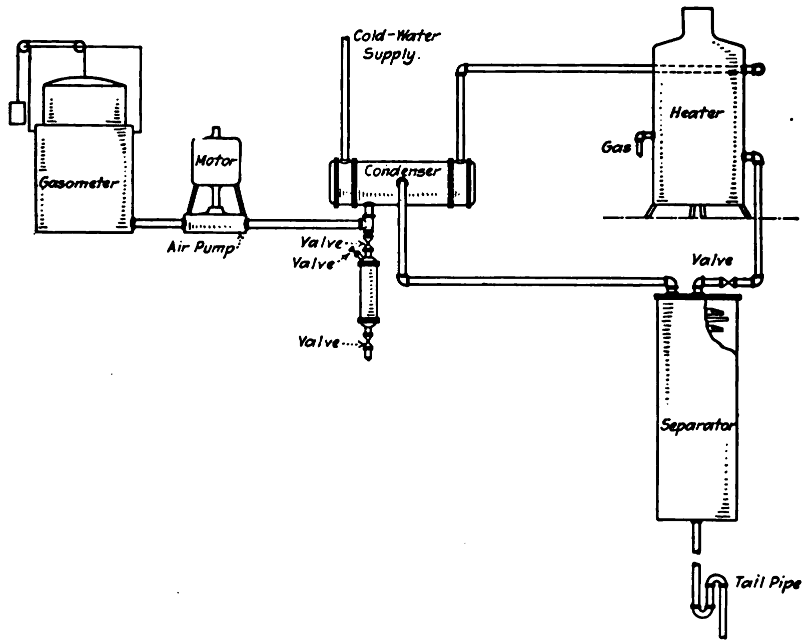

FIG. 1 LABORATORY, SEPARATION EQUIPMENT

after careful checking had established their accuracy. While the accuracy attained will be discussed under the methods, a description of the procedure in testing for precision is essential to an understanding.

5 A sample of distilled water was allowed to saturate itself with air at outdoor temperatures for a period of one week, and a careful record kept of temperature variation. At the end of this period it was placed in a constant-temperature, ventilated, electrically heated oven set to maintain within one-tenth degree a

temperature 5 deg. cent. above the maximum temperature of the saturation period, and held at this temperature for 24 hr. At the end of 24 hr. saturation was assumed to be in equilibrium with the temperature, and samples were tested by any of the methods employed. The measure of agreement between the results of these tests and saturation table values as given by Landolt-Börnstein for a total air pressure of 29.92 in. of mercury was taken as the accuracy of the particular method.

6 The first of the methods in the order of their utility was the iodometric titration method of Winkler for dissolved oxygen; this could be relied upon to give results within 99.8 per cent of the value for dissolved oxygen from the solubility tables, no regard being paid to variation in barometric pressure but all other refinements observed. Since this method was doubtless the one upon which the solubility tables were based, the close agreement is retroactive in indicating the accuracy of the calibration method. The objection to the universal use of this iodometric titration was the fact that it gave results only in terms of oxygen.

7 The second means of testing employed was that of Freeman and Preusse's modification of Reichardt's method of boiling. Without going into excessive detail, it will be sufficient to say that the method employs a closed glass apparatus system which allows of liquid expansion and boiling without contamination. The apparatus was filled with water and boiled for several hours to eliminate all air. A measured sample of the water under investigation was then injected into the system and the boiling continued, the evolved gases being collected and conducted to a water-jacketed gas burette and measured over mercury covered with a film of high-boiling oil. This method gave results that checked within 97 per cent of saturation-table values but suffered from all the errors incidental to volumetric gas measurement, and was tedious and difficult of manipulation. The result, however, was basic.

8 The third method was a displacement with either carbon dioxide or hydrogen. The gas from a compressed cylinder was led through an absorption train to eliminate all impurities; was next bubbled slowly through a measured sample of water for a period of twenty-four hours; and, after passing through the water, was led through another absorption train to eliminate the displacing gas. The residue was conducted into a gas burette and measured as in the methods before. This method gave results which agreed within 95 per cent of the solubility-table values when a suitable correction

was applied to the constituent value of the displacing gas which was dissolved in the sample and removed by the absorption train. This method was valuable only in that it gave a determination which did not involve the phenomena of boiling by heating, but was abandoned very early in the investigation.

9 While no attempt was made to check the accuracy of the methods at extreme dilutions of dissolved gases, it was felt that the agreement between basically different methods which agree with each other at saturation is in itself a sufficient guarantee that no serious errors were made. As experience soon demonstrated, the degree of separation of dissolved gases was so complete that the proportions of constituent gases were roughly the same as for atmospheric air. When this fact was definitely confirmed by a long series of pipette analyses, reliance was placed entirely upon the chemical titration method of Winkler.

FACTS DETERMINED BY LABORATORY TESTS

10 Aside from the numerical data obtained, the laboratory investigation developed two facts that are significant of the process as a whole. As was mentioned in a previous paragraph, the tank and suspension for the spray pans in the separator were of steel, the pans themselves being of spun copper. At the conclusion of about four months of intermittent testing the apparatus was taken apart and examined. Steel tank, copper pans and condenser gave evidence of very accelerated corrosion. Steel surfaces were very badly pitted, and even the copper pans showed signs of general wasting with a black, loosely adherent film of copper oxide coating all parts not washed by the water. Cognizance of this virulent action was taken in the design of the first commercial apparatus, and the separator was made entirely of cast iron and non-corroding bronzes, which materials have withstood the corrosive tendencies for over four years of experimental operation.

11 The laboratory experiments also developed a further fact which was the cause of much confusion at the outset. A glance at the solubility tables for dissolved air will indicate that water at atmospheric boiling point still retains a considerable portion of dissolved air. Water which has been passed through this process and completely degasified is never in equilibrium with the temperature as far as solubility is concerned. Such water has an enormous avidity for redissolving small amounts of air and it is only by the most painstaking manipulation that accurate results are possible in

the determination of the extent of separation. In the practical application of the process, absolutely rigid exclusion of air must be maintained if the full benefits of the process are to be secured.

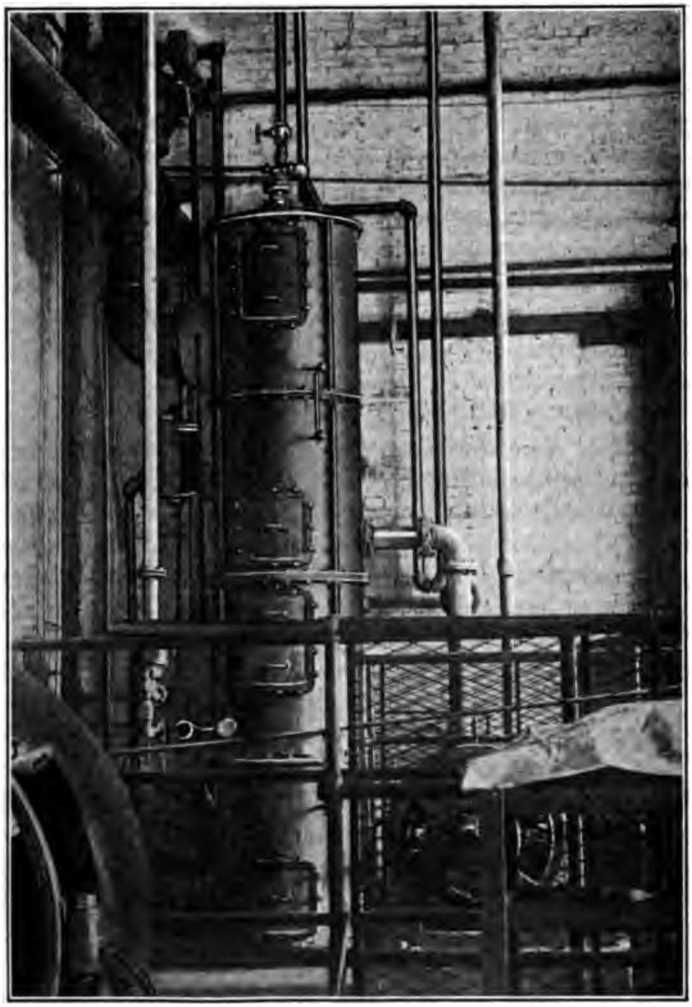

FIG. 2 INTEGRAL VERTICAL-UNIT COMMERCIAL INSTALLATION

COMMERCIAL INSTALLATION

12 Empirical design factors governing separator volumes, pan areas, and the proportioning of the injection nozzle were adopted

from the laboratory installation and embodied in the first commercial installation which was put in service in the Jeannette, Pa., works of the Elliott Company of Pittsburgh in the autumn of 1915. This installation has been in routine operation since that time. The separator proper was designed for 1000 boiler hp. at the outset, and consisted of a 1000-hp. Elliott cylindrical heater shell with modified water injection, different proportions of water-storage volume and pan area, and a change of materials to resist corrosion. This design of separator has been retained intact, but the auxiliary arrangement has been modified three times. At the outset a 300-hp. open feed-water heater was installed to supply hot water, and vacuum was maintained by a rotary motor-driven dry air pump in series with a condenser. Condensate from the condenser was allowed to drain back into the separator, and the air pump discharged non-condensible gases directly into the atmosphere; otherwise the scheme of connections was identical with that in Fig. 1.

13 An increase in boiler capacity later made necessary the installation of a 1000-hp. heater of the same shell type as the separator, and the entire outfit was repiped and made into an integral vertical unit as shown in Fig. 2. Simultaneously the mechanical air pump was replaced by a steam ejector and the condenser was mounted on the atmospheric side to recover the ejector steam, condensate drainage being now into the heater instead of the separator.

14 It was felt at the beginning that condenser conductivities would be very low on such vapor mixtures as are withdrawn from the separator of an air-extraction apparatus, and a condenser of a relatively large capacity was installed. This condenser was removed from the apparatus when opportunity permitted and tested under the same circulation rates and with steam-air mixtures of the same composition as those imposed in practice. These tests gave a condenser conductivity of from 50 to 60 B.t.u. per hr. per sq. ft. per deg. fahr. temperature difference. The condenser operates, however, with a large temperature gradient and the actual transfer per unit of surface is very high. The design is strictly conventional, having four water passes with the circulation through the tubes.

EXPLANATION OF SEPARATOR ACTION

15 The mechanism of the separation appears to depend upon three factors: namely, the natural decrease of solubility with temperature, the mechanical agitation of boiling, and the reduction of

pressure. The reduction of solubility with heating amounts roughly to 70 per cent of the volume at normal temperatures, and separation to this extent was arbitrarily credited to the heater. A careful investigation was made on the performance of the heaters with reference to air separation, with results which will be explained in a later paragraph, and while they did not altogether confirm this 70 per cent value, it was arbitrarily adopted in deference to the opinion of the profession in general. Obviously it is the performance of the separator which is of primary interest.

16 Water at 212 deg. fahr. injected suddenly into the region of vacuum of the separator has considerable heat available to the vacuum and boils with explosive violence at the expense of the heat of the liquid, behaving exactly as though it were superheated. Vapor bubbles form upon minute entrained air bubbles as nuclei, and in passing off into the region of reduced pressure mechanically take some portion of the dissolved air with them.

17 By far the major portion of the separation, particularly in the elimination of the final traces of dissolved gases, is accomplished by the reduction of partial pressure and represents a simple application of Henry's law. The water flows through the separator in thin sheets exposed to a vapor mixture corresponding in pressure to the vacuum maintained. Under steady conditions of operation at a pressure reduction in the separator of 10 in. of mercury and admission temperature of 212 deg. fahr. this vapor mixture corresponds to an air content roughly 0.11 per cent by weight but 0.07 per cent by volume. The water vapor is undoubtedly in equilibrium with its liquid and as such is relatively dense and moist. With a separator water-storage volume corresponding to two minutes' operation at rating it is impossible to detect at any rate of operation a temperature depression in the vapor mixture or in the water leaving the separator below that corresponding to the vapor temperature of the vacuum. Water, vapor, and total pressure are therefore in equilibrium. The most logical conclusion to be deduced from this is that with the continuous sweep of the wet, dense water vapor from the separator the air component of the vapor pressure has been reduced by entrainment substantially to zero, and that the process has furnished a very simple and efficient means of securing such a reduction of pressure. Experiments by the writer have confirmed the results of other investigators that it is impossible to secure complete separation within a reasonable time by pressure reduction alone in excess of 80 per cent of saturation, provided the

water is refrigerated to prevent vapor evolution. This is primarily a question of apparatus, since vacuum pumps of types which permit of this experiment are invariably of low volumetric capacity at extreme vacua.

OPERATION OF SEPARATOR

18 The actual operation of a separator equipment is altogether as simple and straightforward as it can conceivably be. At a constant vacuum and admission temperature the water delivered by the separator is in exact equilibrium with the vapor-pressure temperature of the vacuum, and the increase of pumping head is exactly measured by the pressure head corresponding to the vacuum. The amount of heat in the vapor mixture and the amount of condensed vapor to be handled by the condenser can be calculated from the heat drop in the separator, and is constant at any specified rating. The circulation ratio in the condenser, which handles all the supply water in its cooling circuit, is constant for all rates of flow. Neglecting radiation, heat losses are substantially zero. The arrangement of exhauster and condensing equipment can be made sufficiently flexible to meet all conditions of application. Where a steam ejector is used, however, it is desirable to subdivide condensing capacity, using one section to condense the vapors withdrawn from the separator to relieve the ejector of the duty of evacuating them, and the other section to recover the heat of the ejector steam. The theory of the steam ejector seems best satisfied by the assumption of an entrainment and velocity compression of an inelastic fluid, making no concessions for the intrinsic energy of the fluid compressed, and a very slight one for differences of entrainment with variations in density. An ejector handling the vapor mixture from the separator under a 10-in. mercury vacuum has approximately five times the steam consumption of an ejector evacuating only the air. There are no moving parts in the apparatus except the float-level control valves in heater and separator.

HEATER TESTS

19 Careful tests were made upon the two heaters which at different times supplied the separator. The heaters were of a standard induction open-heater type with spray pans for agitating the water in its passage through the steam. The vent openings which were standard with the manufacturer were freely connected to the atmosphere. Both of these heaters were able to deliver their rated

capacity (300 and 1000 boiler hp.) of water with a 2-deg. fahr. temperature drop on atmospheric steam. Water was taken from below the water line of both of these heaters by a slow, large-volume displacement into special apparatus and was tested for air without releasing the pressure or lowering the temperature. These tests gave reductions of dissolved air of only from 10 to 20 per cent, but when a similar sample was collected by rapid displacement in an open bottle, results indicated from 60 to 70 per cent at 212 deg. fahr. and a maximum of 78 per cent at 224 deg. fahr. Within this range of temperature, water within the 300-hp. heater viewed through a glass window in a blank-flanged connection against a submerged incandescent-lamp bulb appeared to be milky with entrained air bubbles. While any generalization from such meager data as these would be presumptuous, the writer cannot help but feel that from the standpoint of dissolved-air separation the heating of water from mixture with steam is an essentially different proposition from heating from a submerged and wetted source in the matter of temperature gradient within the body of the liquid. These conclusions apply specifically to a heater of the induction type and probably would be very different for a "through" heater with its greater venting.

RESULTS OF TESTS

20 Data on the solubility of gases in water with which the writer is familiar have been worked out in volume figures and the expression of percentage values permits the direct use of such data. A percentage expression, furthermore, agrees precisely with the physical phenomena underlying the process, as the following digression will show.

21 Fig. 3 is plotted between percentages of separation as ordinates and vacuum referred to a 30-in. barometer as abscissæ for a water-temperature admission to the separator of 212 deg. fahr., corresponding to atmospheric equilibrium. This curve is the graphical average of a large number of determinations made both with the laboratory and the commercial-size apparatus. Percentage of separation is defined for the purposes of these curves as the ratio of the quantity of gases removed to that actually present in the raw water, and the calculations were made in terms of volume. Calculation in terms of weight would, however, have had identical values.

22 Fig. 4 represents the same data as Fig. 3, vacuum-temperature drop, plotted, however, to a scale of separator temperature

gradient instead of a scale of vacua. It is also a graphical average of the original data, and not a translation of the former curve into a different scale of coördinates.

23 It was found early in the experimental work that water in pipe lines and in small streams unpolluted by sewerage is likely to

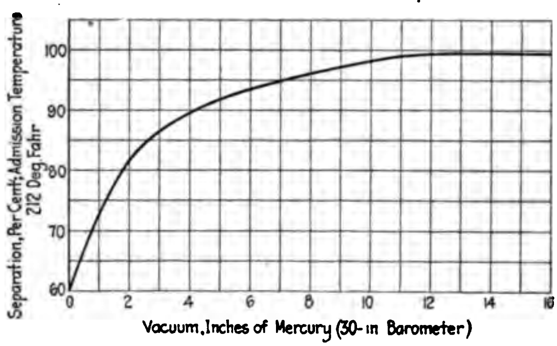

FIG. 3 RELATION BETWEEN AIR SEPARATION AND VACUUM

be supersaturated with air. It was accordingly necessary to analyze water both before and after entering the process to get the measure of the results accomplished. Entering water temperatures fell continuously within the range of 38 deg. fahr. to 70 deg., but did not in

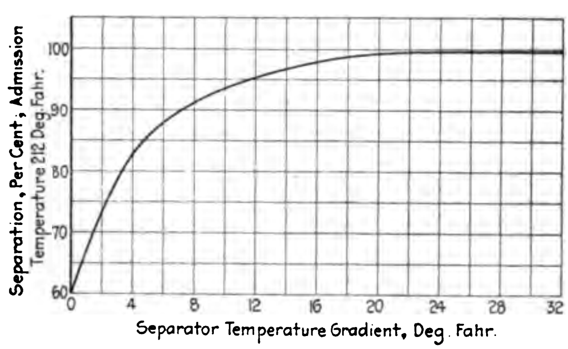

FIG. 4 RELATION BETWEEN AIR SEPARATION AND SEPARATOR TEMPERATURE GRADIENT

the least conform to solubility-table values of dissolved gases. It is significant of the physical phenomena underlying the method that at low degrees of separation (70 to 95 per cent) the test results indicate a definite degree of separation rather than the reduction of dissolved gas content to a definite small value. At separation values much in excess of 95 per cent, the quantity of gases remaining is so exceedingly small that accuracy of testing is a limiting feature in extending such a conclusion into this range. It is the writer's frank opinion that the flat asymptotic portion of the curves in Figs. 3 and 4 represents complete separation of gases, but it is impossible to prove it. The results from which these curves are plotted are as accurate as the most painstaking manipulative skill can make them, but they fall far short of the absolute precision, characteristic for instance of electrical measurements, on account of the enormous avidity of water for redissolving air. In the asymptotic range of these curves samples may be secured which do not respond to qualitative tests, although in the iodometric method a trace of dissolved

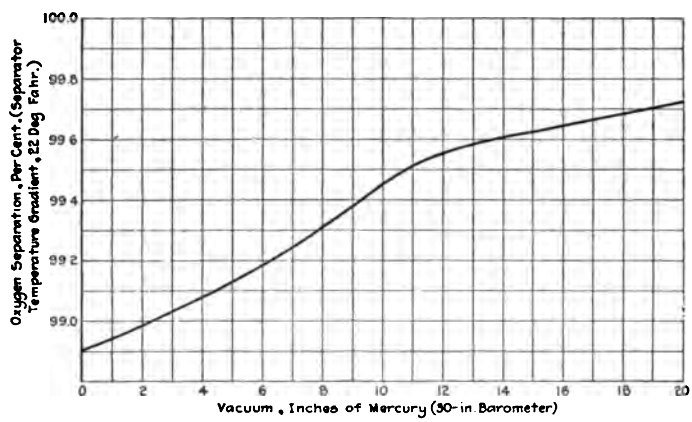

FIG. 5 RELATION BETWEEN OXYGEN SEPARATION AND VACUUM

oxygen has been always obtained either from residual solution in the water or from contamination. The iodine starch indicator is, however, one of the most sensitive known in chemistry.

24 Fig. 5 presents data on oxygen separation plotted as a function of separator vacuum with a constant separator temperature gradient of 22 deg. fahr. and illustrates the effect of variation of pressure. The separator temperature gradient of 22 deg. fahr. was selected as the minimum value that could give maximum separation. This curve is plotted in terms of dissolved oxygen because dissolved oxygen only was determined, the methods of testing for dissolved air being too inaccurate to apply over the major part of the range. It is, however, probably of equal applicability to dissolved air. One point on this curve, that of zero vacuum, deserves special mention.

It was obtained with the apparatus using a steam ejector for producing the vacuum, and was obtained by shutting off ejector steam and raising the admission temperature to the separator to give the desired drop. The cycle of operation of the separator was therefore identical with other tests except that the pressure gradient was of opposite direction. The condition within the separator, from the elimination of diffusion tendencies, is however not in the least comparable with the spraying of heated water into the atmosphere.

ELIMINATION OF CORROSION

25 The matter of corrosion has always received wide attention from the engineering profession and a vast amount of investigative work has been done on it. It was accordingly assumed at the beginning that a recapitulation of existing knowledge was all that was needed to establish the pernicious effect of dissolved oxygen. In the field of surface condensation and the removal of scale-forming materials, however, it was realized that direct experimental proof must be submitted to carry any conviction.

THEORY OF CORROSION

26 Of the gases dissolved in water, nitrogen is altogether inert, hydrogen is inert except in the ionic form, while oxygen and carbon dioxide are the agents producing boiler and economizer corrosion. Similarly, among the dissolved solids magnesium chloride and soluble nitrates are to be included. Some waters, dependent upon locality contain hydrochloric acid and are included in the same category. Irrespective of the direct agent, however, all cases of boiler or economizer corrosion are ultimately either those of oxygen oxidation or of electrolytic currents. The ionic notation is the best vehicle for explaining the mechanism of the oxidation process, because it occurs at large dilutions and high temperatures, the corrosion due to electrolytic currents being only a correlative phenomenon where a metallic current is provided for the circulation of electric currents.

27 Water ionizes according to the ionic theory into the following:

$$\begin{array}{lrcl}
(a) & 2H_2 & \rightleftarrows & 4H \\
(b) & 2H(OH) + O_2 & \rightleftarrows & 4(OH)' \\
 & & & \downarrow\uparrow \\
(c) & & & 4\,H_2O \quad \ldots\ldots\ldots \quad [1]
\end{array}$$

28 Equation [1] is a reversible reaction which may proceed in either direction accordingly as the conditions of equilibrium demand. It represents the simple fact that water ionizes into hydrogen ion ($\dot{H}$) and hydroxyl ion $(OH)'$, and is written in the oxidation reduction form because it contributes the additional information that molecular or atomic hydrogen ionizes into hydrogen ion in solution, and that oxygen going into solution ionizes water into hydroxyl ion. The equilibrium condition for each of the conditions of equilibrium are different, but the condition of equilibrium between water and its ions is significant. At boiler pressures and temperatures water exists in the dihydrol form in equilibrium with its monohydrol form, and the ionization is much greater than at normal temperatures.

29 Iron from its greater electrolytic solution pressure is able to displace hydrogen ion from an ionized solution of water.

$$Fe + 2\dot{H}(OH)' \rightleftarrows \ddot{F}e(OH)'_2 + H_2 \quad . \; . \; . \; . \; . \; . \quad [2]$$

This equation also has an equilibrium condition which from practical considerations of boiler corrosion is minimal for the reason that hydrogen ion is deposited on the surface of the iron plate as atomic hydrogen and by polarizing it interrupts equilibrium. So far oxygen has played no part in the solution of the iron, but if the solution from Equation [2] should be analyzed for iron, ferrous oxide (FeO) from the decomposed ferrous hydroxide would be found, even if no oxygen had been present in the solution.

30 In the presence of dissolved oxygen the nascent atomic hydrogen deposited on the iron electrode plate combines with the dissolved oxygen to form water:

$$2H_2 + O_2 = 2H_2O. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [3]$$

Equation [2] is then permitted to go to equilibrium since polarization has been removed and oxygen appears as an accelerant to the process, and the corrosive effect simultaneously increases.

31 Ferrous hydroxide in ionized solution is oxidized by oxygen in a manner identical with *b* of Equation [1]:

$$4\ddot{F}e(OH)'_2 + O_2 + 2H(OH) = \dddot{F}e(OH)'_3 \quad . \; . \; . \; . \quad [4]$$

Equation [4] is also written in the oxidation-reduction form. The ratio of ferrous to ferric ions is of the magnitude of 10^{17}, although it is complicated by other equilibrium conditions of complex ferric and ferrous ions which might conceivably change the relations very greatly at boiler temperatures. While from the equilibrium constant the amount of ferric ion which would exist in the boiler solution is small, the ferric hydroxide is both very feebly ionized and unstable and continuously precipitates out of solution. The reaction of Equation [4] continuously displaces the equilibrium of Equation [2], although the actual amount of ferric ion at any time in the boiler water is very slight. The process is significant only in that it is continuous.

32 The behavior of magnesium chloride or any other compound producing a stable negative ion which hydrolizes with water is exactly analogous.

$$\ddot{\mathrm{M}}\mathrm{gCl}'_2 + 2\dot{\mathrm{H}}(\mathrm{OH})' \rightleftarrows \ddot{\mathrm{M}}\mathrm{g}(\mathrm{OH})'_2 + 2\dot{\mathrm{H}}\mathrm{Cl}' \quad . \; . \; . \; [5]$$

The ionization of Equation [5] includes both primary and secondary values. Magnesium hydroxide is precipitated, but the hydrochloric acid is ionized almost completely. Iron dissolves in the ionized hydrochloric-acid solution exactly as in Equation [2] except that the concentration of hydrogen ions is increased and the resulting ferrous chloride is itself more highly ionized than ferrous hydroxide and is hydrolyzed by water.

$$\mathrm{Fe} + 2\dot{\mathrm{H}}\mathrm{Cl}' \rightleftarrows \ddot{\mathrm{F}}\mathrm{eCl}'_2 + 2\mathrm{H} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; [6]$$

$$2\mathrm{H}_2 + \mathrm{O}_2 = 2\mathrm{H}_2\mathrm{O} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [7]$$

$$\ddot{\mathrm{F}}\mathrm{eCl}'_2 + 2\dot{\mathrm{H}}(\mathrm{OH})' \rightleftarrows \ddot{\mathrm{F}}\mathrm{e}(\mathrm{OH})'_2 + 2\dot{\mathrm{H}}\mathrm{Cl}' \quad . \; . \; . \; . \; [8]$$

Oxygen has intervened to destroy the polarization according to Equation [7] and will disturb the equilibrium of Equation [8] precisely as in Equation [4]. Hydrochloric acid in its ionized form remains in solution in the quantities in which the chloride ion was introduced by magnesium chloride, and its action is cyclic in the presence of dissolved oxygen, and increases with increasing concentration. The essential danger of magnesium chloride over other chlorides lies in the insolubility of magnesium hydroxide in Equation [5]. The equilibrium condition of Equation [6] permits greater initial corrosion, however, than the equilibrium condition of Equation [2].

33 Nitrate ions are unstable at boiler temperatures; hence nitrates are not the same serious corrosion factors as chlorides. Carbonic acid contributes to the acidity of boiler waters and, as a corollary, to the electrolytic action, and being volatile, mingles with the steam. In this sense whatever portion of it that returns with the condensate is cyclic in its effects. From its ability to dissolve

ferrous carbonate as ferrous bicarbonate it tends to concentrate all the products of pipe-line corrosion in heating systems in the boiler itself, and makes the apparent boiler corrosion much greater than it really is. Certainly it is a very undesirable constituent of boiler waters but not nearly so dangerous as dissolved oxygen.

CORROSION IN STEAM BOILERS

34 A high-pressure steam boiler is so complicated a chemical factory that there can be no general panacea for its ailments, but it is thought that the removal of dissolved oxygen and carbon dioxide is a most decidedly effective step in the elimination of corrosion. Since it is a basic attack at the root of the evil, it will contribute to the success of all methods of eliminating corrosion which engineering experience has found good, but as long as boilers are made of iron and fed with water, some corrosion is inevitable. Boiler compounds, alkalinity, electric potentials, etc., are involved with such an infinitely variable condition of equilibria within the boiler that their success in practice can only be partial, while the effect of the removal of dissolved gases is aimed at the equilibria conditions themselves and is self-regulating.

CORROSION IN ECONOMIZERS

35 The internal corrosion of economizer tubes is a phenomenon closely parallel with that of the corrosion of boilers. The corrosion here is more severe and localized and the corrosion products more alarming to the eye, although this is due primarily to a superficial change in the water of crystallization of the resulting hydrated iron oxides. The increasing cost of coal is bound to increase the use of economizers, but unfortunately both the cast-iron and the steel-tube types are unfavorably regarded, the former because of its uncertain factor of safety and the latter from its liability to corrode. Necessity, however, has already developed the steel-tube economizer closely along the type of header and tube construction used in boilers, and economizer boiler units are now available which show a decided increase in evaporative efficiency and reduction of cost over straight boiler units. The removal of carbon dioxide and oxygen will undoubtedly prolong the life of cast-iron economizers greatly, and will make the installation of steel-tube economizers economically practical and conservative. On this basis the gas-

removal process will have technical significance in the future greater than it has at present.

ELIMINATION OF SCALE PRECIPITATION

36 Carbon dioxide exists in water as carbonic acid (H_2CO_3), which is capable of combining with carbonates to form bicarbonates; thus,

$$Ca(CO_3) + H_2(CO_3) = CaH_2(CO_3)_2 \qquad [9]$$

The carbonates of calcium and magnesium are soluble in the proportions of 23 and 106 parts per million by weight as carbonates, or in other words in water free from carbon dioxide, but soluble as bicarbonates in any proportion in which they are apt to exist in natural waters. Bicarbonates of either of these elements are metastable and readily part with their carbonic acid under the conditions existing in the separation equipment, being reduced to carbonates and precipitated in excess of their solubility values. Sodium carbonate and bicarbonate are soluble in all proportions, although half-combined carbon dioxide is also removed.

37 The air-separation equipment has been used to eliminate scale precipitation and has been tried out on two waters; one from the water mains of the city of Pittsburgh which had been drawn through a filtration plant from the Allegheny River; the other from a small local stream that supplies the Jeannette, Pa., works of the Elliott Company. Both waters were from the same watershed and had the same general analysis, as shown below:

	Allegheny City of Pittsburgh. Parts per million	Surface Stream Jeannette, Pa. Parts per million
$CaCO_3$	24.92	15.7
$CaSO_4$	130.53	18.2
$MgSO_4$	4.59	15.7
$NaCl$	3.63	3.5
Na_2SO_4	1.87	6.6
$FeSO_4$	9.14	3.0
Al_2SO_4	19.07	18.3
$NaOH$	4.94	0.0
Organic volatile	27.00	12.0
Total	225.09	93.0
Total as determined	214.00	87.0

38 A series of analyses on the Jeannette, Pa., water for the period under which these experiments were run gave the following variation:

Date	$CaCO_3$ Parts per million	Analyst
6/27/19	53.8	Spiegelman
7/16/19	61.6	Spiegelman
9/12/19	63.7	McDermet
11/10/19	50.7	Wertheimer
1/12/19	15.7	Wertheimer

These variations are to a certain extent seasonable, and depend on the weather conditions over the water shed. No experiments were performed during flood conditions. The hypothetical combinations in formulating the results of the analyses are entirely as outlined by Jackson in his work entitled Boiler Feed Water.

39 The effect of the process in eliminating calcium carbonate was initially discovered by the soap-titration test of Clark on a cooled and decanted sample of water which had been put through the process. It was difficult, however, to determine the dilution of the water with exhaust steam in the heater and results from the method were only indicative. Careful gravimetric tests over the period June 1 to November 1, 1919, supplemented by careful volumetric titrations with phenolphthalein and methyl orange indicators, gave a uniform series of results running from 18 to 25 parts per million of $CaCO_3$, which agreed closely with solubility values. While exhaust-steam dilution appeared also in these results the reduction was greater than could possibly be accounted for by dilution, and agrees closely with solubility values. The water after passing through the separation process invariably gave a pink color with phenolphthalein, although it gave no coloration before, indicating thereby the elimination of free carbon dioxide.

40 Technical boiler-water analysis is, however, so much a matter of hypothesis that such results as the foregoing may be regarded as confirmatory but not conclusive. Arrangements were made to evolve the residual carbon dioxide from the sample of processed water, and determine it volumetrically in a modified Haldane gas-analysis apparatus, but freezing of the watershed so lowered the carbonate content of the water that the amount of evolved carbon dioxide would have been below the limits of accurate gas volumetric work. This part of the work is necessary to make the results conclusive and has been necessarily deferred until conditions are

more propitious. It will also undoubtedly be necessary to continue the experiments upon a variety of waters to reach a definite conclusion.

41 Calcium carbonate precipitates out of the hot solution of the process as a colloid, which coagulates very rapidly into a flocculent precipitate and becomes granular by the time it arrives in the boiler, where it is deposited as sludge. Scale taken from the boilers during the operation of the separation equipment gave only traces of calcium carbonate, although the sludge from the bottom of the boilers contained a sufficient amount of it to be effervescent when treated with acid. No trouble was ever experienced from scaling of the boiler feed lines.

CONDENSATION OF AIR-FREE STEAM

42 The surface condensation of steam involves three distinct physical steps: the conduction of heat through a definite thickness of metal — the tube wall; and two boundary conditions, the transfer of heat from a vapor to a metal, and the transfer of heat from a metal to a liquid. In the surface condenser this latter condition is dependent upon the velocity, viscosity and temperature of the cooling water and the laws of the heat transfer are not definitely understandable. The other boundary condition is complicated in practice because the area exposed to steam flow is surrounded by a rarefied air film so that in reality two heat transfers are involved: from the steam to the air film and from the air film to the metal surface. The elimination of this film of non-condensible gas and the attainment of an ideal boundary condition is all that the air separation can possibly attain. It is to be expected that it will increase the conductivity by eliminating the resistance of the air film and increase the vacuum by reducing the partial vapor pressure, but the intricacy of the experimental data is very apt to obscure the simple physical fact involved.

43 In the experiments on surface condensation the boilers were fed with water which had been put through the air-separation process and which was practically air-free; and high-pressure steam was conducted to a cast-iron baffled cone bolted to the steam inlet of the condenser, expanded to condenser pressure, and allowed to enter the condenser with considerable velocity but no impingement. Two surface condensers of very dissimilar types were tested under these conditions.

44 The first was of an antiquated type designed for engine service, consisting of 515 tubes, $\frac{5}{8}$ in. O.D., No 18 B. W. G., with an active length of 6 ft., all arranged in a single pass. It was without steam baffles, steam lanes, rain plates or air-pump baffles. The total area on the conventional method of figuring was 550 sq. ft. It was served by a Knowles steam-driven mixed-vacuum pump with cylinder 8 in. by 10 in. by 12 in. The data on these tests are not included in this paper because they cannot be interpreted in the light of modern performance.

45 The second was one furnished by the C. H. Wheeler Manufacturing Company after a design which had been built for the Bureau of Mines for experimental purposes, and was the type of unit that would ordinarily be furnished for a 100-kw. turbine. It consisted of 228 tubes, $\frac{5}{8}$ in. O.D., No. 18 B. W. G., with an active length of 8 ft., arranged in three water passes; calculated area, 301.7 sq. ft. It was furnished with a steam baffle and air-pump suction baffle, and was without rain plates, steam lanes or differential spacing. It was equipped with a motor-driven centrifugal condensate pump with condensate-water-sealed glands, and a 16-in. by 18-in. Rotrex dry-vacuum pump. This pump was separately motor-driven at constant speed, was arranged for separate sealing-water injection, and at its operating speed had a volumetric capacity of 63.3 cu. ft. per min.

46 The tightness of the condenser was maintained by three methods: filling the steam space with water under pressure and watching for leaks; filling with compressed air under a pressure of about 10 lb. per sq. in. and painting with soapsuds; and pump-down tests with a laboratory high-vacuum pump. If pump-down tests showed a leakage greater than 0.1 in. mercury per hour, tests were never performed. Loss of vacuum under pump-down conditions was always a straight-line function of the time over a period of several hours. Curiously, however, tests on the condenser operating with air-free steam showed higher vacuum than it was possible to secure on pump-down tests, and the loss of vacuum in such operation followed an entirely different law.

READINGS TAKEN IN TEST

47 Vacuum was measured at the air-pump suction and at the nozzle, a special gage being made by welding a connection to the top of a U. S. Weather Bureau mercurial barometer. The air entering the top of the mercury column was chemically dried, the vacuum

was never released, and the piping was all of copper tubing. This gage had a reading accuracy of 0.01 in. of mercury, and it is felt that the accuracy is consistent. Comparison was made with a barometer of a type identical with the vacuum gage mounted adjacent to it, and checked weekly with the standard instrument of the Weather Bureau in Pittsburgh by means of a portable compensated aneroid. There was no correction between the gages for temperature; capillarity correction was applied, but those for altitude and longitude were neglected.

48 A thermometer well was inserted into the air-pump suction in the hope that it would furnish a check on the gage. The curious situation developed that, while a thermometer well drilled in a flange of the pipe wall which was insulated with hair felt maintained within 0.1 deg. fahr. of the temperature of the vacuum, the thermometer well projecting into the pipe showed a temperature several degrees higher at extremely high vacuum (i.e., 0.01 to 0.05 in. of mercury of the theoretical absolute), and equaled the temperature of the pipe at lower vacuum. The only plausible explanation was that the air particles were carried along in the rarefied vapor stream with a translatory motion, but with a pressure so close to zero that all sensible molecular vibratory motion had ceased.

49 The temperature of the condensate was measured by a thermometer well immediately below the exit from the condenser, mercury-filled and equipped with a certified thermometer reading to 0.1 deg. fahr. With the centrifugal pump the condensate removal was continuous. Heat transfer was determined from the temperature rise and weight of cooling water. Cooling water was delivered under constant gravity head through a regulating valve, and after going through the condenser passed through a baffled weir box and over a 90-deg. weir. This weir was sharp-edged, cut from $\frac{1}{8}$-in. brass plate and mounted against a rigid iron frame with a commercially plane surface. The weir plate was made precisely according to Barr's specification, and its indications were corrected according to Barr's curve of coefficients. Weir volumes were translated to weights through the specific weight of water at the temperature of flow. Weir heads were measured by a hook gage rigidly mounted in a vertical plane and dipping into the free surface of the water above the weir. The gage point could be observed from both below and above the surface. The reading accuracy was 0.001 ft. Taken altogether it is thought that the determination of water weight represented the highest attainable accuracy.

50 Cooling-water temperatures were measured by calibrated thermometers reading to 0.1 deg. fahr. The logarithmic mean temperature difference was used in all calculations.

51 A method of checking this heat transfer was adopted which is curious in that it was successful, although the apparent odds against it were rather high. Steam was admitted to the expansion valve on the condenser after having been throttled from boiler pressure to 85 lb. gage by an automatic reducing valve, and in its passage from the boiler to the reducing valve the quality was measured. The condensate for an entire test was weighed in a large weighing tank. The sealing water from the air pump was weighed, its temperature rise measured, the amount of condensed steam calculated, and the amount of heat withdrawn with the air-pump vapors determined. The expansion of the steam into the condenser was assumed as one of constant heat content. This obviously did not represent the actual nature of the expansion, yet it was difficult to conceive of an appreciable quantity of heat being lost in the process. This heat value of the steam was corrected for initial quality, heat withdrawn by the condensate and by the air-pump vapors, but no correction was applied for radiation from or to the condenser shell. From this calculation heat transfer was computed and compared with that obtained from cooling-water measurement. The agreement between these values is the part that is curious. In a total of 52 tests the maximum difference is 35 per cent, the agreement perfect in two instances, and the average difference 7 per cent. The conductivities varied from 129 to 949 B.t.u. per sq. ft. per hr. per deg. fahr. difference in temperature.

52 In order to provide for as precise a comparison as possible between tests with air and those without air, a series of tests was run in which air was not extracted from the water before it entered the boiler, although it was heated up to atmospheric temperature in the open heater of the separation equipment. Air was not admitted directly into the condenser because no means of measuring the quantity admitted was known which would be of consistent accuracy with the remainder of the experiments. Such tests as either of these series are evidently not representative of operating conditions, in that they were performed upon a condenser of air-tightness far in excess of anything attained in commercial operation. They are an attempt toward an ideal condition of condensation, however, and are offered as being of specific interest because they apply to an actual condenser.

53 On account of the volume of the data of these tests, some 3400 tabular values, it is not practicable to publish the complete log in the present paper. However, a specimen log of two tests condensed to about one-third of its original volume is given in Table 1 for the use of those who wish to make a numerical comparison between the two groups, and the remainder of the data are presented in the form of curves, between such grouping of coördinates as are conventional in the representation of condenser tests. Obviously such an attempt to reduce a relation between many variables to plane coördinates leaves much to be desired. To each of these graphs such explanations will be added as are pertinent to it.

TABLE 1 SPECIMEN LOG OF TESTS (CONDENSED)

	Separated	With Air
Date	8/23/17	8/23/17
Duration, hours	1	1
Barometric pressure, inches Hg	28.96	28.98
Condensate, lb.	2,988	3,201
Condensate, lb. per sq. ft. per hr.	9.91	10.61
Condensate temperature, deg. fahr	89.1	87.1
Temperature cooling in. deg. fahr.	67.8	64.2
Temperature cooling out, deg. fahr.	82.8	80.4
Heat rise, cooling, B.t.u.	14.96	16.16
Weir head, ft.	0.682	0.682
Weir volume, cu. ft. per min	5.852	5.852
Specific weight, lb. per cu. ft.	62.20	62.23
Weight cooling, lb. per min.	363.8	364.0
Weight cooling, lb. per hr.	218,322	218,424
Total heat to cooling, B.t.u.	3,266,097	3,538,469
Mean temperature differences log, deg. fahr.	76.7	73.6
Vacuum in condenser, inches Hg.	28.60	28.57
Temperature of vacuum, deg. fahr.	89.5	90.2
Absolute pressure in condenser, lb. per sq. in.	1.40	1.43
Heat transfer, B.t.u. per sq. ft. per hr. per deg. fahr.	846	706
Vapor pressure of condensate, inches Hg.	28.62	28.70
Cooling-water velocity, ft. per sec.	8.47	8.47
Cooling-water ratio	73	68

54 Fig. 6 is plotted between conductivity, B.t.u. per sq. ft. per hr. per deg. fahr. difference, as calculated from cooling-water measurements, and vacuum on a 30-in. barometer. It identifies primarily the range within which these experiments lie. High conductivity is largely a matter of juggling, and does not usually accompany high heat transfer. The high values which have been secured by other investigators have been secured by very low temperature gradients which accompany low transfers. At the rates of transfer which are economically necessary in condenser work it is impossible to pump circulating water fast enough to keep the temperature gradient down. As Orrok aptly says in a previous paper before

this Society,[1] "Only as much heat may be transmitted as is actually present." It appears to be impossible to raise conductivity without recourse to methods that are freakish. Most of the heat transfer in the air-free tests took place in the upper tube banks; the transfer of heat was enormous, but without the additional refrigerating effect of the lower tubes it would have been impossible

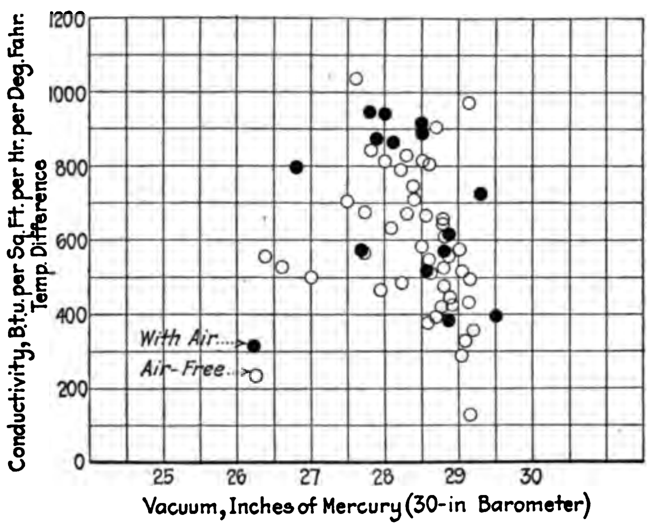

FIG. 6 RELATION BETWEEN CONDUCTIVITY AND VACUUM

to secure a satisfactory vacuum. With the necessary refrigerating area the apparent conductivity was decreased.

55 The following data are significant of the range covered by the tests as a whole:

	Maximum	Minimum
Condensation rate, lb. per sq. ft. per hr......	18.87	2.82
Cooling-water ratio.........................	174.5	8.7
Cooling-water velocity, ft. per sec...........	11.94	2.77
Cooling-water temperature, deg. fahr........	90.9	35.0
Cooling-water rise in temperature, deg. fahr...	122.5	6.8

56 Fig. 7 is plotted between barometer minus the vapor pressure of the condensate as ordinates and vacuum as abscissæ, a solid line representing equilibrium being drawn thereon. The refrigerat-

[1] The Proportioning of Surface Condensers, Trans.Am.Soc.M.E., vol. 38, p. 864.

ing effect is very decisively indicated. The condenser circulation was contraflow, three-pass, and in the air-free tests no transfer could be detected in the first pass. Simultaneously the vacuum drop in the condenser fell from 0.2 in. of mercury for 5000 lb. condensation per hour and 29-in. vacuum air to zero at all rates up to 5700 without air. Unfortunately these data were not systematically taken, since it was not known at the outset just where the advantages of air separation would appear. Ordinarily with steam-air mixtures and very efficient air removal from the condenser the phenomenon

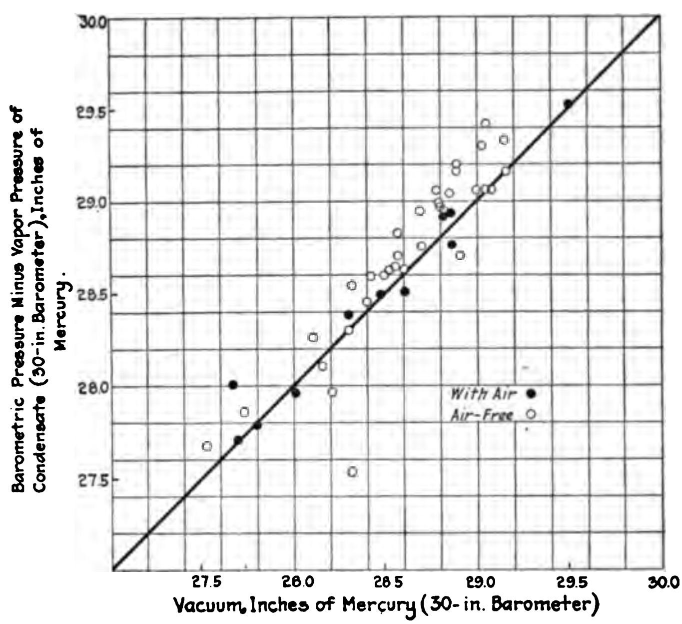

FIG. 7 RELATION BETWEEN BAROMETRIC PRESSURE MINUS VAPOR PRESSURE AND VACUUM

is exactly reversed, the maximum rise and the maximum transfer taking place in the lower tubes. It was possible by closing a valve in the air-pump suction to operate the condenser easily for an hour with a drop of 0.05 in. of vacuum at vacua as high as 29.06 in. When the valve was closed the condensate temperature immediately came into equilibrium with the vacuum. When, however, the loss of vacuum amounted to 0.1 in., the further drop was very rapid, amounting to a complete loss in about ten minutes. The law of loss of vacuum was not definitely determined but it was never a straight-line function of the time. Under the conditions of condensate removal there was never any opportunity for "drowned" tubes.

57 Fig. 8, giving the relation between condensate depression (temperature of the vacuum minus temperature of the condensate) and conductivity, shows again and more pronouncedly than Fig. 7 the effect of refrigeration in the lower tube banks.

CONCLUSION FROM CONDENSER TESTS

58 A careful consideration of the data of these tests in all their phases has led the writer to the following conclusions, the basis

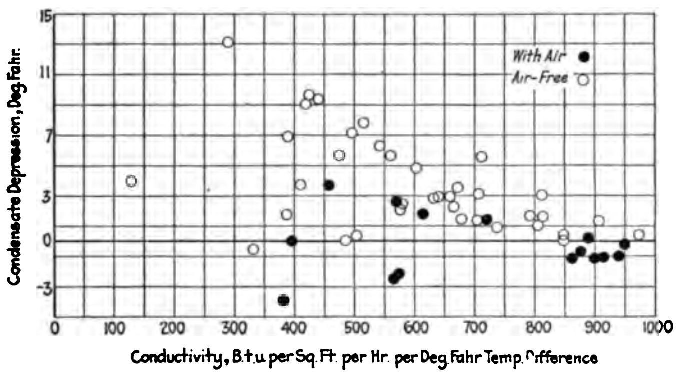

FIG. 8 RELATION BETWEEN CONDENSATE DEPRESSION AND CONDUCTIVITY

for forming which he has unfortunately not been able to make altogether clear:

a A comparison between individual tests on air-free steam and steam with air which are sufficiently alike in the variable quantity values or which it has been possible to reduce to a satisfactory basis of comparison, indicates an increase in conductivity of about 20 per cent figured upon the entire area of the condenser

b It is possible to reduce the air component of the vapor pressure in a tight condenser to within an immeasurable amount of zero, and simultaneously eliminate all dead areas in the condenser due to air blanketing

c The fundamental problem in condensation of air-free steam is not to conduct the heat but to get rid of it after it is conducted. The economical limit of permissible hydraulic losses will limit conductivities to substantially the same values that are used in commercial practice today.

d In the condensation of air-free steam two processes are involved: the actual condensation, and the refrigeration of the liquid, which take place in different zones and are distinctively different processes. One process disposes of the heat and the other produces the vacuum, and each is therefore as indispensable as the other. In the zone of condensation heat transfers reach enormous values. In the zone of refrigeration both heat transfer and heat available are very low, and the conduction is from a liquid to a liquid. In this zone a very close spacing of tubes is permissible and any change in design which would substitute water films for water droplets would be a constructive improvement. It is possible to isolate from the data of the tests the fact that the zone of condensation is limited at least to 30 per cent of the tube area, and the reduction of vacuum loss indicates it to be much less. The ratio of temperature rise between the different circulating-water passes is radically altered from operation with very slight amounts of air.

e From the radically different rates at which heat is transmitted in the different zones, it is impossible in tests on condensers of different types with air-free steam to arrive at any data from which to formulate or verify laws on the conduction of heat. Neither is it possible to apply such laws without an empirical assumption on the distribution of heat between the various tube banks. These tests, while they truly represent the condensation of air-free steam, contribute no information on numerical values of actual conductivities.

SUMMARY

59 It is possible from the data of this paper to design apparatus which will accomplish the separation of dissolved gases from boiler water as completely as it may economically be desired to do it. The process consists of suddenly injecting heated water into a region of vacua and boiling it at the expense of the heat of the liquid. Control of the process is secured through regulation of the temperature drop of the water in passing into and through the region of vacua,

but the underlying laws are more complex and depend partly upon a reduction of solubility by heating, partly upon the mechanical action of vapor evolution in boiling, and partly upon the reduction in partial pressure of the air component of the vapor mixture in the region of vacua.

60 Recourse is had to the ionic chemical notation to demonstrate that dissolved oxygen and dissolved carbon dioxide, both of which are removed by the process, are responsible for the dangerous features of boiler and economizer corrosion; the former by destroying the inhibitive polarization and displacing the equilibria of the inevitable slight solution of iron in water, the latter by increasing the concentration of hydrogen ion and with it the initial tendency of the iron to dissolve.

61 Experiments upon two waters of somewhat analogous constitution in dissolved solids indicate the separation of both dissolved and half-bound carbon dioxide and the simultaneous precipitation of dissolved bicarbonates in excess of their solubility as carbonates.

62 Experiments on the surface condensation of air-free steam indicate a gain of 20 per cent in conductivity figured upon the condenser area as a whole, and a natural division of condensing area into condensing and refrigerating groups.

DISCUSSION

C. M. Garland. There are many pieces of boiler room equipment that have not reached the stage of development that might reasonably be expected. The economizer as it stands today is one piece of equipment which is certainly far from the desired goal. The principal cause for this lack of development in this very important piece of apparatus, aside from a general lack of interest in the subject, is the corrosive action of dissolved gases in water. This corrosive action has rendered it necessary in the economizer, as designed today, to use cast-iron tubes. These cast-iron tubes mean tubes of large diameter, consequently for high pressure, tubes of heavy weight. The necessity of using cast-iron has limited the pressures that can safely be carried on economizers and has rendered the economizer bulky, clumsy and to a large extent, impracticable.

If the author's investigations and apparatus ultimately lead to the elimination of corrosion in economizer tubes, which apparently is to be expected, he has eliminated one of the most difficult problems in economizer design and will render possible the use of steel tube economizers. Steel tube economizers can be designed to occupy about one-eighth of the space now allotted to economizer equipment. They will greatly reduce the cost of economizer installations and make it possible for practically every plant to take advantage of the increased economy possible through the use of the conomizer.

While the corrosive action of dissolved gases has had the most disastrous effect on the heating surface of economizers, yet pipe lines and boiler heating surfaces often suffer from this action.

The author's investigations would seem to have placed the responsibility for this corrosion on the dissolved gases and his apparatus apparently removes these gases. The apparatus is simple to operate, inexpensive to install and would seem to be a happy solution of the problem.

Geo. H. Gibson. This paper states that the method of separating dissolved gases from water by first heating the water and then subjecting it to a vacuum was originated by W. S. Elliott. However, Wm. H. Walker, in a paper read before the Iron and Steel Institute in 1909, pointed out that pitting might be entirely avoided by removing air from the feedwater before its introduction into the boiler and that this might best be done by the employment of an open feedwater heater, or better still, by a feedwater heater connected to the dry vacuum pump of a condenser.

Also, in a report submitted in 1909 to the Goldfields Water Supply, of Perth, Western Australia, by a committee of experts employed to study corrosion problems arising in connection with the famous 350-mile, 30-in. Coolgardie pipe line, it was concluded that "If the oxygen of the contained air (air in the water) can be removed, corrosion will be arrested, and when the air is once removed steps should be taken to prevent its reëntering the water." Filtering the water through metallic iron, boiling and boiling under partial vacuum were tried. The idea of heating the water was finally given over and it was proposed to pass the water into a vacuum in a finely divided state, at the ordinary temperature. An experiment was made at 17 deg. C. (62.6 deg. fahr.) in which the vacuum was kept perfect, except for the pressure of the water vapor at 17 deg. C.; which pressure is only 18 mm. or 0.72 in. of mercury. The dissolved oxygen was found to be reduced from 6.5 to 1.4 cc. per liter. The experts believed that a perfected, large-scale apparatus would give even better results by giving better spray and vacuum and submitted an opinion, based upon the experiment, in favor of this process over chemical treatment. The report was accompanied by a sketch of such a deaërating apparatus.

Throughout the present paper the degree of air removal is expressed in percentage of the air originally present in the water, rather than in measures of weight or volume. It is not certain, therefore, how the results compare with those obtained by other methods. It may be of interest, however, to add that by heating water in a Cochrane open feedwater heater under atmospheric pressure the oxygen content has been reduced slightly below 0.2 cc. per liter with the water at 210 deg. fahr. at the outlet. It was found essential that the heater outlet temperature be near 212 deg. fahr., the oxygen content increasing rapidly at lower temperatures. In one test, for example, with the water at 204 deg. fahr. at the heater outlet, the oxygen content was 0.39 cc. per liter. The maintenance of high temperature in a heater operating under atmospheric pressure requires careful attention to water distribution and to venting. To further insure perfect air removal, the water in the storage space, of the heater can be agitated or boiled by means of submerged steampipes, either perforated or unperforated.

The fact that effective separation of air can be obtained without the complication of vacuum appliances, including an air pump, affects the practical value of the apparatus described in the paper: The vacuum method delivers the water at a temperature below 212 deg. fahr. Where this is desired, as where flue gas economizers are employed, the same result can be obtained by passing the water from an open heater at atmospheric pressure through a heat exchanger in which it will impart some of its heat to the water going to the heater. A heat exchanger of this kind would have about the same coefficient of heat transmission as given by the author for the small condenser through which the vapors from the separator pass on their way to the air pump. The air pump, however, would be eliminated.

Certain of the statements in the paper appear to require either correction or explanation. For example, in Par. 11, it is stated that "A glance at the solubility tables for dissolved air will indicate that water at atmospheric boiling point still retains a considerable portion of dissolved air." However, the results both of Winkler and Bohr and of Bock show that at 100 deg. C. and 760 mm. total pressure, the solubilities of both oxygen and nitrogen in water are zero. Possibly the author has used tables of solubilities at a partial air pressure of one atmosphere, for which condition, Winkler gives the solubility of air in water at 100 deg. C. as 0.01105 by volume. However, if a sufficient flow of steam-air mixture from a heater be secured by venting, the partial air pressure in the heater can be held very close to zero. The statement in Par. 15 that "the reduction of solubility with heating (in open heaters) amounts roughly to 70 per cent of the volume at normal temperatures, and separation to this extent was arbitrarily credited to the heater" should not be accepted as generally valid.

The remark in Par. 16 that "Vapor bubbles form upon minute entrained air bubbles as nuclei, and in passing off into the region of reduced pressure mechanically take some portion of the dissolved air with them" can hardly be reconciled with the usual conceptions of the kinetic theory of gases.

Par. 36 states that "The carbonates of calcium and magnesium are soluble in the proportions of 23 and 106 parts per million by weight as carbonates, or in other words in water free from carbon-dioxide, but soluble as bi-carbonates in any proportion in which they are apt to exist in natural waters. Bi-carbonates of either of these elements are metastable and readily part with their carbonic acid under the conditions existing in the separation equipment, being reduced to carbonates and precipitated in excess of their solubility values." If this means that magnesium carbonate can be precipitated in the way suggested, consideration should be given to the fact that the monocarbonate of magnesium is quite soluble and precipitation is usually obtained only by transforming it to the hydrate.

In Par. 37 are given analyses of two waters treated by the heat-vacuum method. Neither of these appears to have contained any magnesium carbonate, and the amount of calcium carbonate in one is below the limit of solubility as just stated and of the other only slightly above. Neither was, therefore, suitable for testing the value of this method for precipitating scale-forming matter. It may be added that practically complete removal of calcium carbonate has been obtained by heating water to 212 deg. fahr. in open heaters provided with large tray surface.

Par. 48 states that a thermometer inserted in a stream of steam vapor moving at high velocity under vacuum will show a temperature several degrees higher than that given for saturated steam at the same pressure in steam tables. This has been explained as due to the adiabatic compression of the vapor against the thermometer bulb. Other investigators have pointed out that the velocities are usually not sufficient to account for several degrees difference in temperature and ascribe the descrepancies to faulty thermometry.

The explanation given in the paper, namely, "The only plausible explanation was that the air particles were carried along in the rarified vapor stream with a translatory motion, but with a pressure so close to zero that all sensible molecular vibratory motion had ceased" needs either elucidation or deletion.

In Par. 58, under *d*, it is stated that "In the condensation of air-free steam two processes are involved: the actual condensation, and the refrigeration of the liquid, which take place in different zones and are distinctly different processes." Once the steam has been condensed any further refrigeration of the liquid, however, is objectionable, and should if possible be avoided, although it is advantageous to refrigerate the air-steam mixture passing to the air pump, which should be the object sought in the design of the condenser, rather than the refrigeration of the condensate.

F. N. Speller.[1] The method of mechanical deaëration described by the author affords another means whereby the gases may be removed from water, and illustrates again that the practical solution of the corrosion problem is mainly a question of special engineering design. Having been engaged in investigations on the cause and prevention of corrosion for the past 15 years it is gratifying to see the growing interest manifested by engineers in this important matter. At temperatures below 180 deg. fahr. our experiments have shown that the last ½ cc. per liter of oxygen is much more difficult to remove by mechanical means, while above 180 deg. fahr. equilibrium can be easily obtained when the temperature approximates the boiling point for the prevailing pressure. Perhaps the author can furnish some data as to the minimum amount of superheating required to remove all the gases at temperatures down to 130 deg. fahr. The electrolytic theory of corrosion was first advanced by Dr. W. R. Whitney in 1903.

The oxygen contents (which is closely proportional to dissolved air), vary from 5 to 10 cc. per liter in natural water according to seasonal temperature. As it is simple enough to remove most of the gases without any special apparatus other than a well-vented heater, it is perhaps better not to consider percentage removal of gases but rather as to how much remains after leaving the heater and how best can this residual oxygen be removed. The variation in reduction of dissolved air reported from 10 to 70 per cent at 212 deg. fahr. (Par. 19), if the writer interprets these figures correctly, indicates

[1] Metallurgical Engineer; National Tube Co., McKeesport, Pa.

the use of an undersized open heater. It would seem more desirable to plan to remove dissolved gases as far as possible in the heater. An open heater operating at the National Tube Company, McKeesport, Pa. at a temperature of 170 deg. fahr. left a daily average of 2.55 cc. per liter of free oxygen in the feedwater, the theoretical minimum for these conditions being 2.35 cc. per liter. Another series of tests made at another plant with an open heater temperature of 204 deg. fahr. gave us 0.75 cc. residual oxygen, the theoretical minimum under these conditions being 0.50 cc. per liter. A test on a large open heater at the Philadelphia Electric Company reported to me by Mr.

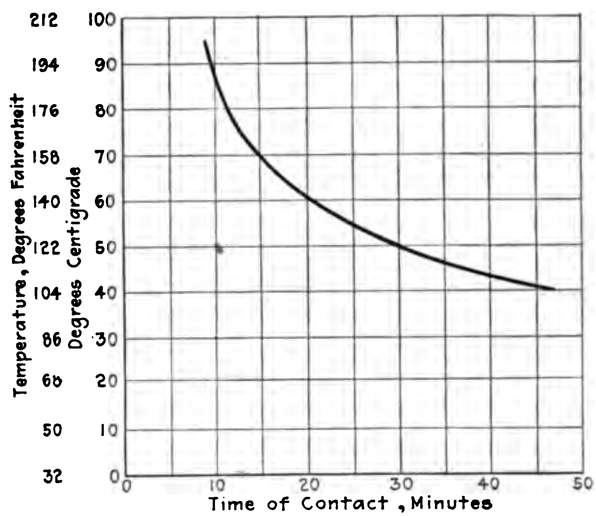

FIG. 9 RATE OF CORROSION

N. E. Funk, Engineer, gave 0.70 cc. per liter in feedwater leaving heater at 209 deg. fahr. with some corrosion in economizer tubes but much less than before when the heater was operated at 140 deg. fahr.; the feedwater after heating to 140 deg. fahr. carried 2.06 cc. per liter of free oxygen. We have shown, however, that it is possible to approach quite close to the theoretical minimum oxygen in an open heater by a larger area of baffles or by spraying the water.

The method of collection of sample is not described and may have a bearing on the heater variations the author refers to. In taking samples of hot water under pressure for oxygen determination, we draw the sample slowly while under pressure through a copper or tin coil, immersed in ice water so that when drawn into sample bottle the temperature of the water is about normal. This insures retention in solution of all gases and if the sample tube extends to the bottom of the bottle a sample can be collected in a minute or two without agitation and without reabsorption of air.

The amount of residual oxygen which will cause serious corrosion in a given time varies with the temperature at which the water is to be used. It has been demonstrated experimentally by the work done by Dr. W. H. Walker of the Massachusetts Institute of Technology and ourselves that corrosion of piping is closely proportional to the free oxygen in solution. Fig. 9 is drawn from experimental data indicating how the time required to use up the dissolved oxygen (and therefore the rate of corrosion) varies with the temperature. Several years experimental work on mechanical and chemical deaëration has been done at the Massachusetts Institute of Technology and National Tube Company Research Laboratories in coöperation, and several plants have been put into service to test out these principles in hot water supply systems for buildings, in central heating stations and power plants to develope the best type of apparatus for such service. This experience indicates that feedwater used with steel economizers should carry less than 0.20 cc. per liter. The above tests therefore indicate that open heaters as usually operated will not reduce the oxygen low enough to stop corrosion. The residual oxygen in feedwater after passing through a properly vented open heater may be reduced below the danger point by further mechanical deaëration such as by passing the water from heaters over baffles at about the boiling point in a separate chamber at lower pressure (no condenser required), or by injecting the superheated water into a low pressure chamber with a condenser in series as described by the author; or the residual oxygen passing through the heater may be quickly and completely removed by chemical treatment or a combination of both systems as is most convenient. We have had a process for removing oxygen by the latter principle in continuous use for nearly five years. This is based on passing the heated water under pressure through a large mass of expanded steel scrap, which I have termed "deactivating" the water in distinction to mechanical deaëration.[1] In this way the last trace of oxygen may be removed within a wide range of temperature and pressure. Fig. 10 shows two pieces of wrought iron pipe; the one on the right failed after two

[1] Bureau of Mines Technical Paper No. 236. Abatement of Corrosion; Proceedings A. S. H. & V. E. 1916–1917–1918–1920. Papers by the writer and associates.

years continuous service carrying water at 170 deg. fahr. under 100 lb. pressure, while the other piece was taken out after three years of similar service with the same water after deactivation, and so far shows no perceptible corrosion.

In Par. 10, the author calls attention to the very accelerated corrosion in the steel tank, copper pans and condenser of his experimental apparatus. We have frequently found when using *closed* heaters that about one-half of the oxygen is removed by contact with parts of the heater or connecting pipes which indicates how very effective is this means of removing dissolved oxygen. Mechanical means of removing free oxygen from water requires careful adjustment and control of temperature and pressure whereas chemical

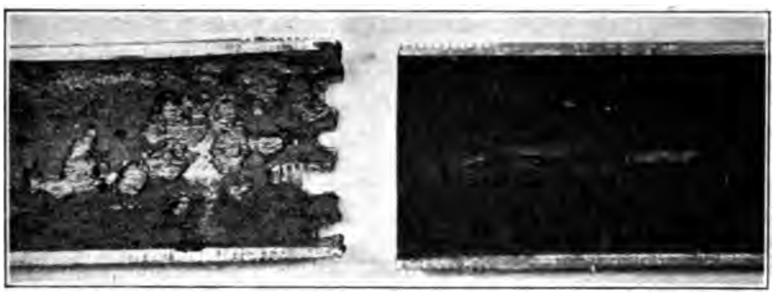

Fig. 10 Effect of Deactivated Water in Preventing Corrosion

means such as the use of a suitable kind of steel scrap, will "fix" the oxygen independent of such variations without further attention except for renewal of material at long intervals.

The author's paper shows an interesting means which may be used for mechanical deaëration of water but he is hardly justified from the data and experience given in claiming that this process involves a minimum of operating expense (see Par. 1). There seems to be no doubt that the controlling factor in fresh water corrosion is the presence of dissolved oxygen, carbonic acid being of minor importance. The most economical and convenient method for controlling corrosion in steel economizers will have to be developed by experience, but that control of corrosion can be accomplished by the application of these principles the writer believes has been established beyond doubt, therefore with attention to the relative temperature of waste gases and feedwater so as to avoid outside condensation on tubes there is no apparent reason why steel economizers cannot be used without any more deterioration than is now experienced in boilers.

Fig. 11 shows the proportionate quantitative relation between dissolved oxgyen and corrosion.

R. N. EHRHART took exception to the statements contained in the discussion by George H. Gibson. An open heater, he contended, was not sufficient to remove the harmful gases from the feedwater. He believed that it was the consensus of opinion of operating companies that some air extraction apparatus was absolutely essential.

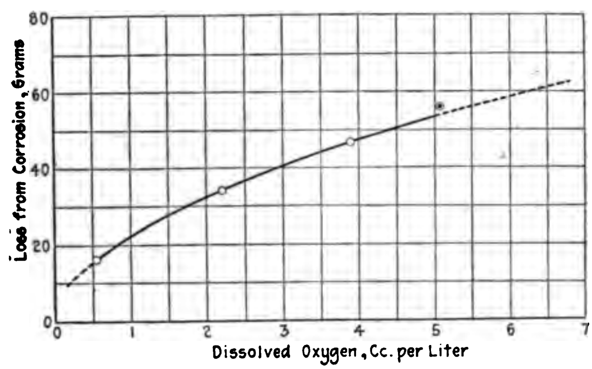

FIG. 11 RELATION BETWEEN DISSOLVED OXYGEN AND CORROSION

E. H. TENNEY said that the subject of corrosion in boilers and heaters was one of the most important with which power plants had to deal. In the central station of the Union Electric Light and Power Co. they had been forced to raise the temperature of the feedwater entering the economizer to 210 deg. in order to avoid excessive corrosion.

In the St. Louis district there was also a serious problem in the dissolved organic matter contained in boiler waters, and he wished to appeal to the Society to make this the subject of investigation.

C. HAROLD BERRY. The writer wishes to take exception to several statements of the author.

In Par. 2, the author asserts that Henry's Law "states that a solvent in contact with a gaseous mixture dissolves the component gases in proportion to the partial pressures which they exert," and he concludes that "if extraction is complete, the gas mixture extracted from water is roughly of the same composition as air from which solution was attained." These statements are quite at variance with previous literature and data. The Landolt-Börnstein tables, which the author accepts as a reliable authority in Par. 5, indicate that if water be saturated in contact with air (21 per cent oxygen, by volume), and if the dissolved gases be subsequently driven off and analyzed, they will show an oxygen content of between thirty and thirty-five per cent, depending upon temperature conditions. The author's erroneous conclusion follows from an incorrect statement of Henry's Law. This generalization states that at a given common temperature of gas and water, the solubility of a given gas in water is proportional to the partial pressure of that gas on the free surface of the liquid. Stating this in mathematical form [Solubility] = [factor] × [partial pressure]. But the factor of proportionality is a function of the temperature, and is, moreover, different for each gas considered. Thus we may write

$$[\text{Solubility}] = \begin{bmatrix} \text{function of} \\ \text{temperature,} \\ \text{different for} \\ \text{each gas} \end{bmatrix} \times [\text{partial pressure}].$$

In these expressions, solubility may be expressed in terms of weight or volume, but if gas volumes are used, they must be computed to standard conditions.

If the solubility proportionality factor assumed the same value for all gases at various temperatures, in other words, if the temperature functions were identical for all gases, the author's statements would be strictly true; but such is not at all the case, as will be obvious from even casual inspection of tables, such as the Landolt-Börnstein or Kaye and Laby tables. Some gases are much more soluble than others, and the process of solution and extraction is one in which the gas mixture is concentrated with respect to the more soluble constituents. It would be interesting to know by what means the author "demonstrated [that] the proportions of constituent gases [liberated from solution] were roughly the same as for atmospheric air." (See Par. 9.)

In Par. 6, the author states that "The objection to the universal use of this iodometric titration was the fact that it gave results only in terms of oxygen." It seems to me that this is not a valid objection, since oxygen is exactly what we wish to determine. Nitrogen is inert, as the author states in Par. 26. Indeed, bearing in mind the fact that the dissolved gases, after liberation, do not have the composition of air, but a composition varying somewhat with temperature conditions, it would seem that the boiling or displacement methods are distinctly inferior to the direct determination of oxygen.

In Par. 16, as well as elsewhere, there seems to be some confusion between entrained and dissolved gases. This is especially prominent in Par. 19. The test made under pressure, as there described, probably collected gas which had been liberated from solution, but was still entrained in the water. Just as it takes time for a precipitate to settle, so it takes time for gas liberated from solution to rise to the surface, and to test the water without allowing time for this to occur is hardly fair to the apparatus. In fact, the function of any gas-separating apparatus is simply to change gas from the dissolved to the entrained state, in which it will separate itself from the water by the spontaneous, though somewhat slow, process of flotation. All that is necessary to complete this process is a chamber from which air is removed as fast as it leaves the water surface.

In Par. 24 reference is made to what apparently was an injection of water at a temperature above 212 deg. fahr. and from a corresponding pressure, into the separator chamber, at atmospheric pressure. We read that this differs from injection from atmospheric pressure into a vacuum in that "the pressure gradient was of opposite direction." This seems to be contrary to the facts. The water must in all cases initially be at a pressure above that in the separator, else it would not be possible to elevate its temperature to a value above the boiling point corresponding to the separator pressure.

In Par. 48 it is noted that a thermometer well projecting into a pipe indicates a temperature apparently too high by several degrees. The author says "the only plausible explanation was that the air particles were carried along in the rarefied vapor stream with a translatory motion, but with a pressure so close to zero that all sensible molecular vibratory motion had ceased." This would mean that the air is at zero absolute temperature! It is difficult to see how the addition of inert molecules to a stream of vapor would give rise to an abnormally high temperature indication. This phenomenon of super-temperature has been observed frequently, and one satisfactory explanation has attributed this effect to impact of the stream (*not* of individual molecules) against the thermometer well.

With regard to terminology, the author uses several unusual expressions, which are open to criticism. He refers in several places to water being "in equilibrium with the temperature," when what he means is that water is "in equilibrium with the overlying atmosphere at the prevailing temperature." We also find reference to "the vapor temperature of the vacuum," whereas "boiling point" or "saturation temperature" are more usual terms for the temperature of air-free saturated vapor at the prevailing pressure. In Par. 22 we find mention of the "vacuum-temperature drop" and of the "separator temperature gradient." The meaning of these terms is somewhat vague to the writer. If what is meant is the fall in water temperature during evaporation, why devise obscure terms? Moreover, the use of the word "gradient" for a progressive change is at least questionable.

Entirely irrespective of the matters of detail referred to above, the author's paper is of great value and interest, in that it describes an apparatus which is apparently a very successful means of overcoming what has in many cases proven to be a very troublesome difficulty.

The Author has always defined the air separation process, when a definition was necessary, as "the rapid injection of heated water into a region of vacua and an explosive boiling of it at the expense of the heat of the liquid available to the vacuum, with a simultaneous recovery of the heat liberated to the vacuum by a condenser cooled by the incoming water preliminary to its heating." Any claim of novelty is based upon the combination of vacuum boiling and useful heat recovery rather than upon vacuum boiling alone.

The author is not at variance with Mr. Gibson in the air removal which may be credited to an open feedwater heater. In an extensive series of surveys made since this paper was published, an oxygen value of 0.28 cc. per liter was obtained from a atmospheric "thoroughfare" heater which received the entire steam output of the boilers through the exhaust of rolling mill engines. The steam venting which it was necessary to secure, however, is prohibitive in most classes of power service. The statements made in the paper were specified as applying to an induction heater operating at rating.

A discussion of the applicability of solubility tables invariably leads to controversy because three variables, temperature, solubility, and partial air pressure are intimately related and are in general, not accurately determinate. In the results given in this paper, the values of Winkler are arbitrarily taken since they represent theoretically and experimentally the closest approximation to the conditions at which saturation is attained at normal atmospheric temperatures. The use of percentage values calculated from the Winkler table automatically eliminates any confusion over the selection of solubility values at high temperatures. The conditions for the solubility value of Bohr and Bock, 100 deg. cent. and 760 mm. total pressure, have no application to any piece of engineering apparatus except possibly an air separator, which perhaps approaches them very closely, but cannot safely be said to equal them.

The author would have welcomed Mr. Gibson's explanation of the pseudo-adiabatic compression mentioned in Par. 48. It is a phenomenon which experimenters working with extremely rarefied water vapor and air mixtures frequently encounter.

The writer is grateful for the agreement which Mr. Speller has accorded the essential points in the paper. Mr. Speller's researches along an analogous field of endeavor are sufficient to make his opinion authoritative.

No. 1749

LOCOMOTIVE FEEDWATER HEATING

By Thomas C. McBride, Philadelphia, Pa.
Member of the Society

This paper describes a system of locomotive feedwater heating which makes use of the so-called open type of heater in which the exhaust steam comes into direct contact with the cold feedwater. The author describes the construction and operation of the heater and discusses the coal saving, water saving, and possibilities of increased capacity due to locomotive feedwater heating. These are discussed first from a theoretical standpoint and second, from the practical standpoint based upon the reports of actual tests. Tables and curves based on tests are presented and analyzed and the paper is concluded with a discussion of the question as to whether the exhaust steam taken by the feedwater heater would cause trouble with the draft. From actual operating conditions the author states that "there was a surplus of exhaust steam left available for the draft."

THE use of feedwater heaters on locomotives dates back to the beginning of the locomotive itself. There are records of the application of an "open" or "injection" type heater to a locomotive in England in 1827, and an English patent of 1828 describes a tubular heater for the same purpose. Ross Winans of Baltimore applied tubular heaters to two types of Baltimore & Ohio railroad locomotives in 1836. It is worthy of note that the desirability of the feedwater heater for the locomotive was recognized at these early dates, although it must be appreciated that the advantages possible at that time were proportionately greater than at the present time because of the low-steam pressure then in use.

2 There is much of interest in the history of locomotive feedwater heating in a paper, Feedwater Heaters and Their Development by J. Snowden Bell which was presented at the fiftieth annual meeting of the American Railway Master Mechanics Association, in June 1918. Every conceivable type of heater and all possible sources of heat, even the hot ashes, seem to have been proposed or tried, and in view of the amount of study that has been given to the subject it seems rather remarkable that locomotive feedwater heating is not now in more general use. Possibly the reason lies in the

Presented at the Spring Meeting, St. Louis, Mo., May, 1920 of The American Society of Mechanical Engineers.

fact that it is only in comparatively recent years that the essentials of heater performance and pump operation have been understood. Furthermore the small size of the locomotives formerly in general use limited the saving per locomotive, and the general adoption of the injector, because of its simplicity and convenience, rendered feedwater heating impossible, for injectors heat the water which they use to such an extent by live steam that there is little or no opportunity for further heating.

3 There are two possible sources in a locomotive from which otherwise waste heat may be obtained to heat the feedwater: (*a*) the waste gases in the smokebox or stack, (*b*) the exhaust steam. Stationary practice with economizers indicates that waste-gas heaters cannot be supplied with cold water because the accumulation of condensation and soot on them seriously interferes with their efficiency. It is, therefore, necessary to turn to the exhaust steam for the source of heat and to develop this type of heater first, leaving the waste-gas heater for later development as may be found either necessary or advisable.

4 The exhaust-steam heater may be either of the open or injection type in which the exhaust steam comes into direct contact with the cold feedwater so that the water condensed from this exhaust steam is added to and mixed with this feedwater; or, of the closed or surface type in which the heat from the condensation of the exhaust steam passes to the feedwater through thin sheets of metal, generally in the form of thin brass tubes. Practically all of the development work to date that has been at all successful has considered only the surface type. There are a great many heaters of this type in service in Europe, but there are objections to it for locomotive use. It is a complex and delicate structure; it also wastes the water condensed from the exhaust steam with its heat or raises complications as to saving it, and it requires enough exhaust steam to heat all of the feedwater. On the other hand, the open or injection type of heater necessarily recovers both the water condensed from the exhaust steam and its heat so that both are returned direct to the boiler with the water taken from the tender. Less water is therefore taken from the tender and less exhaust steam is required for the heating.

DESCRIPTION OF A NEW TYPE OF HEATER

5 With these thoughts in mind, the author has devised a system of locomotive feedwater heating which makes use of a heater

of the open type which is manufactured and marketed by the Worthington Pump and Machinery Corporation. The system follows marine practice in heaters of this type, and the pump and heater have been combined into a single unit which can be conveniently attached to the side of the locomotive boiler as is the practice with air-brake compressors. Fig. 1 shows a unit with a capacity of 60,000 lb. of feedwater per hour in service on a Mikado locomotive. Fig.

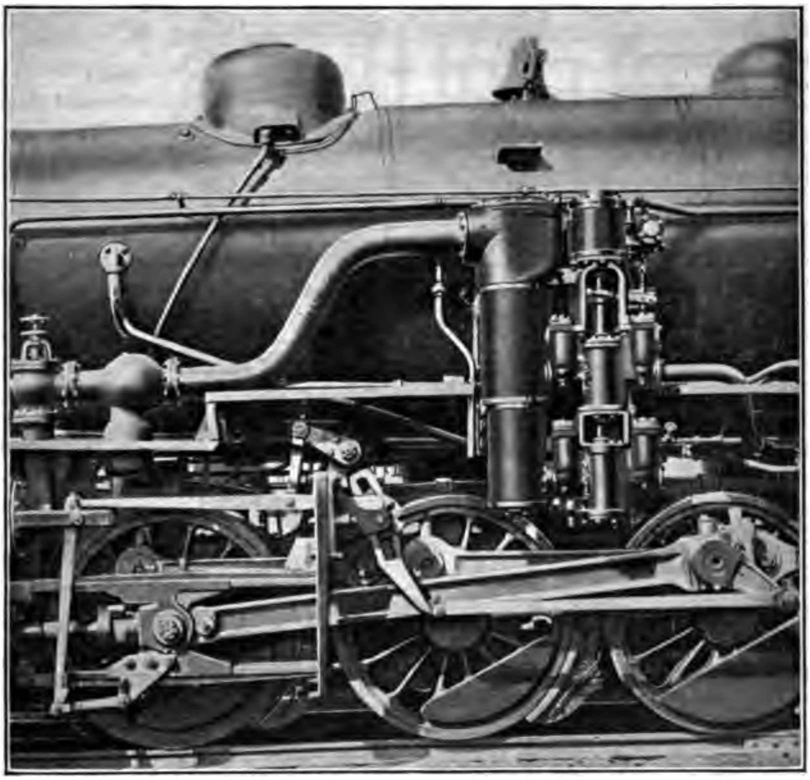

Fig. 1 Pump and Heater Applied to Mikado Locomotive

2 is a sectional view of the heater and pump. In these illustrations the heater is shown at the left, the feed pump at the right.

6 The heater is a simple cast-iron box, the upper part of which is supplied with exhaust steam taken from the locomotive through holes cut in the back of the cylinder saddle and a pipe which has in it a stop check valve and an oil separator, the latter having a small hole in it so that there is a continuous drip for the oil.

7 The pump is of the vertical type attached to the side of the heater by the various pipe connections. The steam cylinder of the pump is at the top, and it has two water cylinders clearly shown in the illustrations, both operated from the same piston rod. The water cylinder in the middle of the pump takes cold water from the tender and delivers it through a port in the side of the heater to a

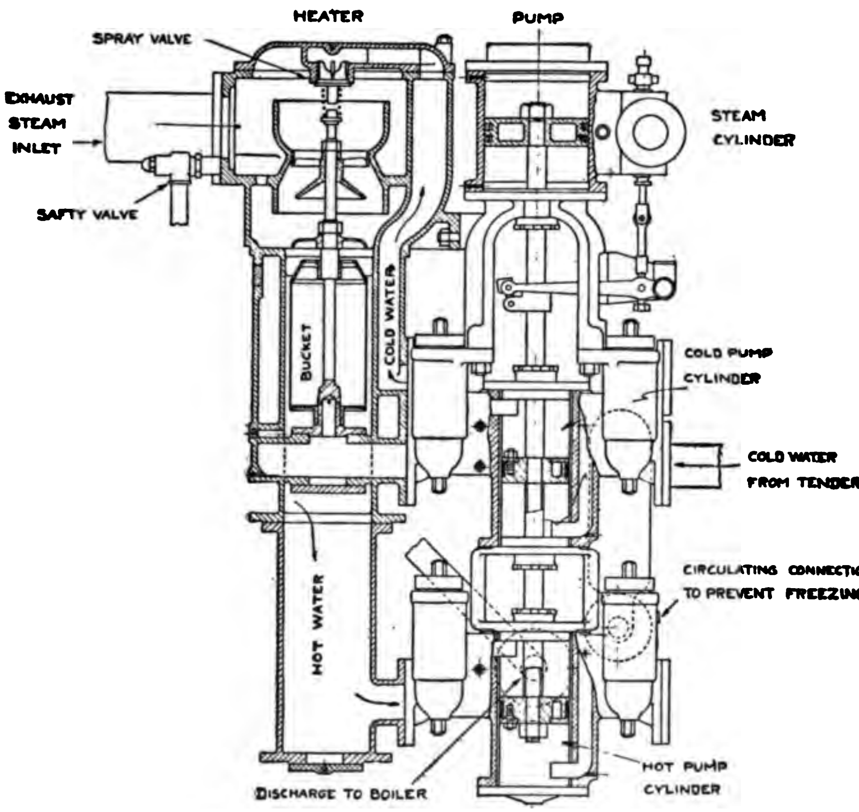

FIG. 2 SECTION OF THE HEATER

spray valve in the extreme top of the heater where it is sprayed into the exhaust steam that fills the top of the heater This condenses as much of the exhaust steam as the water needs to heat it, and the mixture, dropping to the bottom of the heater is taken by the other pump cylinder and delivered to the boiler through the usual check valve. The pump cylinder which takes the cold water from the tender and delivers it to the heater is really a measuring device which feeds into the heater approximately the same amount of water that the lower pump cylinder takes out of the heater, but the varying amounts of exhaust steam condensed in heating the feedwater cause a tendency for a slight excess of water to accumulate so that in time the heater would become flooded. This is prevented by returning this excess water under the control of a bucket in the heater to the pump cylinder in the middle of the pump through one of its pump-suction valves serving this water alone.

8 The pump is driven by saturated steam taken from the locomotive boiler through a 1-in. steam pipe and controlled by a throttle valve in the cab within convenient reach of the engineer. The exhaust steam from the pump is led into the back of the stop check valve where it mixes with the exhaust steam from the locomotive, and passes with it through the oil separator to the heater. These pipe connections as well as the cold-water pipe from the tender passing back of the air-brake compressor and the feed line from the lower pump cylinder to the boiler are shown in Fig. 1.

9 The adoption of the open heater for the locomotive may seem a radical step, but from this description it will be evident that no new elements are used. The heater itself is of a general type that has been in use in marine service for at least twenty years. The open heater with the oil separator to remove the oil from the exhaust steam before it enters the heater because of its merits has been almost universally adopted for stationary plants. One of these heater and pump units was thoroughly tested out on the locomotive test plant of the Pennsylvania Railroad at Altoona, Pa., in 1917, and has been in operation on road service since October 1918. Three more units have been in operation for approximately a year on pooled locomotives and have given no trouble other than the usual pump maintenance. Nothing has developed indicating that the open heater as a type is not well adapted to the locomotive and the coal and water savings have been so marked as to be evident to the crew.

10 The pump is much easier to operate than an injector as it requires only a slight turn of the 1-in. throttle valve to either start it or to make it run faster or slower as the boiler may need more or less water. It does not require priming and can be operated continuously at low capacities where the injector must be stopped and started periodically.

11 The feed pump should not be used, as is often done with the injector, to fill the boiler on the down grade and then be shut

off as long as possible on the up grade; first, because there is not as much inducement or necessity for operating it in this way and, secondly, because the prime purpose of the heater is to recover heat and it can only do so when the locomotive is pulling and exhaust steam is available for the heating. If the injector is operated continuously and maintains a constant water level in the boiler each pound of steam generated will require 1276 B.t.u. from the fire with 200 lb. steam pressure, 150 deg. superheat, and 40 deg. water in the tender. If the injector is shut off each pound of steam will require somewhat more than 925 B.t.u. so that the amount of steam from the same fire will be increased less than 38 per cent. It will later be shown that under the same conditions the feedwater heater would increase the amount of steam about 13.6 per cent. The feedwater heater will, therefore, materially increase the amount of steam on the up grade, although not to the extent possible in the case of the injector and should be depended upon to feed the boiler entirely on the up grade with the consequent greater amount of heat recovered from the exhaust and more nearly constant water level in the boiler.

ADVANTAGES OF LOCOMOTIVE FEEDWATER HEATING

12 It is not possible to make an exact general statement of the advantages to be obtained from locomotive feedwater heating, because much depends on the particular locomotive to which the heater is to be attached, the capacity at which it is worked, the temperature of the water in the tender, and the basis on which the advantages obtained are stated.

13 We are liable to base our notions as to the advantages of feedwater heating on the results obtained in stationary practice where the coal saving is very properly considered as equivalent to the reduction in the amount of heat necessary to evaporate the water because of the higher temperature at which the water is delivered to the boiler. In the case of the locomotive, however, with its very complicated relations of the different operations that are going on, and the very wide range of capacity through which the boiler is operated, there is generally a much greater coal saving than in stationary practice. Three factors enter into the final result. These are as follows:

1 The reduction in the amount of heat required to evaporate the water in the boiler, which might be called the *heat saving due to the feedwater heater.*

2 The reduction in the amount of heat required results in a greater reduction in the amount of coal required because of the better efficiency of the boiler at the lower rate of combustion. These two factors give the *coal saving based on water evaporated.*

3 The exhaust steam taken by the heater from the exhaust ports of the locomotive reduces slightly the back pressure in the locomotive and should show some advantage. All three factors combined will give the *coal saving based on indicated horsepower.*

14 An analysis of these three factors will show the coal and water saving referred to the steam that is used in the locomotive cylinders, and therefore, the minimum saving since it does not include the saving in generating the steam used by the auxiliaries. It is necessary to make this analysis partly theoretical, assuming certain constant operating conditions in order that a fair comparison may be made of the effect of the heater on different locomotives and at different locomotive capacities. It can only apply to the coal burned while running since the heater can have no influence while standing or at terminals. The percentage of coal saving due to the heater, and to a lesser extent the percentage of water saving when based on ton miles, the month, or the year will be less than the percentage while running by an amount determined by the use made of the locomotives.

HEAT SAVING DUE TO FEEDWATER HEATER

15 The first factor, the heat saving, is determined by the amount the feedwater is heated by exhaust steam taken from the locomotive. This factor will evidently be greatest when the tender-water temperature is low and when the exhaust steam pressure in the heater is highest, as this pressure determines the temperature to which the feedwater can be heated. This pressure will be highest when the locomotive is working hard, and when the exhaust steam for the heater is taken from that part of the steam valve chests or cylinder saddles where its pressure is greatest.

16 The exhaust steam from the feed pump need not be wasted, since it can be and is used in the feedwater heater, but this use prevents the recovery of the same amount of exhaust steam and heat from the exhaust of the locomotive. The amount of steam used by the feed pump is, therefore, a direct charge against the saving due

to feedwater heating as compared to injector operation when considered in the light of recovery of waste heat from the locomotive exhaust, or when estimated by the temperature rise of the feedwater in the heater.

17 Test plant results and actual daily operation on the road with superheated steam locomotives show that the feedwater heater described can be depended on to effect a reduction of at least 10 per cent in the summer time and 12 per cent in the winter time in the amount of heat required to evaporate the water and superheat the steam. This heat saving will not be constant at these figures

TABLE 1 HEAT SAVING DUE TO FEEDWATER HEATER

Assumed steam pressure in branch pipe, lb. per sq. in.	200	200	200	200
Assumed superheat, deg. fahr.	None	None	150	150
Assumed temperature water in tender, deg. fahr.	40	70	40	70
Heat content per pound of steam, B.t.u.	1199.1	1199.1	1284.6	1284.6
Heat content per pound of water, B.t.u.:				
at 40 deg.	8.0	......	8.0	......
at 70 deg.	......	38.0	......	38.0
Heat required to generate one pound of steam, B.t.u.	1191.1	1161.1	1276.6	1246.6
Heat saving in B.t.u. when water is heated from:				
40 to 215 deg.	175.0	......	175.0	......
70 to 215 deg.	......	145.0	......	145.0
Heat saving, per cent	14.7	12.5	13.7	11.6
Total heat required by locomotive with feed pump as compared to injector operation, per cent	102	102	101.75	101.75
Heat saving in locomotive with heater as compared to injector operation, per cent	15.0	12.75	13.94	11.80
Heat required, per cent	87.0	89.25	87.81	89.95
Heat saving as compared to injector operation, per cent	13.0	10.75	12.2	10.05

but will vary with the capacity at which the locomotive is worked because the steam pressure in the heater determines the temperature to which the feedwater can be heated. These figures can, however, be conservatively accepted as representing average working conditions at which the greater part of the coal is burned.

18 The tabulation given in Table 1 is based on heating the feedwater to 215 deg. This is quite conservative as it has been found easily possible to obtain this temperature with this heater even with tender-water temperatures of 40 deg. and on locomotives worked at less than two-thirds of their maximum steaming capacity. The tabulation shows that greater heat saving is possible with sat-

urated than with superheated steam locomotives; therefore the percentage of coal saving will be greater with saturated than with superheated steam locomotives. Table 2 gives the exhaust steam required for the heating. The percentage of water saved is shown by Table 3.

19 The allowances which have been made for the steam used by the feed pump are based on carefully conducted tests of the feed

TABLE 2 EXHAUST STEAM REQUIRED FOR HEATING

Assumed tender water temperature, deg. fahr.	40	70
B.t.u. required to heat one pound of feedwater to 215 deg.:		
from 40 deg.	175	......
from 70 deg.	......	145
Heat content of exhaust steam assumed at 7 lb. pressure. B.t.u. per lb.	1157	1157
Heat content of water at 215 deg. B.t.u. per lb.	183	183
Heat given up by exhaust steam in condensing from 7 lb. pressure to water at 215 deg. B.t.u. per lb.	974	974
Exhaust steam required to heat one pound of cold feedwater, per cent.	18	15
Exhaust steam to heater from feed pump, per cent.	2	2
Exhaust steam from exhaust ports of locomotive required by surface heater, per cent.	16	13
by open heater, per cent.	13.8	11.5

pump and heater which show that this feed pump when running with saturated and fairly wet steam required for its operation 2 per cent or less of the weight of steam generated by the boiler. The

TABLE 3 WATER SAVING DUE TO FEEDWATER HEATER

Steam for locomotive cylinders, per cent.	100	100
Steam required for feed pump, per cent.	2	2
Water required by locomotive with surface heater discharging water condensed from exhaust steam to track, as compared to injector operation, per cent.	102	102
Exhaust steam condensed in heating feed water, per cent:		
0.18×102	18.4	......
0.15×102	......	15.3
Water required by locomotive with open heater or surface heater saving condensation from exhaust steam compared to injector operation, per cent.	83.6	86.7
Water saving, per cent.	16.4	13.3

feed pump, therefore, uses 2 per cent or less of the heat required from the fire by the saturated steam locomotive, but in the case of superheated steam locomotive the feed pump will require but about 1.75 per cent of the heat from the fire of the locomotive because of the greater heat required to generate the superheated steam. The tabulation is based on steam leaving the branch pipe at 200 lb.

pressure with 150 deg. superheat in the superheated steam locomotive and gives value of the heat saving, water saving, and percentage of exhaust steam required by the heater for tender-water temperatures of 40 and 70 deg. respectively. The term "heat content" is used to express the amount of heat in the water or steam above that in water at 32 deg. fahr. temperature.

20 From the tabulation it will also be noted that in the case of the feedwater heater for the locomotive, because of its comparison to injector operation, it is not possible to apply the usual short rule customary in stationary practice stating that a certain number of degrees of heating of the feedwater results in 1 per cent heat saving, but rather that it is more exact to state, assuming the conditions of the table, that in the saturated steam locomotive, after the first 23.8 or 23.2 deg. heating, each 11.9 or 11.6 deg. of heating represents 1 per cent heat saving, and that in the superheated steam locomotive after the first 22.3 or 21.8 deg. of heating each 12.8 or 12.5 deg. of heating represents 1 per cent heat saving, the first figures in each case applying with 40 deg. and the second figures with 70 deg. water temperature.

21 European practice, as far as the author is informed, involves the use of tubular heaters differing only in details of their construction and discharging to the track the water condensed from the exhaust steam in the heater, thus losing both this water and its heat, necessarily with some additional steam that must be blown through the heater to prevent it becoming air-bound. These locomotives require 2 per cent more water than when operated with the injector because of the steam used by the feed pump and discharge to the track from 15 to 18 per cent of this with from 2.6 to 1.8 per cent of the heat of the saturated and 2.4 to 1.6 per cent of the heat of the superheated steam locomotive in the water alone for 40 and 70 deg. water respectively. There are a few surface heaters in operation on locomotives in this country in which this water is saved by pumping it back to the tender through filters to remove the oil, but in doing so approximately 80 per cent of its heat is lost by radiation from the tender. This loss of heat from the locomotive with the surface heater does not count directly against the coal saving since it is supplied by the exhaust steam, but it does count indirectly since it decreases the amount of exhaust steam left available for the draft. The open heater is more economical in the use of exhaust steam and leaves more available for the draft with the possibility of lower back pressure and saving because of this.

22 The locomotive with the open heater or the surface heater saving the water condensed from the exhaust steam requires the extra 2 per cent steam to operate the feed pump, but from 15 to 18 per cent of this 102 per cent is recovered, so that the net water saving is from 13.3 to 16.4 per cent respectively for the assumed summer and winter conditions as compared to injector operation.

COAL SAVING BASED ON WATER EVAPORATED

23 To combine these results with the second factor in order to obtain the coal saving based on water evaporated it is necessary to consider particular locomotives since the second factor involves the change in the efficiency of the boiler with varying boiler capacity. For this purpose the curves of Fig. 3 have been prepared. The curves on the left marked A are on the basis of 10 per cent heat saving; those on the right marked B on the basis of 12 per cent heat saving. The upper solid curves marked "Injector" are test plant results of a Consolidation, Atlantic, and Mikado type locomotive respectively and show the total amount of water evaporated per hour against dry coal fired. The dotted curves marked "Heater" for each of these three locomotives show the amount of water that would have been evaporated by the same amount of coal with a feedwater heater reducing by 10 per cent and 12 per cent respectively the amount of heat required to evaporate the water and superheat the steam. The vertical distances between the solid and dotted curves represent the amount of coal saved and these quantities are shown in the lower set of curves. This lower set of curves, therefore, shows the coal saving based on water evaporated due to the feedwater heater that follows a reduction of 10 per cent and 12 per cent respectively in the amount of heat necessary to evaporate the water and superheat the steam. These curves show that the feedwater heater attached to a locomotive will effect a coal saving beginning at low capacities practically identical with the heat saving, or the coal saving which we have been accustomed to in stationary practice, but which increases at first gradually and then more rapidly as the capacity of the locomotive is increased, reaching values twice or more than twice these figures as the maximum capacity of the locomotive is approached.

24 It would be difficult to credit this result were it not warranted theoretically by the second factor mentioned above, and had it not been confirmed in test plant results with a wide margin to spare.

A study of the upper curves in connection with the lower curves of both figures will show that the very great coal saving possible is accompanied by an increase in steaming capacity due to the feed-water heater and that it should be possible to work the locomotive with the heater at greater steaming capacities than are possible with the injector but how much greater would depend largely on the particular locomotive.

25 Referring to the lower set of curves of Fig. 3 and assuming that the Consolidation locomotive would be operated normally at a steaming rate of say 25,000 lb. per hr., it is noted that the 10 and

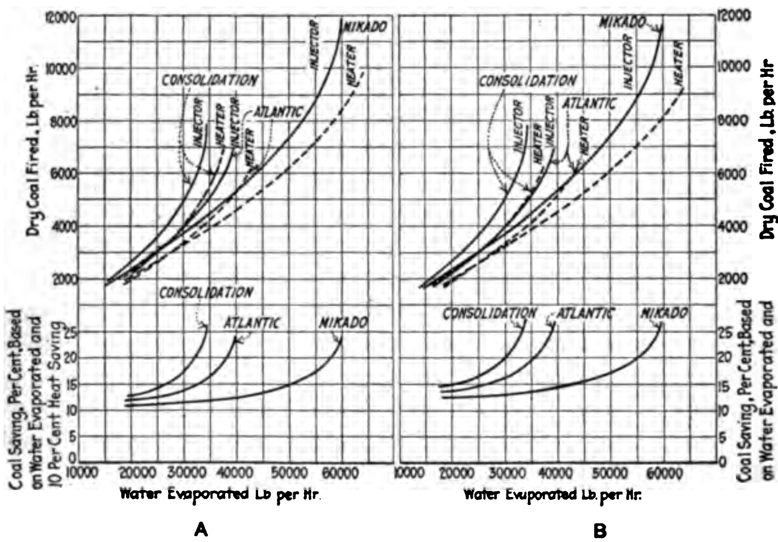

FIG. 3. CURVES SHOWING THEORETICAL COAL SAVING BASED ON WATER EVAPORATED

12 per cent heat savings assumed as summer and winter conditions would result in 14 and 16.6 per cent coal savings respectively. Assuming that the Atlantic type locomotive would also normally be operated at a steaming rate of 25,000 lb. per hr., the coal saving would be 12.8 and 14.5 per cent respectively. Assuming that the Mikado locomotive is hand fired and operated at a steaming rate of 37,500 lb. per hr., the coal saving would be 12 and 14.2 per cent respectively.

26 On the other hand, it is altogether possible that advantage should be taken of the feedwater heater either in increased speed or increased hauling capacity and the same amount of coal burned

with it as would have been burned if the locomotive had been operated with the injector. In this case, the summertime 10 per cent heat saving would result in a $\frac{100}{100-10}$ per cent steaming capacity or 11.1 per cent increase; and the 12 per cent heat saving would result in a $\frac{100}{100-12}$ per cent steaming capacity or an increase of 13.6 per cent. But as compared to injector operation at this increased steaming capacity the coal saving in the Consolidation locomotive would be 15.5 and 19 per cent, in the Atlantic locomotive 13.3 and 15.5 per cent, and in the Mikado locomotive 13 and 15.3 per cent respectively.

27 The feedwater heater does not appear to as great advantage in the Mikado locomotive as in the other type locomotives because it is not as fully loaded. If the Mikado locomotive was operated at a steaming rate of say 50,000 lb. per hr. the assumed 10 and 12 per cent heat saving would have resulted in a 15 and 17.3 per cent coal-saving respectively, and if advantage had been taken of the feedwater heater for the increased capacity burning the same amount of coal as with the injector the coal saving would have been 18 and 21.2 per cent respectively.

COAL SAVING BASED ON INDICATED HORSEPOWER

28 Any modification which the third factor makes in the combination of the first two factors as shown in Fig. 3, leading up to the coal saving per indicated horsepower, must be but slight and depend on the particular locomotive. Saturated steam locomotives should show a further gain of a few per cent from reduced back pressure. Test plant records show some reduction in this back pressure in superheated steam locomotives accompanied by a reduction in superheat and indicate that, at least until further tests are made, this factor should not be considered for superheated steam locomotives.

29 The curves of Fig. 4 are test plant results of a Mikado locomotive, comparing its operation with the injector and with the feedwater heater described in this paper. Table 4 shows the heat recovered by the feedwater heater and Table 5 the coal saving and increase in efficiency. They show a coal saving curve of the same general character, but slightly higher than the curve for the Mikado locomotive of Fig. 3, based on water evaporated, although the tests

were made under practically identical operating conditions as to temperatures and pressures.

30 The coal-saving curve of Fig. 4 has been reproduced in Fig. 5, and a curve added showing the reduction in the amount of heat required to evaporate the water and superheat the steam, because of the heater. The cross-hatched area below this curve shows the saving in heat due to the heater. The cross-hatched area above this curve shows the additional saving presumably all due to the better efficiency of the boiler with the reduced amount of heat required from the fire because of the heat recovered by the heater.

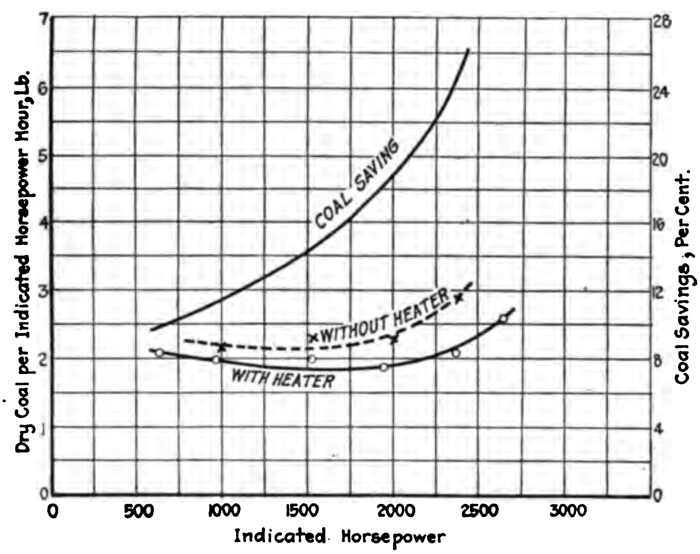

FIG. 4 CURVES SHOWING COAL SAVING BASED ON INDICATED HORSEPOWER

EXHAUST STEAM AVAILABLE FOR THE DRAFT

31 The question has been raised as to whether the exhaust steam taken from the locomotive by the feedwater heater would cause trouble with the draft. This question could be analyzed from the partly theoretical curves of Fig. 3, but as actual operating conditions are of more interest, curves have been added to Fig. 5 showing the total exhaust steam taken by the heater and the exhaust steam taken from the exhaust ports of the locomotive. The distance between these two curves represents the exhaust steam furnished by the feed pump to the heater. It is noted that the curve of ex-

haust steam taken from the exhaust port is below the curve of coal saving. The feedwater heater, therefore, reduced the amount of coal burned at a greater rate than it reduced the amount of exhaust steam, so that there was a surplus of exhaust steam left available

TABLE 4 HEAT RECOVERED BY THE FEEDWATER HEATER

Test No.	1	2	3	4
Steam to engine, lb. per min.	314.5	492.8	599.1	757.3
Steam to feed pump, lb. per min.	7.4	10.5	12.5	14.6
Steam to safety valve, lb. per min.	0.8	0.7	2.1	5.5
Exhaust steam condensed, lb. per min.	42.3	71.4	90.0	113.7
Heat in steam, B.t.u. per min. to engines	402,749	639,063	780,867	996,531
to feed pump	8,876	12,594	14,994	17,393
to safety valve	960	840	2,519	6,597
Total heat in steam	412,585	652,497	798,380	1,020,521
Heat recovered from exhaust, B.t.u. per min.	50,725	85,694	108,063	136,736
Per cent of total heat	12.3	13.1	13.5	13.4

for the draft no necessity for reducing the size of the blast nozzle and consequent possibility of reduced back pressure in the locomotive cylinders because of this heater. For instance, at 2000

TABLE 5 COAL SAVING AND INCREASE IN EFFICIENCY WITH FEEDWATER HEATER

Test No.	1 and 5	2 and 6	3 and 7	4 and 8
Speed in m.p.h.	14.6	14.6	22.0	22.0
Indicated horsepower:				
Without heater	990	1549	2001	2388
With heater	965	1534	1949	2373
Coal per indicated horsepower hour, lb.:				
Without heater	2.2	2.3	2.3	2.9
With heater	2.0	2.0	1.9	2.2
Coal saving by feed heating, per cent	9.1	13.1	17.4	24.7
Drawbar horsepower:				
Without heater	824	1289	1690	2019
With heater	783	1293	1679	2055
Thermal efficiency of locomotive, per cent:				
Without heater	7.0	6.7	6.7	5.4
With heater	7.6	8.0	8.6	7.8
Increase in efficiency with heater, per cent	8.5	19.4	28.3	44.4

i.hp., the reduction in the amount of coal burned was 18 per cent, and in the amount of exhaust steam, 12½ per cent. As compared to injector operation, 82 parts of coal had to be burned and 87½ parts of exhaust steam were left available for the draft to burn it.

CONCLUSIONS

32 The feedwater heater on the locomotive may be used to reduce the coal and water consumption working the locomotive at the same capacity as with the injector or to increase the capacity of the locomotive, or partly for both, as occasion demands. It will save coal and water all the time, and permit of increased capacity when needed.

33 The feedwater heater might be considered as alternate

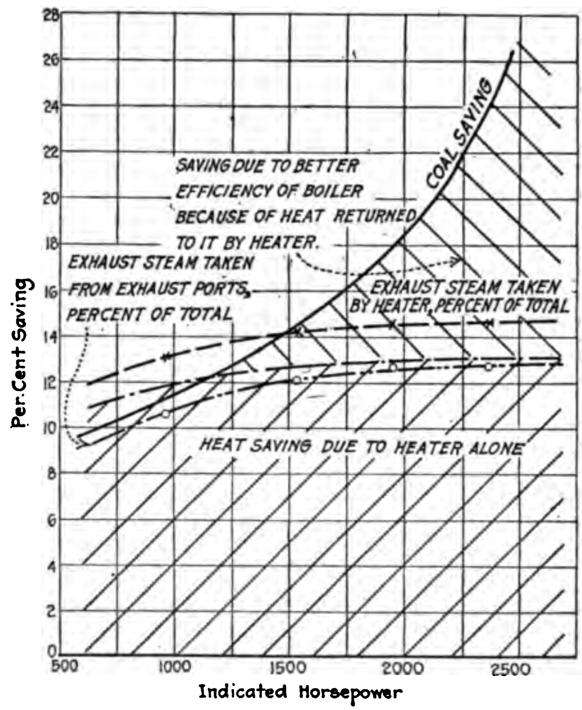

FIG. 5 ANALYSIS OF COAL SAVING BASED ON INDICATED HORSEPOWER WITH EXHAUST STEAM LEFT AVAILABLE FOR DRAFT

to large size of locomotive boiler and grate as a means to coal economy, if for any reason both are not considered advisable or possible for still further economy. This feature is well illustrated in Fig. 3, where it is noted that the Consolidation locomotive with an open heater gives more steam per pound of coal burned than the Mikado locomotive without a heater up to 2800 lb. of dry coal fired per hr., with the 10 per cent heat saving, and up to 3000 lb. per hr. with the

12 per cent heat saving. Test plant records show the Consolidation with heater burns less coal per indicated horsepower than the Mikado without heater up to a still greater rate of coal fired. The smaller amount of coal burned by the Consolidation locomotive at terminals and its lower first cost and maintenance cost favor it still further in the comparison.

34 Viewed from the standpoint of coal and water saving, the adoption of the heater becomes a question of interest on first cost, depreciation and maintenance cost against these savings. Viewed from the standpoint of increased capacity, the question becomes one as to whether the feedwater heater does not offer increased capacity at the lowest first cost, especially if, as in the case of the comparison of the Consolidation locomotive with and the Mikado locomotive without a heater, the adoption of the heater permits of the use of a simpler and less costly type of locomotive.

35 The feedwater heater offers a solution of the pressing problem of these times, when so many fairly old and small locomotives have outgrown their usefulness because they are not large enough for present-day demands. The feedwater heater will not only increase the capacity of these locomotives, or what might be called their economical operating capacity, at a comparatively low cost and with little change in them, but it will also show a larger percentage of coal saving because of their full loading than would be obtained with present-day larger locomotives.

36 The feedwater heater offers the peculiar advantage of being of most assistance to the locomotive just at the time when the locomotive needs assistance. It offers its greatest saving in fuel or increase in capacity in the winter time, and these advantages are not effected by reduced steam pressure or superheat, but continue in spite of them and at a proportionately greater rate.

37 The study of the subject of the feedwater heater for the locomotive that has been reviewed in this paper has resulted in the adoption of the open-feedwater heater as a type. It has been developed into a unit with the pump, so that it can be attached to the locomotive at minimum expense. These units have been thoroughly tested both on the test plant and on the road, and every confidence is felt in their ability to meet the severe requirements of locomotive service, and to effect the maximum possible saving of fuel and water with a minimum amount of attention from the crew.

DISCUSSION

G. M. Basford. The author has presented some very interesting and valuable data in a very clear manner. He has covered some phases of the subject which have not been well understood and gives some new data that will be appreciated by those who wish to investigate this subject which is now one of the most important before the mechanical officers of the railroads.

It is unfortunate, however, that this paper is confined so completely to the consideration of but one type of feedwater heater and that the one which has been in use on but few locomotives. This leads the author to present his subject in a somewhat less favorable light in some particulars than he might have done if the results that have been and are now regularly being obtained with the closed type of heaters on a number of railroads in this country and many in Europe were included.

Grafton Greenough wrote: "The more I have to do with locomotives the more impressed I am with the truth embodied in the old time saying that 'necessity is the mother of invention,' and it may be added that any invention which develops before it is needed is apt to lie dormant until the necessity for it arises. That this applies to the heating of feedwater for locomotive boilers is evidenced by the author's researches, showing that the conception of the feedwater heater is almost as old as the locomotive itself, and whilst the feedwater heater has lain dormant for over three-quarters of a century, we have every evidence that the necessity for its reappearance is upon us.

"This necessity is two-fold — in the first place the sizes of our most powerful locomotives are seriously restricted by physical conditions and no device which will enable us to obtain greater horsepower without increasing the size of the locomotive boiler can be ignored. In the second place the need of conserving labor and the transportation facilities required in producing and delivering coal, in addition to the saving in money to be effected, combine to create the necessity for saving fuel.

"The locomotive feedwater heater is a practical agency whereby the relative size of the locomotive may be reduced and a saving in coal be effected. It has proved its practicability and I predict that before long it will be regarded as an established device in common with superheating and compounding needful to bring about greater efficiency and economy in the locomotive engine.

F. N. SPELLER.[1] The author does not give any data on the amount of gas separation in the locomotive heater he describes which will of course depend on the temperature to which the water is heated and the design of the heater. Such information would be interesting as an indication as to whether or not there would be less corrosion in the boiler by the use of an open heater of this type. A well-designed open heater of adequate size has a marked effect in retarding corrosion in stationary boilers. One serious objection to closed heaters is the very rapid deterioration of the heater itself through the action of corrosive gases in solution.

The writer knew of feedwater heaters which had been tried on locomotives and which failed because of the very rapid deterioration of the tubes of the heater and this may be one of the reasons for the delay in the development of the feedwater heater for locomotives. In the ordinary closed heater, about half of the dissolved oxygen is removed in passing through the heater at the expense of its metal parts, and it is not surprising that the upkeep is such a serious item.

THE AUTHOR. It is very true, as has been stated by Mr. Basford, that the paper does not present the subject of locomotive feedwater heating in its most favorable light. The paper was not intended for this purpose, but rather to present what is believed to be the only tests ever made on a locomotive test plant of an open feedwater heater, first establishing the results to be expected from theoretical considerations, and finally analyzing the test plant results. This was an experimental heater which was rebuilt and is now giving still better results on the road, as are also heaters built later but not tested on the locomotive test plant.

It is also true that better results are being obtained with closed heaters, but it has been pointed out in the paper that the results, especially as to water temperatures, are controlled more by the particular locomotive than by the heater. The pressure of the exhaust steam from the locomotive in the heater determines the water temperature and this pressure in turn is determined by the pressure in the exhaust ports of the locomotive and the point in the exhaust ports from which the exhaust steam is taken. Tested

[1] Metallurgical Engineer, National Tube Co., McKeesport, Pa.

on the same locomotive on the test plant under exactly similar conditions, a number of different types of closed heaters gave less favorable results than were obtained from the open heater described. At the time the paper was written there were four of these open heaters in continuous service it this country; more than any other heater.

Answering Mr. Speller's question, there is a considerable volume of air discharged from the air vent of the open feedwater heater described, but no attempts were made to measure the amount. At low locomotive capacities with low pressures of exhaust steam in the heater, the air discharged was cold; at the highest capacities the temperature of the mixture of air and steam was as high as 180 deg. There must have been a considerable volume of the oxygen removed from the feedwater and discharged through the air vent so that it did not reach the boiler, but no studies of this feature of the heater were made. After almost two years of service no corrosion of the internal parts of the heater were evident, possibly because most of these parts are simply unfinished iron castings.

No. 1750

TIGHT-FITTING THREADS FOR BOLTS AND NUTS

By Chester B. Lord, St. Louis, Mo.
Member of the Society

Although in general the thread forms now in use are quite satisfactory, perfection has by no means been reached. There is still considerable search for what the author of the following paper terms "a thread that will not loosen," and in the preliminary portion of his paper he discusses the fundamental principles involved in the manufacture of threads. The problem to be solved, he states, is as follows: "Without sacrifice of strength, without increase of rejection, without additional manufacturing costs, find a method whereby a male and female thread of the same lead and pitch diameter may be made after repeated loosenings to fit right without the aid of a locking device." The reasons for departing from accepted practice are presented and discussed, and as a result of experimental work, the author draws the following conclusions: (1) The cause of stripped threads is lack of room into which the metal can flow; (2) The pitch diameter should be the same in both threads; (3) The lead should be the same; (4) The thread angle should differ by not more than 10 deg.; (5) The limits for the inside diameter of nut need not be adhered to closely, as the inner part of the nut thread holds very little, if any; (6) The outside diameter of plug and pitch diameter of both plug and nut are important and should be adhered to fairly closely.

WHAT is the cause of our periodic dissatisfaction with threads when in general they are so satisfactory? What other machine element is so easily made or is so satisfactory as regards strength? Why are there so many different kinds of threads when all are equally satisfactory, or rather unsatisfactory? Furthermore, is the dissatisfaction founded on performance or merely upon theory? Also, is the form or angle of thread a matter of importance, or merely an excuse for mathematical gymnastics? The only answer the writer has been able to elicit in reply to these various questions is that we are looking for a better thread; which statement, however, is rather indefinite and usually simply means a thread that will pass

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the gage. Of course, the real object of the search has been to find a thread that will not loosen.

2 In the past we have attributed our troubles to the fact that our fits were not close enough — the engineer's alibi for a poor design. But fundamentals cannot be violated in mechanics any more than elsewhere in nature, and we are attempting to violate two by insisting upon our present methods of inspection: (*a*) that interchangeable manufacture is a matter of percentage which depends upon tolerance and cost; and (*b*) that a force fit is not possible between two parts the surfaces of which are complements one of the other.

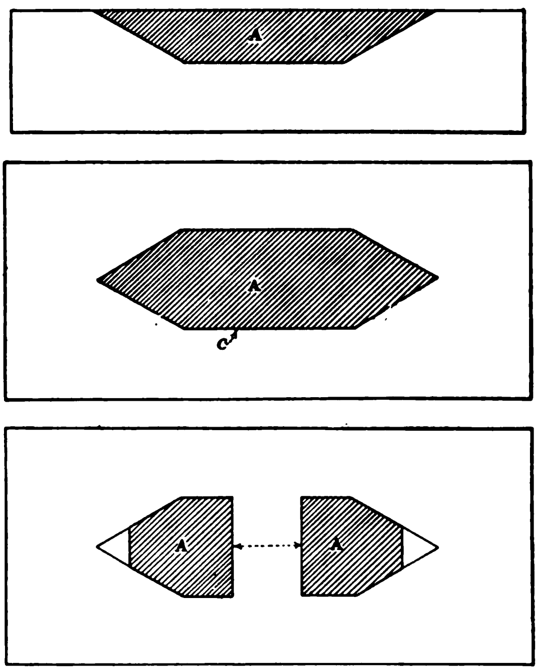
3 Having in mind the first fundamental, it is obvious that the chances of securing a perfect fit are limited by the cost, and the second fundamental would seem to render this entirely hopeless. It is therefore proper to conclude that a good fit is usually due to error, and that if changing both male and female threads produces no relative change, changing one thread must of necessity do so. It is the object of this paper to demonstrate that by making this latter change threads can be produced that are interchangeable practically regardless of tolerance, that will not loosen, and are cheaper to manufacture.

4 A physician always diagnoses a case before prescribing, so let us do likewise. The loosening of a thread fit is caused by vibration or repeated shock, the chief result of which is to flatten and burnish the parts of the thread that are in contact. This produces a slight looseness and the nut tends to follow the thread incline until it again fits. The same performance is repeated until finally the nut reaches an obstacle too large to flatten, or else the bolt and nut vibrate in unison and there is no further relative movement. This same phenomenon occurs in the case of a bolt screwed in a tapped hole.

5 Experiments with spring lock washers having a projection that will not allow the nut to turn without cutting the metal, and the same type of washer made without holding projections, show very little difference in their holding power; and this would seem to indicate that their function is to prevent vibration. But we also find nuts without lock washers in places where there is vibration, and still they do not loosen. The answer is that they are individually fitted.

6 The writer can remember in his shop days picking out and numbering nuts and bolts, and every mechanic knows that he can-

not take the bolts out of a cylinder head on an engine of good make and put them back again indiscriminately. They must go back into the holes from whence they came. All have demonstrated to their satisfaction the fact that a tight thread will not loosen by vibration, and that one with tolerance will, unless it is prevented from loosening by a lock washer or its action limited by some type of nut lock. We cannot lessen our tolerance because the tap wears



FIGS. 1–3 APPLICATION OF PRESSURE IN THREAD FITS

small and the die wears large, and the lesser cannot contain the greater.

THE PROBLEM TO BE SOLVED

7 To gain a better conception of the problem to be solved, consider a board cut as shown in Fig. 1. We cannot obtain a forced fit between the two pieces because one surface is the complement of the other. If we apply a force to A, we are no better off because the pieces cease to fit the instant the pressure is released. We may therefore state as a rule that where two surfaces are complementary to one another a tight fit cannot result without some means of main-

taining pressure. But if we cut the board as shown in Fig. 2, theoretically removing no material, we can replace A without force, and any pressure exerted will not make a tight fit unless we distort A or drive a wedge at C. This is analogous to a perfect thread, and driving the wedge at C is equivalent to introducing a slight difference in lead. If, now, we cut off the ends of A as shown in Fig. 3, and apply a force in the direction shown, we can obtain a forced fit because only the angles are complements and because we have a method of maintaining pressure. We can also even distort A because we have room for it to expand. We have only then to provide for three things: a method of making the parts in contact absolutely complementary; the introduction of sufficient metal; and a method of maintaining pressure. With our present type of thread we can only meet one of these — namely, that of partly maintaining pressure. This is demonstrated in Fig. 4, which shows in an exaggerated manner the effect of off lead.

8 To secure a good fit it is necessary to exert pressure on all flanks of the thread, and not on only two as we do with a standard thread. Practice demonstrates that the nut and bolt must be held by internal pressure on all flanks, and that the force maintained by screwing the face of the nut against another surface does not prevent loosening; but the writer again affirms that for practical purposes an off-lead thread is better than a perfect one, because in our present practice it is the lead or burrs that usually determine the fit.

9 While two slightly varying leads make a better fit, both as regards gaging and in actual use, this practice is not to be commended. Using a different lead to secure a fit is doing imperfectly on one side of the thread what the different-angle method does perfectly on both sides, because by having the leads identical and the thread supported on both sides, we secure a uniform finish instead of a distortion. Where the leads are different, the amount of distortion necessary to secure a fit increases with each thread. Thus, if the lead of a 20-thread stud is 0.05 in. and we make it 0.052 in., it will be 0.002 in. off center on the second thread and 0.018 in. on the tenth thread. This is entirely possible, and superior to a so-called perfect thread as regards fit, but a distortion, unmechanical, and unnecessary.

10 This is what is done with railroad fish-plate bolts where the specifications state the minimum foot-pounds at which the nut and bolt may be assembled. The impossibility of meeting these

requirements in quantity production is recognized by purchasing agents and most engineers, and so the lead is slightly changed. This, however, is merely a subterfuge and really defeats the purpose of the specifications in that it permits of a poorer and weaker thread than would otherwise be possible.

11 We are thus confronted with this problem: Without sacrifice of strength, without increase of rejection, without additional

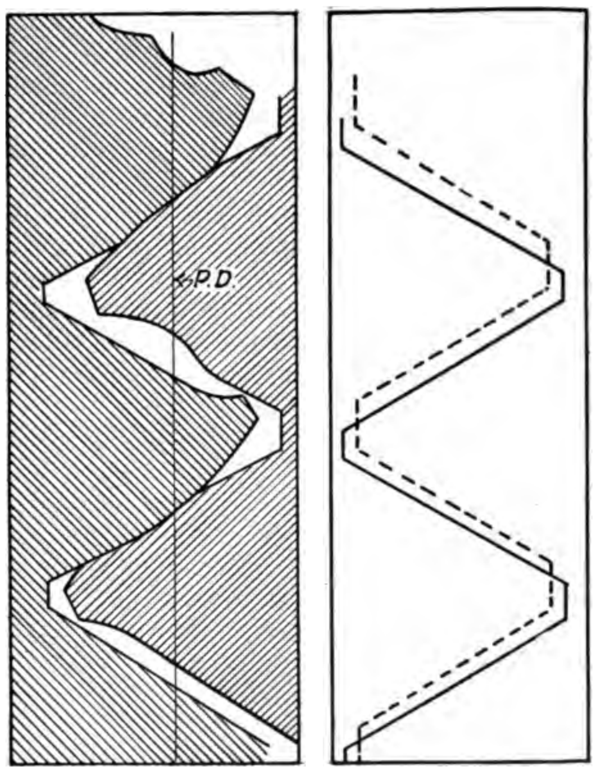


FIG. 4 THE EFFECT OF LEAD (GREATLY EXAGGERATED)

manufacturing costs, find a method whereby a male and female thread of the same lead and pitch diameter may be made after repeated loosenings to fit tight without the aid of a locking device. This, according to specifications, calls for a full thread at contact points, pressure applied continuously on all flanks, and maximum strength at the pitch line. This means the addition of surplus metal to the male (which is the only one affected) sufficient to fill out the female threads, which would be an impossibility were it not for the ductility and elasticity of steel. If we add this surplus metal we will find that we can more than fulfill the required conditions by changing the angle of the male thread to a lesser one than that of the female, having the two intersect on the pitch line.

12 Theoretically, this means a smaller amount of material below the pitch line of the male, compensated for by an increased amount above the pitch line. Actually it means *more*, because we do not have to recognize as great a tolerance below the pitch line because of our lesser angle, nor above the pitch line because we can take care of a reasonable surplus and the question of initial fit need not be considered. The space below the pitch line in the male, due to difference of angle, is not as great as the tolerance usually allowed.

13 Every nut is bell-mouthed, and every bolt, whether cut or rolled, is tapered. This is fundamental and cannot be avoided. This being so, we have no trouble in entering, for one or two threads, a bolt of larger angular but less total displacement than the female. It is also true in practice that the first threads of either nut or bolt are weaker than the rest, because they lack support on one side and hence flow easily.

14 When a male thread of this type and a standard mate are screwed together, we will have transformed a male of lesser angle to one of larger angle or of wider base than the standard U. S. thread, because we have filled out the female thread tolerance as well as that of the male. We will also have uniform pressure on all flanks; the maximum possible material at the pitch line; a hard, smooth surface analogous to a case-hardened one; a fit that will remain snug despite repeated removals and that may be screwed together by ordinary means; and yet we still retain an interchangeable bolt and nut according to U. S. standards. The nut has not been changed or distorted in any way, but has simply served as a finishing roll.

15 The writer has stated in a previous article that, aside from threads, nowhere else in machine work do we expect micrometer limits on a roughing cut; and the question naturally arises whether an operation similar to that described would not be an effective finishing and sizing operation for commercial work. This would be the equivalent of *making* them fit the gage, and would greatly reduce the cost due to rejections. We would, of course, still have variable nuts and the necessity of sizing them.

REASONS FOR DEPARTING FROM ACCEPTED PRACTICE

16 Let us now see what authority and precedent we have for making so radical a departure from accepted practice. To do so, let us consider how threads are made, especially by rolling. Generally speaking, thread rolling is circular knurling, knurling being that process whereby the diameter of a part is increased at certain points by a corresponding reduction of diameter at other points, due to pressure alone. It is applicable to both flat and round surfaces, but for the purpose of this article we will consider only round surfaces.

17 In turning screws and bolts to size the diameter is held, generally speaking, to the pitch diameter. The displacement of metal from the root and lower flank forms the addendum under the process. Cutting a thread on a bolt with a die is a somewhat analogous operation, the similarity increasing as the die becomes duller. For a die-cut 1-in. bolt it will generally be found that with a diameter of 0.990 in. a fuller thread may be secured than with one of 1.000 in., the explanation being that with the die we secure a combined cutting and rolling operation. Due to lack of clearance, if the diameter is too large, part of the metal is pressed into the bottom of the die and with such force that it drags and is torn off, thus leaving a thread of smaller outside diameter than would have been otherwise secured. This same phenomenon occurs when a nut is too tight.

18 If we require a holding fit on a shaft, do we use tolerances that allow of the shaft being several thousandths smaller than the hole it is to fit? Quite the contrary. We not only specify the fit but also the minimum pressure allowable to assemble the two parts, and we do this both for single units and for quantity production. We might term a shaft and rotor spider a nut and bolt with microscopic threads and assembled with a right-angled instead of a helical motion. Why not fit our bolts the same way, making our tolerances plus instead of minus and using a force fit we can depend upon when it means no change in the tools necessary for assembly?

19 If we look at a finished commercial thread through a microscope, it will be seen that the edge is serrated and that slivers stand up all over its surface. By running a nut over it once we but slightly change its appearance, but by repeatedly doing so — always using a tight nut — we may finally burnish the thread so that it will not thereafter change its size and will have a surface somewhat comparable to a case-hardened one.

20 The writer has stated that the different-angle thread was practically independent of tolerance. This is self-evident on the finishing operation. Let us now see how true it is as regards the roughing operation. Excluding the ideal condition, there are six variables that may be met with, and these are caused by large and small angle on both male and female. A diagrammatic sketch will show that of six possible variations due to angle alone, the different-angle thread is usable on four, and the U. S. thread only two. Any multiplication of these errors of angle by maximum or minimum tolerance would obviously still further favor the different-angle thread.

21 Fig. 5 shows diagrammatically the different-angle thread with the angles of the male greatly exaggerated to demonstrate the principle. We are complying in this case with all the conditions we have just been discussing: we are making the lesser contain the greater, angularly speaking; we are applying force from both directions, at right angles to the axis of the bolt; we have made the angles complementary — not merely two equal angles; we are securing the maximum strength at and near the pitch line, and transforming a thread with a lesser angle than that of the standard male to one with a larger angle, thus filling out the space perfectly and so doing away largely with vibration; and we are securing, whether under pressure or not, contact on all flanks, whereas the standard thread when under pressure secures contact on possibly one-half the flanks, both because it is compressible and because it does not fill the female thread. The only problem to be solved, therefore, was to find an angle of such slope that it could be formed without distortion of the nut or requiring too much force to screw home. To demonstrate this, threads as small as No. 10-32 were used, and as being of possible interest, the writer presents a brief outline of the engineers' report of the experimental work.

RESULTS OF EXPERIMENTAL WORK

22 Diagrams of the different threads were first laid out on a 100 to 1 scale so as to determine approximately the most suitable angles to be tested; the Löwenherz thread with an angle of 53 deg. 8 min. being used as a basis. The nut was to have the regular Löwenherz thread with same diameter and pitch as in the 155-mm. shell adapter used by the U. S. Government.

23 Threads of 44 deg. and 45 deg. for the plugs seemed most

favorable and accordingly the following cold-rolled steel plugs were made up, with nuts having the same pitch diameter and lead as the plugs but a thread angle of 53 deg. 8 min.:

1 Angle of thread 44 deg. 54 min., pitch diam. 0.8748, lead 0.0787
2 Angle of thread 45 deg. 2 min., pitch diam. 0.8748, lead 0.0787
3 Angle of thread 44 deg. 0 min., pitch diam. 0.8748, lead 0.0787

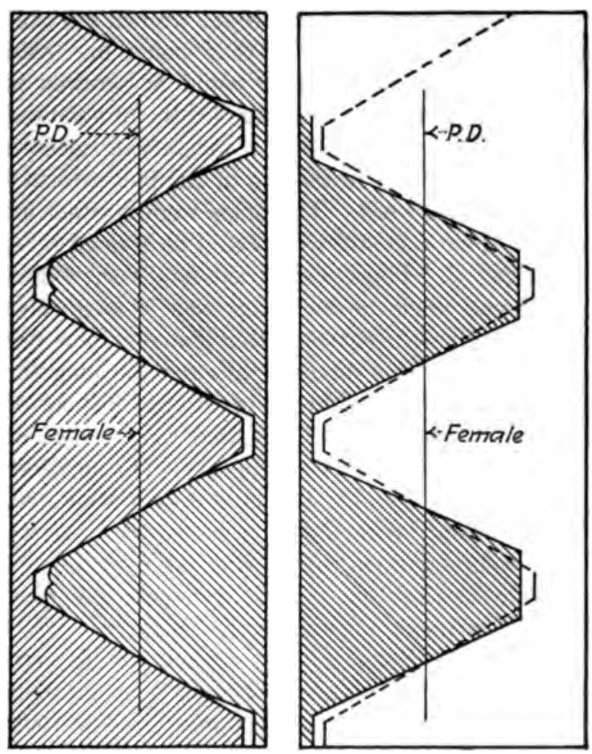


Fig. 5 Different Angle Threads (Greatly Exaggerated)

24 *Test No. 1.* Nut No. 1 and plug with thread angle of 44 deg. 54 min. were screwed together *without* a lubricant. They were started about a half a thread by hand, and then an 8-in. wrench was used for about four threads. The plug was then in so tight that a 10-in. wrench was required to turn it to full depth. After a couple of backward turns the plug stuck so tight that a 20-in. wrench would not move it. The nut was then sawed open and removed from the plug and about one-third of a thread of the plug

was taken off in a piece of the nut. A magnifying glass showed that the threads in both plug and nut were drawn and cut out of shape where there was a tendency for them to overlap, due apparently to too much metal and no lubricant.

25 *Test No. 2.* Nut No. 2 and plug with thread angle of 45 deg. 2 min. were screwed together *with* a lubricant. They were started by hand for about one-half turn, then an 8-in. wrench was used for five or six threads, and a 10-in. wrench for the remainder. The plug came out slightly easier than going in. After this had been repeated three times the plug could be screwed in by hand. The maximum and minimum plug gage for the nut showed no change in the thread of the nut. Under a magnifying glass it was seen that the metal had flowed to the top of the plug thread from about the pitch diameter outward. The plugs were screwed into the nuts fifty times and there was still what could be termed a "snug fit."

26 *Test No. 7.* Plugs were tried out with commercial 1-in. nuts. The thread angle of the plugs was 58 deg., and the pitch diameter 0.9228 in., with one plug this diameter plus 0.001 in., and one minus 0.001 in. This diameter allowed the plug thread in the layout to overlap the entire thread of the nut instead of only half, as in the previous cases. The nuts used were picked out of stock for size with a standard 1-in. thread gage. The smallest plug went into the nut easily by hand. The largest two went together easily with an 8-in. wrench. After being twice screwed together with the wrench, they went together with a snug fit by hand. The magnifying glass showed that the thread from near the pitch diameter outward had been drawn and compressed slightly.

27 *Test No. 9.* Plugs of ½-in. diameter with pitch diameters of 0.4684 in. and 0.4699 in. (same as ½-in. S. A. E. nuts) and a thread angle of 50 deg. were tried with commercial ½-in. S. A. E. nuts. The difference in the two pitch diameters made practically no difference in the fits, as they both readily went in with an 8-in. wrench. After they had been screwed together four times, they would go together by hand, but without shake. After they had been screwed together 75 times there was still what could be termed a "snug fit."

28 *Test No. 10.* An attempt was made to compare the strength of an S. A. E. standard ½-in. thread with a 50-deg. thread of the same size. The plug with the standard thread on one end and the special thread on the other end was used with standard nuts. A pull of 14,000 lb. was gradually applied and the metal began to give way, which prevented an additional load. During this

pull observations were made to determine if there was any "give" in either of the threads, but both remained the same throughout. The nuts were removed and there was no apparent distortion of the threads.

29 *Test No. 12.* In this test ¼-20 plugs with pitch diameters of 0.2165 in. and 0.2181 in. with 50-deg. angle were tried out with ¼-in. U. S. standard nuts, one being a commercial nut and the other of a standard size but made in our toolroom — the tap being 0.250 in. in diameter. Both nuts were tried with a ¼-in. standard plug gage, both being apparently the same size. The nut made in the toolroom was screwed on the maximum plug and went on about one and a half times its length and then stuck and would not go either way. It was finally removed by hammering it on the sides. The threads were rolled and torn from about the pitch diameter outward, but there was not that tendency for the metal to roll upon the outside of the thread as in the previous tests, the outside diameter being only 0.251 in. as against its original 0.250 in.

30 The commercial nut went on the minimum plug with an extremely tight fit, but it came off very readily with the wrench and left quite a different thread from the previous one. The thread was not torn at all but rolled out to almost a perfect V-thread with outside diameters of 0.2555 in. as against the original 0.250 in. The outside diameter of the tap for the commercial nut must have been over 0.250 in. to allow this metal to flow out to 0.2555 in. and not jam the nut. If the toolroom tap had been sharper on the flats or its outside diameter greater, there would have been room for the metal in the test plug to flow out to a larger diameter and avoid tearing the thread, and consequent jamming of the nut. Two facts are clearly demonstrated: first, necessity of room for metal to flow; second, one of the limitations of the thread gage.

31 *Test No. 13.* Plugs corresponding to standard 10-32 with 50-deg. angle were tried out with standard 10-32 nuts; pitch diameters 0.170 in. and 0.1716 in. They were a trifle too tight a fit to go together by hand, but after being screwed together once with a wrench they went together by hand snugly.

CONCLUSIONS

32 In conclusion it may be stated that the tests would seem to indicate the following:

a The cause of stripped threads is lack of room into which the metal can flow

b The pitch diameter should be the same in both threads
c The lead should be the same
d The thread angles should differ by not more than 10 deg.
e The limits for the inside diameter of nut need not be adhered to closely, as the inner part of the nut thread exerts very little, if any, holding power.
f The outside diameter of plug and pitch diameter of both plug and nut are important and should be adhered to fairly closely.

DISCUSSION

Donald A. Hampson. A word of explanation would make the paper clearer from the start.

The author draws several conclusions, one of which is "the cause of stripped threads is lack of room into which the metal can flow." Except in the case of a relatively small number of bolt and screw manufacturers, the term "stripped threads" immediately brings to mind a thread which has been destroyed through too much pressure on the wrench, or through long usage, or through a poor initial fit. With this viewpoint, the reader wonders, as he progresses, why more room is needed when his ideas are that stripped threads are a product of too much room.

The author's comparison of a dull die to a nut that does not have room enough for the metal to flow at the tops of the threads, and consequently strips them, is a most excellent one. Test threads cut on two bolts, one of which is over size and the other under size, will prove his conclusions as right and that a fair thread may be produced with an old die if there is not too much metal at the top.

Wilfred Lewis. The author has not only offered a very happy solution of a perplexing problem, but he has, at the same time, found a very good way to reconcile the advocates of the Whitworth and Sellers standards with each other and bring about another strong bond of union between Great Britain and the United States.

Curiously enough, the Whitworth bolt has the form of thread recommended by the author, and on this he would jam a Sellers nut. All that remains, therefore, to bring about complete accord, is to agree upon the number of threads per inch, which in many sizes already correspond, and take care that enough clearance space is provided in the nut for the metal to flow as intended. This can be done by reaming out, if necessary, but the depth of thread corresponds so closely that the writer believes commercial nuts having U. S. standard threads can now be jammed onto Whitworth bolts to make a tight fit wherever the pitches are alike.

A nut or bolt to be tight in the thread must fit on both sides, and for this reason the late Wm. Sellers was a strong advocate of tapered studs in steam hammers and other machines subject to shock. Such bolts could be set firmly in place and they did not rattle out, but it was more difficult to secure the nuts which were always tapped straight. Now the author suggests a method of securing both nuts and bolts which seems to the writer commendable, if the metal is not too hard. The writer would not care to try it on steel bolts having more than 30 carbon, and care must be taken in making these tight fits not to exceed the torsional strength of the bolts, especially in small sizes.

It must also not be forgotten that it is one thing to screw a nut on and often quite another matter to get it off, especially after some interval of time. Sellers nuts on Whitworth bolts may, therefore, in time become like rivets to be twisted off or cut apart, rather than unscrewed. Sometimes this would be a desirable feature and sometimes it would not. Generally, nuts are put on with the idea of taking them off if necessary, and, before it is safe to go ahead as the author proposes, no doubt quite a lot of experimental work must be done to determine safe limits in various materials.

H. L. Van Keuren. An occasional departure from standard practice is often justifiable, in fact it is necessary for the realization of progress. For a tight-fitting thread, the suggested expedient of using a bolt, or male member, with an intentional small angle will probably meet with more favor than either the off-lead thread of track bolts, or the proposed taper thread for stud fits. In fact, many engineers have testified that a satisfactory interchangeability resulted with the 60-degree U. S. standard thread and the 55-degree Whitworth thread.

With the intention of furthering present accomplishments in screw thread standardization, attention will be directed to certain fundamentals of direct bearing on the subject matter under discussion. These fundamentals are as follows:

a The uniform threaded hole or nut
b Tolerances are a necessity in quantity production
c Gages are the physical representations of the tolerance limits.

The Uniform Threaded Hole. Both the logic of the procedure and the opinion of a large number of American engineers serve to establish the first fundamental. The threaded hole or nut should be kept uniform, and the size of the screw varied to secure the degree of fit desired.

It is of great convenience to be able to purchase and use standard taps and gages for the threaded hole. As a rule, the die which produces the screw is adjustable, while the tap is not, and it is thus more practicable to vary the size of the screw.

Commensurate with the uniform practice, the suggestion is offered herewith that a range of tight fits might easily be obtained by the following procedure:

1 For the tightest fit, stock screws of 60-degree angle and between definite oversize limits would be used
2 For a moderately tight fit, these stock screws would be rethreaded with an adjustable 50-degree die, removing metal only below the pitch line
3 For a light fit without shake, such as a screw-driver fit, more metal would be removed in the rethreading by closing in the die.

Thus, as a practical application, there would result a double-angle thread on the screw.

Tolerances. In making but two mating parts for a desired class of fit, tolerances do not enter into the problem. The difference in dimensions between these two parts is the allowance for fit.

For quantity production, it is not possible to secure absolute uniformity in any of the elements of the threads, and so tolerances, or variations between fixed limits, must be determined and agreed upon in making the design of each part.

Gages. It is true that commercial thread gages have their limitations. However, these limitations are mainly due to the fact that thread gages are almost generally incorrectly used.

A thread gage, as previously stated, is but a physical representation of the tolerance limits expressed on the part drawing. It may be regarded simply as a model with which the finished product is compared, or it may be used directly as a measuring tool. It is evident that the design of the thread gage should depend upon the intended application.

The fact that a thread is composed of several elements — diameters at crest, pitch line and root, lead and angle — complicates the matter of using a thread gage directly as a measuring tool. In consideration of these facts, the following fundamentals regarding the design of thread gages for direct gaging would be stated:

1 The "go" gage should check simultaneously all elements of the thread
2 The "not go" gage should check separately the elements of the thread.

Thus, in gaging the effective or pitch diameter, a "not go" gage which makes contact only on a small portion of the thread sides, and which is cleared on all other elements is required. Only one or two turns of thread should be used in order to eliminate the effects of lead error in the thread being gaged.

Conclusion. The subject of tight-fitting threads is of such importance that an effort should be made to secure special consideration by the National Screw Thread Commission of the data presented in the paper under discussion, as well as to inaugurate further research and investigations necessary to determine tolerances for tight fits of standard-size threads.

P. V. VERNON. The author's argument seems to be based on the idea that screw threads should fit together tightly and should also be interchangeable.

When it is considered that a tapped hole is usually produced by a single tool without the use of a finishing tool, and that in most cases the threads of a bolt are produced by a single die without the use of a finishing die, it appears to be unreasonable to expect that male and female threads can be produced commercially with sufficient accuracy to ensure tightness, the commercial solution for interchangeable threads being to provide for sufficient slackness to ensure the threads coming together in all circumstances.

No one would expect a shaft finished by a single cutting tool without a finishing cut to fit tightly into a hole bored by a single tool without a reamer, although the turning of a shaft and the boring of a hole are much easier operations than the producing of male and female screw threads.

Although much progress has been made in producing accurate taps and dies, they have not yet reached that degree of accuracy that would ensure tight-fitting interchangeable threads.

The writer's experience is that if satisfactory results are to be obtained, threads should always be assumed to be easy fits except when specially machined for tightness, and designs should accommodate this ease of fit by the necessary locking arrangements where such are essential.

The author's statement that for a die-cut 1-in. bolt a fuller diameter would generally be produced with a bar 0.990 in. diameter than the 1-in. diameter indicates that his dies are of an obsolete nature, as with such dies, bolts produced in a turret lathe would need reducing for the thread which would otherwise be too large. With modern threading tools this is not necessary and the author's statement does not apply.

The author's method of distorting the thread of the screw to fit the nut only gives bearing for the thread about halfway along it and it is not clear that a better fit could not be obtained without this disadvantage by making the effective diameter slightly high without altering the thread angle.

It is difficult to believe that the author's system of nuts will remain standard even after they have been used to distort the thread of the bolt, general experience being that if the nut is subjected to heavy pressure in tightening the tendency is to distort the thread off both bolt and nut.

With regard to Test 2, after the plug had become an easy fit in the nut there seemed to be little point in continuing to screw it in 50 times as in the case in question, as little would be gained or could be expected to result from this.

The same applies to Test 9. The author's method of making threads could, of course, be used only in steel or other plastic metals. If used in cast iron, brass or aluminum, there would be greater danger of stripping and distorting the thread, and as the same bolts and screws are used in steel, cast iron, brass and aluminum there would certainly be trouble from this cause.

In the writer's view, one of the chief difficulties in producing interchangeable threads of the United States standard screws is the fact that the sharp corners at the top of the teeth of the tap and the bottom of the teeth of the die very soon wear away and become round, thus causing the threads to bind at these points. By providing clearance at top and bottom, these difficulties are avoided, and the advantage of introducing such clearance is very great.

This means making the outside diameter of a tap rather larger than standard while maintaining the standard effective diameter, and in the case of a die making the core diameter smaller without altering the standard effective diameter.

In the writer's own experience in producing hobs, dies and taps, the Whitworth form has proved better than the U. S. or Systeme International for producing screws to specified limits on account of the absence of sharp corners on taps and dies. The rounded threads of the Whitworth standard maintain their size very much longer than the sharp cornered threads of the above standards.

Dealing with the author's proposal to alter the angle of the male thread, it is difficult to believe that threads having a difference of 10 degrees in angle would be distorted to fit the nuts without injuring the material, as in the case of the 1-in. 8-pitch U. S. Standard bolt which formed the subject of one of the author's tests, the maximum interference at the point of the thread would be as much as 0.092, which would have to flow outward.

FIG. 6 DIFFERENT-ANGLE THREAD USED FOR SPARK PLUGS

The writer is convinced that if tight-fitting interchangeable threads are required, something better must be used than a single tool for threading the bolt and the hole respectively, and that finishing taps and finishing dies of high accuracy will have to be employed.

The writer does not believe that the use of such tools would be commercial under ordinary conditions, but does believe that continual improvements are being made in the manufacture of threading tools by which the fits of interchangeable screws and holes are gradually approaching one another very closely, although it is hardly to be hoped that tight interchangeable fits will be within the range of practical policies for a long time to come.

THE AUTHOR. In the main the discussion of Messrs. Lewis, Hampson and Van Keuren is favorable, while that of Mr. P. V.

Vernon is the opposite. Mr. Lewis and Mr. Van Keuren both suggest the possibility of using the 55-deg. Whitworth thread and the 60-deg. Sellers nut. The author agrees with them that this would be practical and would make a tight fit, but the Whitworth thread being round would not carry as much metal as a sharp cornered one and would not flow as easily; but for all practical purposes it would be better than the present practice, and, as stated in a previous article, it does

FIG. 7 TRANSFORMATION OF THREAD AFTER NUT HAS BEEN SCREWED ON

not matter whether we choose "the Whitworth standard" or not; we eventually approximate it as the die wears. Apropos of Mr. Vernon's discussion, it seems hardly germane in that he advocates a "slackness of fit" and "the necessary locking arrangements" which the rest of us are trying to get away from and as for securing interchangeable tight threads with similar angles, the probabilities are strongly against it because of the necessary three tolerances. This is indicated by the records of the Society's Thread Committee which shows no nearer approach to a solution than was shown ten years ago.

"Distortion" no more applies to the operation of screwing the nut on a different-angle bolt than it does screwing a die upon a bar of metal to produce a thread as it is a definite controllable operation which does not show any distortion of the nut, and any enlargement of the nut is merely due to laying down the rough burrs and finishing the thread. In other words, we are doing what Mr. Vernon advocates, finishing the thread in two operations instead of one, and there is no more danger of "stripping the thread" with brass or aluminum than there is with steel, as the resistance decreases with the strength.

The effective strength of a thread lies in either side of the pitch line. Generally halfway between the pitch line and the outside diameter of the thread is the limit of effective strength, and, taking into consideration the area of the standard and the different-angle male thread, the different-angle thread would seem to have much more strength than the standard 60-deg. thread.

As regards making a fit between the two without injuring the material, consider the drawing operation by which a cup is made from flat material drawn cold or what is more difficult still, the taking of a $\frac{1}{8}$-in. piece of material and increasing it to $\frac{1}{4}$ in. thick cold without distortion or lack of homogeneity. If those who doubt will try the experiments as outlined, they will secure the same results that the author did, provided they take care of the little things. This thread has been used on some difficult positions where too much wrench pressure would loosen it from its fastening.

Figs. 6 and 7 show the different angle thread used on a spark plug to make a gas-tight joint, and the transformation of the thread after the nut has been screwed on. The difference of the thread above and below the nut should be particularly noted.

No. 1751

THE FIRST TRANSCONTINENTAL MOTOR CONVOY

By E. R. Jackson,[1] Washington, D. C.

~~Non-Member~~ Member of the Society

THE Transcontinental Motor Convoy of 81 vehicles and trailers which started from Camp Meigs, Washington, D. C., at 8 A.M., on July 7, 1919, and finished at San Francisco, Cal., on September 6, 1919, was the first attempt to push through a large train of heavily loaded trucks from one end of the country to the other. Back of it were placed the resources of the War Department, and some measure of its success was due to the friendly coöperation of the local bodies of the communities through which the Convoy passed."

"The four purposes of the trip were as follows:

"1 The War Department's contribution to the Good Roads movement for the purpose of encouraging the construction of through-route and transcontinental highways as a military and economic asset.

"2 The procurement of recruits for the enlisted personnel of the Motor Transport Corps or any other branch of the U. S. Army, young men to be enlisted with the train as candidates for the mechanical training schools opened under the direction of the Motor Transport Corps.

"3 An exhibition to the general public, either through actual contact or resulting channels of publicity of the development of the motor vehicles for military purposes, which is conceded to be one of the principal factors contributing to the winning of the World War.

"4 An extensive study and observation of terrain and standard army motor vehicles by certain branches of the army, particularly

[1] Former 1st Lieut., Ord. Dept., U. S. A. Ordnance Observer.

Delivered as a lecture at the Spring Meeting, St. Louis, May 1920, of The American Society of Mechanical Engineers. For complete address see Mechanical Engineering for March 1920.

the Field Artillery, the Coast Artillery, the Air Service, the Corps of Engineers, and the Ordnance Department."

The address dwells upon the itinery, which was over the Lincoln Highway from Washington to San Francisco and following this is given the personnel of the expedition, its operations, and a discussion of the problem of maintenance.

The address ends with a statement of general conclusions and results and contains the log of a typical day's run. Numerous illustrations are included.

No. 1752

THE GERMAN DEFENSES OF THE COAST OF BELGIUM

By Lt.-Col. H. W. Miller,[1] U. S. A.
Non-Member

ON October 15, 1914, the Germans occupied Ostend and all of the Belgian Coast north of that point. During the next few days their offensive gained for them the coast for a distance of about 10 miles farther south, including the city of Nieuport. They were unable to hold all of their gains against the offensives of the Belgians and British during the next few months, and their line was pushed back to a point between Nieuport and Ostend, at Westende, about 8 miles south of Ostend. This line became a part of what has been known familiarly as the Hindenburg Line. This line was heavily fortified by the Germans, and their men were provided with adequate shelters. Four years after the original occupation, almost to a day, the Germans were driven out of Ostend, on October 17, and from Zeebrugge farther north, on the 18th."

The lecture, following this introduction, deals with the description of the city of Ostend, the coast of Belgium and the installation of the fortifications. Concerning the locations of the armament, the lecturer says:

"As a consequence of the practical impossibility of making any landing except at Zeebrugge and Ostend, no attempt was made to fortify the coast at other points. The heaviest of the fortifications were centered about Ostend, with others not much less effective about Zeebrugge. The primary armament, including the heaviest guns, was generally located at a distance of from two to five kilometers from the shore. The 38-cm. gun of the Battery Pommern

[1] Office of Chief of Ordnance, Artillery Division, Washington, D. C.

Delivered as a lecture at the Spring Meeting, St. Louis, Mo., May 1920, of The American Society of Mechanical Engineers. For complete account see Mechanical Engineering for June 1920.

at Leugenboom, thė farthest point inland, was likely placed primarily for land defense."

The distribution and caliber of all of the 30 batteries is shown by a map. The lecturer takes up the subjects of fire control, shelter, scope of service, the allied attacks on Zeebrugge and Ostend, the destruction from shell fire and bombs, the method of final destruction of the guns before abandonment, and details of the mechanisms of the various batteries, illustrated with photographs and drawings.

Speaking of the purpose of the investigation, the lecturer says:

"Two investigations were made of these defenses. One of these was for the purpose of studying the tactical distribution of the entire armament. . . . The second investigation was made by the writer for the purpose of determining whether the Germans had followed fixed policies in the designs of the various parts and mechanisms of their heavy guns and carriages."

The lecture ends with a number of significant conlcusions drawn from the investigation.

No. 1753

ANNUAL REPORT OF THE COUNCIL

A NOTABLE matter before the Council this year was the participation by the Society in the new Federated American Engineering Societies, a comprehensive organization of the engineering and allied technical bodies created to "further the public welfare wherever technical knowledge and engineering experience are involved."

The movement to establish such an organization grew from recommendations made by the Joint Conference Committee of the Founder Societies, which was established last year to consider matters of common interest to the members of these four societies, and the Joint Conference Committee was in turn the outcome of the several committees on policy of these four societies (our own committee was called the Committee on Aims and Organization), which were appointed to "review the policies of the societies in the light of present-day developments and make recommendations."

A resolution passed by the Society at the Annual Business Meeting in December 1919, requested the Council to authorize the Joint Conference Committee to take the necessary steps to bring about a national organization for the purpose as stated above.

The important actions of the Council accepting membership in the Federation are incorporated in the following resolutions of the October meeting:

> WHEREAS The Council of The American Society of Mechanical Engineers at its meeting of January 24, 1920 approved the report and recommendations of the Joint Conference Committee of the American Society of Civil Engineers, the American Institute of Mining and Metallurgical Engineers, The American Society of Mechanical Engineers, and the American Institute of Electrical Engineers, for the establishment of a comprehensive organization, and
>
> WHEREAS The Council authorized its representatives on the Joint Conference Committee to take the necessary steps in coöperation with the representatives of similar bodies, to put this organization into effect, and
>
> WHEREAS At its meeting of May 24th, 1920, the Council appointed delegates to the Organizing Conference held in Washington, June 3–4, at which The Federated American Engineering Societies was formed,

Resolved That the Executive Committee of the Council hereby confirms the action of the delegates of the Society at the Washington Conference pledging The American Society of Mechanical Engineers to charter-membership in The Federated American Engineering Societies, and

Resolved That The American Society of Mechanical Engineers accepts the invitation of the committee as the Ad-Interim Committee by authorization of the Organizing Conference, to become a Charter-Member of The Federated American Engineering Societies.

Resolved That the following be appointed as the Society's representatives on the American Engineering Council, for the first term of that body — Messrs. L. P. Alford, Charles T. Main, Arthur M. Greene, Jr., E. S. Carman, Arthur L. Rice, V. M. Palmer, H. P. Porter, R. H. Fernald, L. C. Nordmeyer, with alternates at large, Fred J. Miller, and Charles Whiting Baker.

The organization of the American Engineering Council, the governing body of The Federated American Engineering Societies, was effected in Washington in November, 1920, and the regular meeting of our Council was held in that city on the day previous.

Under the plan that the American Engineering Council would supersede the Engineering Council, resolutions were adopted authorizing the Society's representatives on the Board of Trustees of the United Engineering Society to concur in an amendment to the By-Laws of the latter to terminate Engineering Council.

Complete reports of the committees of the Council were received and were transmitted to the Society at the Annual Business Meeting in New York in December, 1920. They are summarized in the following paragraphs.

Finance. Despite the continued increased costs of all work, the Finance Committee managed to get through the year with a favorable balance of $7,232.16. The expenditures per member for the fiscal year, October 1, 1919, to October 1, 1920, were $36.50, as against $32.43 and $25.65 for the two preceding years.

Invested Funds and Trust Funds. The present standing of the invested funds of the Society is shown in the following table:

Balance Sheet, September 30, 1920

Assets

Society's one-quarter Interest in the Building Land and Real Estate Equipment, No. 25 to No. 33 West 39 Street		$486,792.79
Library Books	$13,000.00	
Furniture and Fixtures	5,000.00	
		18,000.00
Stores, including plates and finished publications		36,216.56
Engineering Index		10,000.00

Trust Funds Investment:			
Liberty Bonds		7,000.00	
New York City 3½ %, 1954 (par $45,000)		39,696.81	
St. L., Peoria & N. W. 1st 5%, 1948 (par $10,000)		10,613.89	
United New Jersey Canal Co. (par $1,000)		970.00	
Cash in banks representing Trust Funds		753.78	
			59,034.48
Liquid Assets:			
Liberty Bonds		48,000.00	
United Engineering Society		2,500.00	
Accounts Receivable:			
Members' Dues	$17,810.42		
Initiation Fees	8,169.58		
Sales of Publications, Advertising, etc.	84,438.87		
		110,418.87	
Cash: In banks for general purposes	4,643.63		
Petty Cash Fund	2,000.00		
		6,643.63	
			167,562.50
Advance Payments			33,972.96
Unexpended Appropriation 1918–1919			4,175.81
			$815,755.10

LIABILITIES

Trust Funds:			
Life Membership		$46,326.92	
Library Development		4,902.71	
Week's Legacy		1,957.00	
Melville Medal Fund		1,138.86	
Hunt Memorial		208.99	
Hess Juniors' Prize		1,000.00	
Hess Students' Prize		1,000.00	
Charles T. Main Award		2,500.00	
Total			$59,031.48
Notes Payable			39,603.33
Accounts Payable			18,323.35
James Watt Memorial			296.00
Initiation Fees uncollected			8,169.58
Replacement Fund			1,163.18
Dues paid in advance			2,471.23
Unappropriated Revenue			8,899.07
Unexpended Appropriation 1919–1920			14,133.09
Capital Investment		514,792.79	
Surplus and Reserve		148,859.00	
			663,661.79
			$815,755.10

Meetings and Program. The rapid increase in size of the Spring Meetings led the Committee to solicit an opinion as to what character these meetings should assume in the future. This opinion was in favor of continuing the Spring Meetings at industrial centers and making trips to study engineering developments and industrial plants.

The Committee reported a working arrangement with the new Standing Committee on Professional Sections regarding the conduct of meetings of Professional Sections held at the Annual or Spring Meetings. The Committee broadened the work of publicity working with an informal Publicity Committee consisting of the Chairmen of the Standing Committees on Meetings, Publications, Research, Local Sections, and Professional Sections, and with an experienced publicity man to assist in interpreting to the public the important activities of the Society.

Publication and Papers. In spite of continued increased costs, all the publications of the Society, namely MECHANICAL ENGINEERING, the TRANSACTIONS, CONDENSED CATALOGUES, THE ENGINEERING INDEX ANNUAL and the YEAR BOOK have been maintained.

The Committee reports the further development of MECHANICAL ENGINEERING. The matter in this publication has received wide notice in the technical press of the world.

In the second year of THE ENGINEERING INDEX ANNUAL, the circulation of this volume has been increased to 4,000 copies.

CONDENSED CATALOGUES, now in its tenth year, has been increased in size and usefulness.

The TRANSACTIONS was published in the usual form, although the material had to be condensed slightly.

The YEAR BOOK was issued in a new form, consisting virtually of a complete geographical list of the members, with business connections, supplemented by an alphabetical key to this list.

Whereas the balance sheet of all the publications taken together shows a small return this year to the Society, this is the first time that this is true, and when considered over a period of years it is obvious that they have been paid for both from dues and advertising.

Membership. The Membership Committee, always one of the most active committees of the Society, held eighteen meetings and received 3,106 applications. The full table of activities for the year is as follows:

Recommended for Membership	1,728
Deferred	13
Deferred indefinitely	12
Denied promotion	4
Reconsiderations	26
Reinstatement	18
In regular course of procedure	1,305
Total	3,106

The membership as of October 1, 1920, was 12,232, as against 10,504 on October 1, 1919.

Honorary Membership has been conferred by the Council upon the following:

HONORABLE SIR CHARLES ALGERNON PARSONS, of London, England, for his valuable contributions to the development and construction of the steam turbine, which has now become an essential prime mover.

LORD WILLIAM WEIR, of Glasgow, Scotland, for his energy and genius in the organization of war work in the British Isles, and particularly as Director-General of Aircraft Production in the Ministry of Munitions.

GRANDE UFFICIALE INGINIEUR PIO PERRONE, of Genoa, Italy, for the unselfish devotion of his engineering and manufacturing skill to the cause of his country in the Great War.

CAPTAIN ROBERT WOOLSTON HUNT, of Chicago, Illinois, beloved Past President of this Society and John Fritz Medallist, for his life-long and eminent achievements in the field of engineering, and particularly in connection with the steel industry.

REAR-ADMIRAL ROBERT STANISLAUS GRIFFIN, Chief of the Bureau of Steam Engineering, U.S.N., Washington, D. C., for his signal contributions to the engineering work of the naval forces of the United States during the Great War.

DR. SAMUEL MATTHEWS VAUCLAIN, of Philadelphia, Pa., a Past Vice-President of this Society, for his many and important contributions to the mechanical engineering of the steam locomotive.

In Memoriam. The Council lost by death its esteemed member, Vice-President John R. Allen. Professor W. B. Gregory was appointed in his place.

Our beloved Past-President, Dr. John A. Brashear, died during the year and a Memorial Committee was appointed which framed resolutions. Also a Memorial Service was held at the Annual Meeting, 1920; an account of this appears elsewhere in this volume.

Resolutions upon the death of Dr. Brashear and Past-President Samuel T. Wellman were adopted and ordered spread upon the records of the Society.

The losses to the Society by death are recorded in the Necrology, published in this volume.

Local Sections. Nine new Local Sections have been established during the year at Akron, Columbus, Oregon (Portland), Pittsburg, Providence, Syracuse, Toledo, Utica and Washington (Seattle). This brings the total number of Local Sections up to 38.

An important event in the history of the Committee was the transference to it by the Council of the supervision of increasing the membership of the Society. As an incentive to the Local Sections to develop this activity, the Council voted to return for the benefit of the section a portion of the initiation fees of new members.

The officers of the Society and the Secretary have paid their usual visits to the Sections to assist in their development. At the conferences of Local Sections delegates held at the Spring Meeting in St. Louis and at the Annual Meeting in New York, important matters of policy were discussed. These conferences have become a useful means of ascertaining an opinion of the cross-section of the Society upon any matter, and in fact the conference at the Annual Meeting was requested by the Council to consider and report back upon financial policies.

The conference at the Annual Meeting also nominated to the Annual Business Meeting the Regular Nominating Committee for 1921, in accordance with the precedent established last year.

The report of the Committee includes suggestions for the future work of the Local Sections. An important recommendation is the appointment of regional secretaries in large centers.

Professional Sections. This year saw the inauguration of Professional Sections, similar to the Local Sections, but consisting of professional groups instead of geographical groups.

Eight such sections were organized, Fuels, Machine Shop, Management, Materials Handling, Ordnance, Power, Railroads and Textiles, and sections on Aeronautics, Cement, and Gas Power are in contemplation.

The Council adopted new By-Laws governing the Professional Sections, by which the elected chairmen of the Sections became the Standing Committee on Professional Sections, and in turn the chairman elected by this committee took a seat in the Council. Mr. Edwin B. Katte was elected chairman of the Standing Committee.

Over 6,000 members registered in the Professional Sections during the year. Six of the sections held sessions at the Annual Meeting in 1920.

Upon the organization of the Standing Committee, the Special

Committee which was appointed by the Council to inaugurate this work was discharged.

Constitution and By-Laws. The report enumerates the following amendments to the Constitution, which were put into effect at the Spring Meeting, 1920:

1 That the Committee on Professional Sections become a Standing Committee of Administration
2 That the chairman of all Standing Committees have a seat in the Council without vote
3 That the Society may approve or adopt reports, standard formulæ or recommended practices
4 That membership on the Council or any committee terminate automatically on account of absence.

Two new amendments to the Constitution, one on extending the privilege of voting to Juniors, and the other on shortening the time necessary to complete amendments to the Constitution, were presented in St. Louis.

The continued tendency to amend the Constitution led the committee to recommend a Constitutional Convention with a view to having the general policy of the Society unified and adopted and then phrased into a new Constitution.

The report enumerates changes in the By-Laws and Rules made during the year.

House. Since the Society has been housed in the Engineering Societies Building, the duties of this committee have been confined to the custody of the memorabilia and the pictures, and to supervise the furnishings, upon joint recommendation of the President and the chairmen of the respective committees. These duties have by act of the Council been transferred to the Finance Committee, who will hereafter assume charge of all the rooms of the Society and their contents.

Research. Progress reports of sub-committees on Lubrication, Bearing Metals, Fluid Meters, Heat Transmission, Permanency and Accuracy of Indication of Instruments, Riveted Joints, Cutting Action of Cutting Tools, and Vibration Strains in Shafting are included.

The Committee has been coöperating with the Publication and Papers Committee by preparing the section on Engineering Research, published in MECHANICAL ENGINEERING, and with the Committee on Local Sections through the appointment of local sub-committees on Research.

Standardization Committee. The work of this Committee has proceeded along three distinct lines, (*a*) presenting recommendations to the Council for the acceptance of sponsorships in standardization, proposed by the American Engineering Standards Committee, (*b*) forming sectional committees under the rules of the A. E. S. C., and (*c*) selecting members of the Society to represent it on sectional committees organized by other standardizing bodies in the United States.

The five sectional committees of the A. E. S. C., the personnel of each of which is entirely new, are:

1 Standardization and unification of screw threads, in coöperation with the Society of Automotive Engineers
2 Plain limit gages for general engineering work
3 Standardization of gears, jointly with the American Gear Manufacturers' Association
4 Standardization of ball bearings, jointly with the Society of Automotive Engineers
5 Safety code for mechanical transmission of power, jointly with the International Association of Industrial Accident Boards and Commissions, and the National Workmen's Compensation Service Bureau.

Boiler Code. The Committee has continued its regular meetings for the purpose of rendering interpretations and formulating replies to communications from those interested in boiler construction and the Society's rules.

Sub-Committees on Boilers of Locomotives and on Air Tanks and Pressure Vessels have presented progress reports, which have been widely discussed.

The work on air tanks and pressure vessels is necessitating further study of the question of welding of seams and the American Welding Society and the American Refrigerating Engineers have taken special interest.

The maintenance of the affidavits covering the sale of boiler stamps is still continued, as well as the sale of the publications connected with the Code. The volume of this work has greatly increased.

Power Test Codes. The work of revising the individual codes of the Power Test Codes, inaugurated last year, has proceeded satisfactorily and two of the revised codes have been presented to the Council and ordered submitted to the Society for discussion before final adoption.

Student Branches. The Special Committee on Relations with Colleges presented detailed rules for the applications of colleges for the establishment of Student Branches.

Education and Training. This is now a permanent Committee authorized by the Council on the recommendation of the Special Committee on Education Below College Grade, the actions of which Special Committee the report enumerates.

The Special Committee recommended a permanent Committee to communicate with other agencies and associations toward classifying and reporting the different systems of training for men in the industries, in the hope that greater unity of purpose in engineering education may be brought about.

The Committee recommends the formation of sub-committees, (*a*) for relations with colleges, (*b*) for relations with public and private vocational schools, and (*c*) for relations with corporation schools.

Past-President Ira N. Hollis, Chairman of the Committee, introduced a resolution before the Council recommending that this subject be taken up in a larger way by Engineering Foundation and offering the coöperation of the Council in carrying out our Society's share of investigations suggested in the report of the Committee.

Awards and Prizes. The Special Committee recommends a permanent Committee to institute and administer the following awards by the Society: (*a*) Honorary Membership, (*b*) Life Membership, (*c*) A. S. M. E. Medal, (*d*) Honorable Mention, (*e*) Scholarships or Fellowships, (*f*) Junior Medal (the present Junior Prize), (*g*) Two Student Medals (the present Student Prizes).

(*a*) is to be awarded as now; (*b*) for the best contribution to the literature of Mechanical Engineering; (*c*) for some notable invention or striking improvement in connection with the industries; (*d*) for such inventions or improvements in engineering as merit the attention of the Society; (*e*) for exceptional attainment in college work; (*f*) for the best paper contributed by a Junior Member; (*g*) for the best papers submitted by members of Student Branches.

The report lists the funds already established and available to carry out some of the above awards. A new fund is that established by Past-President Charles T. Main and which bears his name. The income from this fund may be used to make awards in addition to any of the above awards.

Code of Ethics. The Council received the report of this Special Committee in the Spring and it was referred to the Spring Meeting.

No. 1754

MEETINGS SEPTEMBER — DECEMBER

MEETINGS OF SECTIONS

A RECORD of the meetings of the 38 local sections of the Society held during the latter half of the year is given in the following pages. On November 5, a special meeting was held in New York to celebrate the 40th anniversary of the first meeting of the Society, and simultaneous meetings of a similar character were held by a majority of the local sections at their headquarters.

AKRON

July 14: Organization meeting.

October 29: Business meeting.

November 5: 40th anniversary meeting held in the rooms of the Engineering Society of Akron. Prof. F. O. Ellenwood, chairman of the Akron Section, presided.

December 17: Fuels of the Future, by Prof. F. O. Ellenwood.

ATLANTA

November 5: 40th anniversary meeting. Inspection trip in the afternoon to the Federal Prison. History and Development of the Society, by Earle F. Scott and Dr. J. S. Coon.

November 23: Manufacture of Cotton Goods, by J. T. Wikle.

December 28: Impressions of the Annual Meeting, by N. C. Harrison.

BALTIMORE

October 5: The Industrial Use of Gas, by Thompson King, Consolidated Gas Electric Light and Power Co.

November 4. 40th anniversary meeting. Joint meeting with Engineers' Club of Baltimore. Inspiring Outlook Before American Engineering, by M. L. Cooke.

BIRMINGHAM

November 5: 40th anniversary meeting. Trip through the American Cast Iron Pipe Co. plant, Sloss-Sheffield Steel & Iron By-Product Plant and the Preston Motor Corp. automobile factory. Dinner and meeting in the evening at Southern Club. Address on The Manufacture of Phonograph Records and the Technic of Recording, by A. M. Kennedy.

December 29: The Land of Promise as Seen Today, by H. Y. Carson, former Captain, U. S. A. (with General Allenby), Illustrated with lantern slides.

BOSTON

October 19: Inspection of the Fore River Shipbuilding Co., Quincy, Mass. Address on shipbuilding.

November 5: 40th anniversary meeting. Conditions in the Far East, illustrated, by John R. Freeman. Early days of the Society, by F. W. Dean. Mayor Peters also spoke.

December 17: Joint meeting with Boston Society of Civil Engineers. Water Powers of New England, by H. K. Barrows and W. S. Murray.

BUFFALO

October 26: Steam Railroad Electrification, by S. T. Dodd, General Electric Co.

December 21: Industrial Application of Electric Furnace, by E. F. Collins, General Electric Co.

CHICAGO

October 22: Railroad Situation of Today, by W. J. Finley of the Chicago Northwestern Railroad.

November 22: Celebration of the 40th anniversary of the Society at the Sherman Hotel. Industries' Supply of Energy, by Dr. G. O. Smith,[1] The A.S.M.E., by H. S. Philbrick, and Early Recollections, of the Society, by Captain Robert W. Hunt.

CINCINNATI

November 5: 40th anniversary meeting. The A.S.M.E., by James B. Stanwood and The Work of the Local Engineers, by George W. Galbraith.

[1] See MECHANICAL ENGINEERING, March 1921.

CLEVELAND

October 5: Relation to the Industry of the National Screw Thread Commission's Report, by E. C. Peck of the Cleveland Twist Drill Co.

October 12: Dinner and celebration of Mr. Carman's election as President of the A. S. M. E.

November 5: 40th Anniversary meeting at the Business Men's Club. Addresses by President-Elect Carman, Past-President W. R. Warner, and W. R. Rose.

COLORADO

November 26: 40th anniversary meeting. Engineering at the University of Colorado, by Dr. George H. Norlin, President of the University of Colorado, and The Activities of the A.S.M.E., by T. L. Wilkinson.

December 29: Rubber Compounding and Tire Construction, by John G. Gates of the Gates Rubber Co.

COLUMBUS

November 5: 40th anniversary meeting. Observations During my Recent European Trip, Col. Edward A. Deeds, Dayton, O.

November 12: Design and Construction of the Uniflow Boiler, by Howard J. Webster, Uniflow Boiler Co., Philadelphia, Pa.

December 10; Emergency Work in Power Plant Operation, by E. A. Hitchcock and movies of A. M. Byers Genuine Wrought Iron Pipe Manufacture.

CONNECTICUT

Bridgeport Branch

October 29: Joint meeting with New Haven Am. Chem. Soc., Production of Lead, by John R. McGregor.

November 5: With Meriden and New Haven Branches. 40th anniversary meeting. Excursion to plant of Seamless Rubber Co., New Haven. After dinner speech on Observations in Europe, by Mason Britton, *American Machinist*. Developments of Superpower, by W. S. Murray.

Hartford Branch

October 18: The Superpower System, by John A. Stevens, Lowell, Mass., and W. S. Murray.

November 5: 40th anniversary meeting. The Development of the Pumping Engine, by Prof. Arthur M. Greene, Jr., Rens. Poly. Inst., Troy, N. Y.

December 6: The Prevention of Grinding Wheel Accidents, by A. Rousseau, Norton Co., Worcester, Mass. The Art of Grinding Becomes Science in 1920, by W. H. Chapman, Norton Co.

Meriden Branch

September 8: Business meeting.

October 12: Visit to plant of Connecticut Telephone and Electric Co. Demonstration of Gasoline Motor Testing on Dynamometer, by J. H. Bartholomew. Oscillographic Demonstration of Ignition Coil Operation, by J. A. Terrell. Address on Generation of Electrical Oscillations by Thermionic Tubes, by H. P. Donle.

November 5: 40th anniversary meeting. See Bridgeport Branch.

November 23: Making of Power, by George A. Orrok.

December 21: Heating Systems, by Charles N. Flagg; Automobile Storage Battery, by Norman H. Schleiter; Spark Advance, by Charles E. Stahl.

New Haven Branch

September 8: Business meeting.

October 13: Hydro Power Developments in Connecticut, with Special Reference to the Housatonic River and Dam at Stevenson, by A. J. Campbell and Mr. Hughes of the Conn. Light and Power Co.

October 21: Joint meeting with Winchester Engineers Club. Dry Cells, by D. H. Gleason: Industrial Motor Control, by Prof. L. W. W. Morrow; Mechanical Reproduction of Pictures, by Coit Colburn; Motion Pictures on the Anatomical Structures of the Heart and Microscopical View of the Blood Circulations.

November 5: 40th anniversary meeting. See Bridgeport Branch.

Waterbury Branch

October 2: Inspection of the new hydroelectric plant at Stevenson, Conn.

October 29: Joint meeting with the New Haven American Chemical Society. Production of Lead, illustrated with motion pictures, by John R. McGregor.

November 5: 40th anniversary meeting. In a very real sense this was the keynote session of the anniversary, for it was at the Platt Mills Laboratory of Professor Bristol that the phonograph records by President Miller and Past-President Hollis were synchronized with the moving films. It was the first public exhibition of the synchronizing device.

December 6: Experiences in the Shipping Board, by Darragh de Lancey.

DETROIT

November 5: 40th anniversary meeting. History of the A.S.M.E. by Henry M. Leland, and An Engineer's Observations in Europe, by E. J. Mehren, Editor, of the *Engineering News-Record.*

EASTERN NEW YORK

November 15: Tendencies in Modern Armament, by Brigadier General W. I. Westervelt.

HOUSTON

October 14, 15, 16. Joint meeting with Mid-Continent Section, at Dallas, Texas. (See under Mid-Contiment Section.)

INDIANAPOLIS

November 5: 40th anniversary meeting. Past and Future of Engineering, by Dr. C. L. Mees, Pres., Rose Poly. Inst., Terre Haute, Ind.

LOS ANGELES

November 5: Professor C. C. Thomas, who addressed the meeting, was entertained as guest of the Section.

December 11: Inspection of Los Angeles Shipbuilding & Drydock Co. and inspection of harbor by boat. Address in evening on Some Interesting Phases of Shipbuilding, by James Reed.

METROPOLITAN

(New York and Vicinity)

October 22: Subject: Relative Advantages of Modern Steam and Electric Locomotives. Papers: Steam Locomotive Advantages, by J. E. Muhlfeld; Electric Locomotive Advantages, A. H. Armstrong, and F. H. Shepard. Published in abstract, MECHANICAL ENGINEERING, December 1920.

November 5: 40th anniversary meeting. Addresses by A. P. Davis, W. L. Saunders, Chas. F. Scott (representatives of the Four National Engineering Societies) and also by J. Herbert Chase, Samuel Gompers and William B. Dickson, on the Opportunity and Responsibility of the Engineer.

November 17: Joint meeting with A.I.E.E. and A.S.C.E. Urban and Suburban Passenger Transportation.

December 3: Joint meeting with the A.I.E.E. and Taylor Society. The Three-Shift System in the Steel Industry, by Horace B. Drury; The Point of View of the Manufacturer, by Wm. E. Dickson.

MID-CONTINENT

October 14, 15, 16: Joint meeting with Houston Section. The following papers were presented: Lubricating Oils for Kansas Institutions, by Professor J. P. Calderwood; Evaporation Losses from Crude Oil in the Mid-Continent Field, by J. H. Wiggins; Standardization of Drill-Pipe Threads, by F. L. Scott; General Discussion — Oil-Field Boilers; Flow of Fluids Through Pipe Lines and the Effect of Pipe-Line Fittings, D. E. Foster; Report of Research Committee, by F. P. Walker; The Manufacture of Pipe, by Chas. Fitzgerald; Federated American Engineering Societies, by H. P. Porter; Helium, by Lt.-Com. R. G. Walling, U. S. N.; Cement and Cement Manufacture, by Wm. Moehler.

MINNESOTA

October 4: Water Treatment for Industrial and Municipal Purposes, by L. I. Birdsall, Supt. Minneapolis Filtration Plant.

November 5: 40th anniversary meeting. Ladies night in Palm Room of St. Paul Hotel. Members of all engineering societies of the state invited.

NEW ORLEANS

November 5: 40th anniversary meeting held at Tulane University with addresses on Society Affairs.

ONTARIO

November 5: 40th anniversary meeting. Queenston-Chippawa Power Development of the Hydroelectric Commission of Ontario, by Prof. James Trail.

November 19: Joint meeting with Toronto Branch of American Society of Steel Treating. The Manufacture of High-Speed Steel, by F. Kremp, Canadian Atlas Crucible Steel Co. Illustrated by motion pictures.

December 17: Visit to the Central Technical High School, at which time Harry H. Angus, engineer of the school gave a talk on the engineering features of the plant and building.

PHILADELPHIA

October 26: Bearings, by Albert Kingsbury.

November 5: 40th anniversary meeting. Smoker at Hotel Adelphia. Manufacture of Tobacco, by Ben Litchy and The Engineer's Place in History, by Dr. Hollis.

December 13: Joint meeting with A.I.E.E. of Philadelphia. Superpower Survey, by W. S. Murray.

PROVIDENCE

October 5: Management, by President F. J. Miller.

November 5: 40th anniversary meeting. Reminiscences of Early Days of Society, by L. D. Burlingame. Oil Refining, by C. E. Emmons, Chemist, Texas Co.

November 6: Trip through East Providence Plant of the Standard Oil Co.

December 14: Joint meeting with Machine Shop Section, Providence Engineering Society. The Engineer in a Republic, by Charles H. Norton, Norton Co., Worcester, Mass.

ROCHESTER

November 26: Debate on The Compulsory Adoption of the Metric System, between Howard Richards, Jr., of New York City and Luther D. Burlingame of Providence, R. I.

ST. LOUIS

July 1: Business meeting.

September 24: Business meeting.

November 5: 40th anniversary meeting. Address by Dean C. R. Richards, Univ. of Illinois. Motion Pictures of C. H. Howards Steel Foundry.

November 26: What Constitutes Jay Automobile Driving, by Hugh K. Warner.

SAN FRANCISCO

November 4: 40th anniversary meeting. Excursion to Mare Island Navy Yard.

December 30: Motion pictures on Refining of Copper. Addresses on the past work of the Society by Dr. W. F. Durand and Robert Sibley.

SYRACUSE

November 4: 40th anniversary meeting. Dinner and smoker at Chamber of Commerce Club House. Address by Dean Dexter S. Kimball.

TOLEDO

November 4: 40th anniversary meeting. New Developments in Steam Distribution, by George H. Hartman.

December 1: Joint meeting with the Toledo Chapter of the Doherty Fraternity. Moving picture lecture on the Manufacture of Wrought Iron Pipe, by Fletcher Collins, Pittsburgh, Pa.

WASHINGTON, D. C.

November 5: 40th anniversary meeting. Review of Work and History of the Society, by Prof. Robert H. Fernald; Points on Burning Fuel Oil, by Lt-Com. W. R. Burnell; Manufacture of Gas as an Industrial Fuel, by Thomas King; Flow Meters, by C. L. Davison.

December 2: Development of Artillery Material,[1] by Major G. F. Jenks, U. S. A. The Scientific and Engineering Work of the Government,[2] by Dr. Edward B. Rosa, Chief Physicist, National Bureau of Standards, Washington, D. C.

WASHINGTON STATE

October 1: Thermal Conservation, by H. V. Carpenter.

November 5: 40th anniversary meeting. Joint meeting with Student Branch, Univ. of Washington, at Faculty Men's Club.

December 15: Mechanical Engineering of Public Buildings, by E. L. Weber.

WORCESTER

November 5: 40th anniversary meeting. First of a series on Woıcester industries, Steel. Address on accomplishments of the Society by George I. Alden. A Unique Rolling Mill, by Jerome R. George.

December 16: The Construction of Electric Power Cable, by C. F. Hood; Looking Backward at the Wire Industry, by A. G. Warren; Reducing Scıap and Waste in a Wire Mill, by W. G. Hall.

[1] See MECHANICAL ENGINEERING, March 1921.

[2] See MECHANICAL ENGNIEERING, February 1921.

THE ANNUAL MEETING

The Forty-First Annual Meeting of the Society was marked by the richest professional program of any meeting held by the Society. To this the newly formed Professional Sections contributed greatly. In the arrangement of general Professional Sessions no more than three papers were assigned to any session, and the time for presentation and discussion was limited by newly formulated rules.

The meeting was formally opened on Tuesday evening, December 7 with President Miller's address on The Engineer's Service to Society, following which honorary memberships were conferred upon the following men:

CAPTAIN ROBERT WOOLSTON HUNT, of Chicago, Ill., Past-President of the Society and John Fritz Medallist, for his life-long and eminent achievements in the field of engineering and particularly in the steel industry:

REAR-ADMIRAL ROBERT STANISLAUS GRIFFIN, of Washington D. C., for his signal contributions to the engineering work of the naval forces of the United States during the war;

DR. SAMUEL MATTHEWS VAUCLAIN, of Philadelphia, Pa., Past-Vice-President of the Society, for his many important contributions to the mechanical engineering of the steam locomotive;

HONORABLE SIR CHARLES ALGERNON PARSONS, of London, England, for his valuable contributions to the development and construction of the steam turbine;

LORD WILLIAM WEIR, of Glasgow, Scotland, for his energy and genius in the organization of war work in the British Isles, and particularly as Director-General of Aircraft Production in the Ministry of Munitions; and

GRANDE UFFICIALE INGEGNERE PIO PERRONE, of Genoa, Italy, for the unselfish devotion of his engineering and manufacturing skill to the cause of his country in the war.

The report of the tellers of election was presented by Dr. H. G. Tyler, who announced the election of officers for the ensuing year as follows: President, Edwin S. Carman, Clèveland, Ohio; Vice-Presidents, Leon P. Alford, Montclair, N. J.; J. L. Harrington, Kansas City, Mo.; and Robert B. Wolf, New York. N. Y.; Managers, Henry M. Norris, Cincinnati, Ohio; Louis C. Nordmeyer, St. Louis, Mo.; and C. C. Thomas, Los Angeles, Cal.

There then followed simple ceremonies in which President-Elect Carman was welcomed by President Miller. At the conclusion of the ceremonies, the Presidential Reception was held and the remainder of the evening devoted to dancing.

Contrary to the usual procedure, three sessions of the meeting were held on Tuesday, previous to the formal opening. On Monday, December 6, the Local Sections delegates gathered for their annual conference, which proved of great value as a meeting place for the discussion of Local Sections' problems and for the advancement of new ideas in Society policy. For a more complete account of the Local Sections' Conference, see MECHANICAL ENGINEERING, February 1921, Part Two.

The social functions consisted of the Presidential reception on Tuesday evening, and the Ladies' Reception, Tea and Dance on Thursday afternoon. Thursday evening being left open for other than Society activities, a number of Technical Schools availed themselves of the opportunity for Reunion Meetings.

Wednesday evening was devoted to a memorial meeting in honor of the late Dr. John A. Brashear, Past-President of the Society. Mr. Worcester R. Warner, after a few words bearing on his intimate connection with Doctor Brashear, introduced the speaker of the evening, Dr. Henry S. Prichett, President of the Carnegie Foundation for the Advancement of Teaching.

Other notable features of the meeting were the Keynote Session on Transportation and the Management Session which was devoted to an appreciation of the work of the làte Henry Laurence Gantt.

The registration exceeded that of any previous meeting, being a total of 2171, with 1368 members and 803 guests.

An account of the business meeting of the Society will be found in MECHANICAL ENGINEERING, January 1921.

PROGRAM

Monday Morning, December 6

Meeting of Power Test Codes, Boiler Code Committee, and Local Sections Committee.

Monday Afternoon

Conference of Local Sections Delegates.

Tuesday Morning, December 7

Opening of Headquarters and Registration Bureau.
Council Meeting.

Tuesday Afternoon

SIMULTANEOUS PROFESSIONAL SESSIONS

FUEL

FUEL SUPPLY OF THE WORLD, L. P. Breckenridge.[1]

FUEL CONSERVATION: THE NEED FOR A DEFINITE POLICY AND ITS REQUIREMENTS, D. M. Myers.[1]

DISTILLATION OF FUELS AS APPLIED TO COAL AND LIGNITE, O. P. Hood.

FORM VALUE OF ENERGY IN RELATION TO ITS PRODUCTION, TRANSPORTATION AND APPLICATION, Chester G. Gilbert and Joseph E. Pogue.[1]

FOREST PRODUCTS

A PHOTOGRAPHIC STUDY OF THE WOODWORKING INDUSTRY, F. F. Murray.

ENGINEERING IN FURNITURE FACTORIES, B. A. Parks.

USE OF WOOD IN FREIGHT CAR CONSTRUCTION, H. S. Sackett.

MACHINING RAILROAD CROSS-TIRES, D. W. Edwards.

CREOSOTED WOOD-BLOCK FACTORY FLOORS, L. T. Ericson.

PROCESSES AND EQUIPMENT USED IN WOOD PRESERVATION, E. S. PARK and J. W. Weber.

ELECTRICALLY DRIVEN SAWMILLS, A. E. Hall.

MACHINE SHOP

SIDE-CUTTING OF THREAD-MILLING HOBS, EARLE BUCKINGHAM.

CYLINDER GRINDING IN 1920, W. H. Chapman.

MECHANICAL ENGRAVING AND DIE-SINKING, J. F. Keller.

Tuesday Evening

PRESIDENTIAL ADDRESS, Fred J. Miller, President of the Society.

Conferring of Honorary Membership upon LORD WILLIAM WEIR, Glasgow, Scotland; HON. SIR CHAS. ALGERNON PARSONS London, England; COMMANDA

[1] See MECHANICAL ENGINEERING, January 1921 for abstract of paper and discussion.

TORE PIO PERRONE, Genoa, Italy; REAR-ADMIRAL R. S. GRIFFIN, U. S. N.; CAPTAIN ROBERT W. HUNT, Chicago, Ill.; DR. SAMUEL M. VAUCALAIN, Philadelphia, Pa.

Report of Tellers of Election.
Introduction of President-Elect.
Reception and dance.

Wednesday Morning, December 8

Business Meeting; reports of special and standing committees.
Meeting of Committee on Relations with Colleges.

Wednesday Afternoon

SIMULTANEOUS PROFESSIONAL SESSIONS MANAGEMENT

MANAGEMENT

MR. GANTT'S CONTRIBUTION TO INDUSTRY, Fred J. Miller.

MR. GANTT'S CONTRIBUTION TO SHIPBUILDING, SHIP OPERATION, ORDNANCE AND AIRCRAFT, Marshall Evans.

LIFE WORK OF H. L. GANTT, E. A. Lucey.

AN APPRECIATION FROM FRANCE, C. de Freminville.

HENRY LAURENCE GANTT, J. F. Butterworth.

PRINCIPLES OF INDUSTRIAL PHILOSOPHY, W. N. Polakov.[1]

DESIGN

FOUNDATIONS FOR MACHINERY, N. W. Akimoff.

RATIONAL DESIGN OF HOISTING DRUMS, E. O. Waters.

TESTS OF REAR-AXLE WORM DRIVES FOR TRUCKS, K. Heindlhofer.[2]

RAILROAD

STATIC ADJUSTMENT OF TRUCKS ON CURVES, R. Eskergian.

INCREASING THE CAPACITY OF OLD LOCOMOTIVES, C. B. Smith.[3]

MODERNIZING LOCOMOTIVE TERMINALS, George W. Rink.[3]

RESEARCH

CALIBRATION OF NOZZLES FOR THE MEASUREMENT OF AIR FLOWING INTO A VACUUM, Wm. L. de Baufre.

THE HEAT INSULATING PROPERTIES OF CORK AND LITH BOARD, A. A. Potter, J. P. Calderwood, A. J. Mack and L. S. Hobbs.

THE CONSTITUTION AND PROPERTIES OF BOILER TUBES, Albert E. White.

Wednesday Evening

Oration on JOHN BRASHEAR AS SCIENTIST AND HUMANITARIAN, in memory of the late Dr. John A. Brashear, by Dr. Henry S. Prichett.

[1] See MECHANICAL ENGINEERING, April 1921.
[2] See MECHANICAL ENGINEERING, November 1920.
[3] See MECHANICAL ENGINEERING, January 1921.

Thursday Morning, December 9

TRANSPORTATION SESSION

RAILROADS, Daniel E. Willard.
RAILROAD FEEDERS, Charles A. Morse.
WATERWAYS, General Frank T. Hines.
MOTOR-TRUCK TRANSPORTATION, Francis W. Davis.

Thursday Afternoon

SIMULTANEOUS SESSIONS

TRANSPORTATION

TERMINALS, Col. William Barclay Parsons.
THE NEW YORK TERMINAL PROBLEM, Gustav Lindenthal.

RESEARCH

EFFECT OF FITTINGS ON FLOW OF FLUIDS THROUGH PIPE LINES, Dean E. Foster.
STEAM FORMULAS, R. C. H. Heck.
Organization meeting of the Ordnance Session.
Student Branch Conferences.

Friday Morning, December 10

SIMULTANEOUS SESSIONS

POWER

POLICIES OF FUTURE POWER DEVELOPMENT, John Jackson.
EFFECT OF LORD FACTORS ON COST, Peter Junkersfeld.
EFFECT OF SIZE OF PLANT ON COST, George A. Orrok.[1]

TEXTILE

ORGANIZATION AND CONSTRUCTION OF DYE HOUSES, A. W. Benoit.
POWER APPLICATIONS TO COTTON-FINISHING PLANTS, Leo Loeb.
CLOTH FOR MECHANICAL USES, James W. Cox, Jr.[2]

DESIGN

EXPERIENCES WITH LARGE CENTER-CRANK SHAFTS, Louis Illmer.
DESIGN OF FLYWHEELS FOR RECIPROCATING MACHINERY CONNECTED TO SYNCHRONOUS GENERATORS OR MOTORS, R. E. Doherty and R. F. Franklin.

GENERAL

PRINCIPLES OR THE GYRO COMPASS, George B. Crouse.[3]
THE ARMOR-PLATE AND GUN-FORGING PLANT OF THE U. S. NAVY DEPARTMENT AT SOUTH CHARLESTON, W. VA., Roger M. Freeman.

[1] To be published later.
[2] See MECHANICAL ENGINEERING, March 1921.
[3] See MECHANICAL ENGINEERING, November 1920.

No. 1755

THE ENGINEER'S SERVICE TO SOCIETY

PRESIDENTIAL ADDRESS, 1920

By Fred J. Miller, Center Bridge, Pa.
President of the Society

IN what turned out to be his inaugural address as President of The Federated American Engineering Societies at its first meeting held in Washington last month, Herbert Hoover said many things that showed very clearly the long distance we have traveled within recent years in our search for industrial peace and industrial efficiency.

Speaking of the part being and to be taken by engineers in this movement and after referring to the grave dangers that may result from contention and warfare between the various economic groups, each seeking to promote its own interests, he said:

> The engineers should be able to take an objective and detached point of view. They do not belong to the associations of either employers, or of labor, of farmers, of merchants, or bankers. Their calling in life is to offer expert service in constructive solution of problems, to the individuals in any of these groups. There is a wider vision of this expert service in giving the group service of engineers to group problems.

Further on he said:

> The employer sometimes overlooks a fundamental fact in connection with organized labor in the United States. This is that the vast majority of its membership and of its direction are individualists in their attitude of mind and in their social outlook; that the expansion of socialist doctrines finds its most fertile area in the ignorance of many workers, and yet the labor organizations, as they stand today, are the greatest bulwark against socialism. On the other hand, some labor leaders overlook the fact that if we are to maintain our high standards of living, our productivity, it can only be in a society in which we maintain the utmost possible initiative on the part of the employer; and further, that in the long run we can only expand the standard of living by the steady in-

An address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

crease of production and the creation of more goods for division over the same numbers.

The American Federation of Labor has publicly stated that it desires the support of the engineering skill of the United States in the development of methods for increasing production, and I believe it is the duty of our body to undertake a constructive consideration of these problems and to give assistance, not only to the Federation of Labor but also to the other great economic organizations interested in this problem, such as the employers' associations and the chambers of commerce.

Last March, speaking before the Chamber of Commerce at Boston, he said:

> We must surround employment with assurance of just division of production. We must enlist the interest and confidence of the employees in the business and in business processes. To do these things requires the coöperation of labor itself, and to obtain coöperation we must have intimate organized relationship between employer and employee. They are not to be obtained by benevolence. They can only be obtained by calling the employees to a reciprocal service.

In a story published in one of our magazines last year, the author, Ellen N. La Motte, remarked: "The ability to perceive the obvious is a rare and disconcerting gift."

I wish to call your attention to some obvious things; not only because they are related to, or intimately connected with, the work of the engineer, but also because I believe that engineers in general have more than some others this rare and sometimes disconcerting gift.

It is clear that ours is the professional society of the industries, none of which are or can be carried on without the work of the mechanical engineer, who not only provides the machinery, but also plans the plant and organizes and manages the force of workers.

The engineer must increase the effectiveness of labor, not by driving, by oppression, or suppression — the education and training of the engineer are not needed for that — but by the application of brains in industrial organization and management of men as well as of materials.

A certain deservedly popular railroad president, now retired, used often to say, in his charming after-dinner speeches, that the humblest employee of his road might aspire to its presidency. Thereupon we were regaled by a new broadside of editorials based upon the assumption that the man tamping ties ten hours per day at a wage of \$1.10 had no cause for complaint, because not being president of the road at a salary of \$50,000 per year was entirely his own fault.

We have gotten past that sort of thing because it is clearly seen that if we are to have railroads some one must tamp ties and the welfare of the tie tamper is the concern of the public, especially where the man is an actual or potential citizen and is able not only to express his dissatisfaction in his vote but to pass on to his children his attitude toward the industrial and social organization.

We must face the fact that so far as we can see a large proportion of working people must continue throughout their lives to work, under the direction of others, at tasks intrinsically wearisome and distasteful; and when accompanied by long hours, harsh treatment and wages below the standard of comfortable family maintenance, actually dangerous to our social well-being.

We must remember also that in matters which concern the public welfare and with regard to the general policy to be followed by the nation all must have a voice, including now the women of the nation. It will be a mistake for those who direct industries or who take part in them in any capacity, to assume that the conscience of the nation will not assert itself upon the social effect of misdirected industrial activity.

It is not a misfortune that it does so, annoying as it sometimes appears to be, for it distinctly leads to the maintenance of a proper standard of civilized existence and also tends to protect the fair, enlightened and public-spirited employers from the unfair competition of the other kind of employer.

No one class may safely control in matters of public policy, but all may safely do so, and unless all do, there will be little chance of permanent stability or security.

Our Past-President, Doctor Hollis, has characterized different periods of history as:

> The 17th century, the period of exploration
> The 18th century, the beginning of religious tolerance
> The 19th century, the sowing of the seeds of democracy
> The 20th century, the period of crowded invention, and this
> The 21st century, the period of adjustment.

If this is correct, as I believe it is substantially, then I hope and believe that the influence of the engineer will be in the direction of constructive adjustments — such adjustments as will show recognition of the fact that there are both property rights and human rights, and that if human rights are, as Lincoln declared them to be, superior to property rights, then it is still to be remembered, nevertheless, that security of all genuine property rights is a necessary condition of human rights.

The improvement of the human race that is always going on takes many different directions and goes forward in many different fields. The engineer shares with his fellow human beings the results of the efforts of others, and his own peculiar contribution to the cause of human progress must always continue to be an increasing power to control the forces and modify the materials of Nature for the benefit of mankind. A very important part of this work is what we term the problem of industrial relations, which, it is certain, can never be satisfactorily solved except by the methods of the engineer, and it is equally certain that it can never be settled until settled right; that is to say, not until all concerned — the employed, the employer and the public — are convinced that substantial justice has been secured and is being maintained.

Of course there are and always have been employers who have been fair toward their employees — have been real leaders of men, able to arouse and maintain enthusiastic coöperation. Notable successes have been founded mainly upon this human ability or quality. In too many cases, however, the most profound thought that seems to have been applied to an industrial problem has led to the conclusion that industrial management consists in hiring as cheaply as possible and driving as hard as possible.

The day for that sort of thing is passing and industry generally is beginning to be conducted upon a much higher plane of intelligence. It is being recognized that there is a science of industrial management. Engineers have, so far, developed and formulated it; they must go on with it and conduct the country's industries n accordance with it, recognizing that the old order has passed away. Management of an industrial group is not a matter of brute force, but of intelligent skill, fairly and sympathetically applied with a view to getting the best possible results, not only for the employer, but for the workers and for the public as well.

If research is important in physics and in chemistry it is at least equally important in the domain of industrial science, and when fundamental facts or laws have been made known by such research we must face them; not to do so may mean disaster.

In general, the engineer bases his opinion and his acts upon definitely ascertained and carefully studied facts. We must do the same in our industrial-management problems, and when we do, most of our industrial difficulties disappear.

I have myself known of a large factory, believed by its owners to be well managed, but in which among the employees there was much unrest, much distrust of the management, general bad feeling and lost motion, to be transformed completely under the influence of new management with an enlightened policy and a plan which included no spies nor underhanded methods of any kind, but instead, methods of correcting administrative faults, for helping every employee to be successful with his work and to earn as well as receive high wages.

The better spirit that pervaded the place was remarkable and was reflected not only in greatly improved human relations, but also in greatly reduced labor costs and in the total cost of goods produced, while at the same time the employees earned substantially more money, usually without greater effort than before and sometimes with less effort than before.

We have been hearing a great deal about a falling off in production per man-hour, and there has been a good deal of that. But the fact is that too many of our industrial establishments have been conducted upon the hire, drive, and "fire" plan. I am speaking now of industries in general and of all kinds, not solely of the engineering industries with which we are best acquainted.

In ordinary times when there are more men looking for jobs than there are jobs, that method will answer, though it is never a good method. Men will stand for more or less harsh and unsympathetic treatment when not to do so may mean deprivation for themselves and those dependent upon them.

But in times such as those we have been passing through that method breaks down and the difference between real leaders of men and mere ignorant drivers becomes apparent. There are establishments in which the output per man-hour has not decreased and some in which it has been all along greater than in the prewar period. Some have of course been favored, others hampered by exceptional circumstances, and the necessity of replacing with new help those who were called for military duty tended strongly to reduce efficiency, in some industries much more than in others. But in general, those who suffered least in that respect were those whose policy was and had been an enlightened and intelligent one rather than the opposite, as indeed we should expect would be the case.

CONTACT BETWEEN EMPLOYER AND EMPLOYEE

Too often "results" are demanded by financiers and others who know little or nothing of industrial matters — who are far distant from the locality where the work of production is carried on and ask no questions as to methods, or whether of two local executives producing equal results so far as the books show, one has built up confidence and good will while the other has bred distrust and hatred among the body of employees.

Unfortunately morale and esprit de corps are not itemized on balance sheets, and some of those who are at the head of large manufacturing concerns know little or nothing about manufacturing except balance sheets prepared by men who also know little or nothing of management science and do not suspect that there is any such thing.

The growing number of engineers who stand high in the councils of large manufacturing concerns gives promise of better things in this respect.

In the older and simpler times when the workers very generally came into direct contact with those who were to use what the worker produced, the workers could not forget that they were rendering service to the public; nor could the public forget or overlook that fact. While those who work are today just as truly rendering service to the public, we have between the worker and the consumer whom he serves, a retail merchant, a jobber or wholesaler, the directors of transportation systems, a board of directors with its officers, a superintendent and a foreman.

The foreman is usually the only link in this long chain that the worker knows much about. This would seem to make plain the fact, repeatedly proved by experience, that our foremen should be provided with the machinery, so to speak, of true leadership and then carefully and systematically selected and trained for such leadership. That is a part of the work of the industrial engineer.

It is becoming generally recognized that, whether or not the head of a large industrial corporation will upon request confer with his employees, any group of them or any freely chosen representative of them about conditions of employment and of living is not a question between him and his employees solely, but that the general public may be and often will be interested in the matter. A corporation is created and exists by authority of the people; they may depend upon it for imperative necessities of modern civilized existence; they are responsible for the maintenance of law and order in times of industrial warfare and must usually contribute to the support of workers and others whose means of livelihood are suspended or destroyed by these upheavals. Also they have in most cases provided by means of protective tariffs for the maintenance of prices for the corporation's products higher than the free working of economic laws would permit. The public is thus, in a vital sense, a partner in interest in much of what the corporation does or does not do and we must face the fact that this interest will be increasingly insisted upon unless the industries themselves overcome in a fair and equitable manner the difficulties that now confront them. In accomplishing this means must be found for the coöperation of employer and employee.

The best minds seem to be agreed that this cannot be done by force or suppression, but must be done by intelligent and broad-minded study of what industrial conditions really are and how to improve them; especially how to secure and maintain a spirit of coöperation and of enthusiasm in work, without which no great thing can be accomplished.

As our modern industrial life develops it becomes more and more clear that the broader interests of all of us, whether humble employee or mighty head of an industrial organization, are interwoven and inseparable. Formerly it seemed clear that the public was interested in industrial strife only where public service functions might be disturbed or suspended; but events have made it clear that, from the standpoint of public health, public education, service to the country in time of war or other emergencies, and of the general well-being of society, present and future, industrial strife concerns everybody; and there is a growing public sentiment according to which any industrial executive who refuses to confer with his employees, with their freely chosen representative, or with the representatives of any group of them, about the conditions of employment, is assuming more responsibility than can safely be left to him. And we may be sure that it will in the end be much better for all of us to respect that public sentiment voluntarily. If we do not, then we must consider the possibility, if not the probability, that legislation as an expression of this public sentiment may follow.

Such a conference, properly and fairly conducted, will at least enable each side to know just where the other stands and why. Often this is sufficient to point out a clear way to a satisfactory adjustment. And a conference about business matters of mutual interest is not the handing over of one's business to employees or their representatives. Even if no concessions are or can be made, such a conference is almost sure to promote mutual understanding and good feeling; and I speak from personal knowledge and experience when I say that it may and often has prevented what might have been serious disturbances.

Such conferences however, in order to be effective for good, must be as friendly and courteous as other ordinary business conferences between men whose duty it is to coöperate for a common purpose and for mutual advantage; men who, in other words, desire to, or must, maintain a decent friendly relation, not an antagonistic or a patriarchal relation.

If it be understood that any matter which any employee, or any group of employees, wishes to have adjusted can be brought to the attention of the management, either by an individual, by a committee of employees freely chosen by the men for that purpose, or by any other agency whatever that seems advisable to the workers themselves, acting freely and without coercion, and that such a matter will be regarded as a legitimate business affair and especially that no prejudice will arise against any member of such committee for having thus served his fellow-employees, very much misunderstanding will be avoided. If this plan is followed there will be no need for spies among the employees, spies being themselves not preventives of trouble and misunderstanding but promoters of them, often deliberately and intentionally so.

If the conditions of employment are fair and right, if there are few or no substantial reasons for dissatisfaction, if the channels of communication between owner and workers are kept free and open, agitators and trouble makers will seldom if ever make much headway.

Sometimes, however, under present conditions, a strike may be unavoidable. When it takes place if the management will make it plain to the men that they are not blamed for having the ambition to improve their condition; if there are no bitter denunciations; if there are no newspaper interviews condemning the men as being mentally incompetent or intentionally unfair and unreasonable; if it is regarded simply as a human error, or as a fever which will soon pass away; and that when it does, such of the strikers as have been decent and orderly and can be reëmployed will be welcomed back, singly or in groups; if foremen and superintendents who have known the men they now find acting as pickets and have been in the habit of greeting in friendliness, will continue to do so; a satisfactory settlement will be much more easily, more quickly, and more surely reached than by the usual plan of recrimination and bitter denunciation.

This more advanced attitude must not be assumed for the purpose, however, but must be the reflection of a genuine feeling of sympathy and of tolerance. It is human nature for a man to be much more influenced by the opinion and advice of one who is friendly and sympathetic than of one who is contemptuous, harsh, or filled with hatred.

In commending this plan of procedure, which I realize may seem to many employers impracticable or impossible, I speak, nevertheless, upon the basis of experience, not only of my own, but of that of others, in large as well as smaller organizations, and it is not advanced as being in the interest of employees only, but of employers as well.

FOREMEN TRAINED FOR LEADERSHIP

Particularly do foremen usually need to be trained, or educated, if you please, to know how to treat men in order to get their enthusiastic coöperation and their best efforts. The foreman being ordinarily the only man representing an employer with whom the worker comes into contact, the "local color," so to speak, of the shop, as seen by the worker is given by the foreman, and many an employer suffers, in reputation and in pocket, by reason of a foreman who may be obsequious and ingratiating in the employer's office, but brutal and inconsiderate in the shop. Cases are on record, for instance, in which the good intentions of an employer in providing rest rooms for women employees were, unknown to the employer, set at naught by a foreman who cared nothing for rest rooms, but believed in driving women so long as they could stand at their machines.

Too little attention has been given to this and foremen need to be trained for leadership and led to understand that most workmen, if properly handled, are as ambitious to "make good" as they are themselves, and that by sympathetic leadership and teaching where needed, excellent workmen can often be developed from what at first may seem like very unpromising material. This is not mere theory or idealism, but is based upon wide observation and a very considerable personal experience in establishments run upon all sorts of plans, from very good to very bad.

Difficulties with men are often due to causes that are obscure. A group of workmen who resisted a perfectly fair and mutually advantageous proposition pertaining to their work, were not understood until it was finally discovered that a thoughtless and ruthless "comptroller" recently placed in power, had, merely for his own convenience, arbitrarily made a change in their pay day which had seriously inconvenienced them. He had no more thought of consulting them about it or consulting the superintendent under whom they worked, than as though they were so many soldiers compelled to obey without question any regulations made concerning them. By this they were for the time being convinced that the company cared nothing for their interests and, reciprocally, they cared nothing for the company's interests and were suspicious of any proposition made to them.

I am not, of course, to be understood as saying that they were right in that, but we are dealing with human nature and must take men as they are and make the best of them. Workmen are, after all, about the same as other people, will respond to given treatment in about the same way, and act about the same under given conditions. When we fully realize this we shall be on the right road to an understanding of them and of how to get along with them.

Successful industrial management has been declared to be, more than anything else, a "state of mind." The various systems we are now hearing so much about vary greatly in their capacity to help in arriving at good conditions, but under the best of them there must be a foundation consisting of a determination to be fair, considerate and entirely candid and aboveboard, or they will be of little use.

If I seem to stress unduly the duties and responsibilities of engineers and employers in these troublous times, it is of course not because I believe the workers to be faultless, but in the daily and in the trade press their shortcomings have been abundantly set forth and the present pressing problem is as to what the engineer and the employer may and ought to do to bring about a better understanding. That can be done, but not by dwelling upon old grievances or by calling hard names. There must be mutual forbearance, mutual understanding, and a general recognition of the fact that men who are to be of any use to society or to the industries must have their personal ambitions, and in order to have them must see some chance of attaining them.

We are hearing constantly more about service as constituting the only just claim to rewards. Certainly the engineer need not fear comparison with others on that score. Yet there are those who, with the best intentions I am sure, charge a large share of our industrial and social difficulties to features of modern industry that have been created and are maintained by the work of the engineer.

Especially do they charge that doing things by machinery instead of by hand, and the multiplication of large manufacturing establishments in which the work is minutely divided, have had a bad effect — have, indeed, made men and women slaves of the machine.

I think we may claim that there is a misapprehension about this and that machinery and large industrial establishments do not nor can they by themselves enslave or oppress human beings.

It is easily demonstrated by reason and by human experience that division of labor, machinery for increasing man's productive capacity and the use of capital in production all tend, by themselves considered, to help the worker — to release him from burdensome tasks and from the necessity for working too hard or too many hours per day for too little money.

TWO KINDS OF MONOPOLY

But certain monopolies may and do have the opposite and injurious effects referred to; especially monopoly of the earth's resources and the holding of such resources idle and beyond the reach of labor and capital that might otherwise be applied to them for the satisfaction of the wants of mankind.

And our tax system acts as though it were especially designed to promote this result, to discourage industry of all kinds and to encourage the holding idle of the earth's resources for higher prices to be brought about by pressure of population and the enterprise and labor of others.

Many a manufacturer has been hampered in making enlargements of his plant by the high prices of vacant and idle land needed for that purpose; said high prices having been actually created by the activities of himself and his employees and in no degree by those who had been holding the land idle.

Fourteen years ago our Society held its Spring Meeting in Chattanooga and visited a water-power development in the Tennessee River, near there. At that time the large dam was about half-finished and I was told by a prominent resident member of this Society, himself a manufacturer, that already in anticipation of the coming cheap power, the owners of factory sites on which it might be utilized had doubled or trebled the prices at which they were holding them.

We hear much of the blessings to humanity of the cheap water power that may be developed in large quantities from our streams. As a matter of fact, it will be found that in nearly every case where water-power rates are substantially less than the cost of steam power the total cost of the water power is nevertheless about the same as for steam; for the owner of available land on which to build factories and who, as such, has rendered no service of any kind either in producing power or in any other way, pockets the difference in prime cost — not the engineer, the manufacturer, nor the consumer of the goods produced.

Moreover, while as a general thing the manufacturer whose efforts and enterprise build up these values is taxed heavily for doing it, the vacant-land owner escapes with nominal taxation on the ground that his property is non-productive, if you please.

I mention these things, not because it is the engineer's special province to deal with them, but it is at least as much his business as that of any other citizen and it has a direct and vital effect upon his work and his opportunities for employment, or for going into business.

If it is the engineer's business to render service, then it is equally his duty to see to it that so far as possible all others do the same, and he has only to look about him to see that very many do not render service, yet are "clothed in purple and fine linen." The cure is not to take away their purple and fine linen, but simply to see to it that their having it is not by reason of their being able to restrict the opportunities of others for access to Nature's storehouse of raw materials and to work for the general welfare.

It is to be hoped that the day will soon come when every able-bodied and able-minded man will find it necessary to render actual service for what he receives and in fair proportion to what he receives. When that has been accomplished we need not concern ourselves overmuch about the way in which he spends his money. It will be clearly and of right his money, and the opportunities for spending it in such a way as to injure society will be much less than now.

That engineers are giving much more thought to these matters than formerly and are perceiving their important bearing upon the work and life of the engineer and all workers, is indicated by the fact that the President of the American Institute of Electrical Engineers, in his presidential address last year, discussed them, and the President of the American Society of Civil Engineers, the well-known Chief of the Reclamation Service, in his presidential address also discussed them; and we three, at least, are in agreement concerning them and their commanding importance, particularly in their effect upon the work of the engineer in whatever field of endeavor he may be engaged.

Of course an engineer or a manufacturer may be also the possessor of special privilege or a monopoly of one kind or another but comparatively few of them really are, and where an engineer has acquired enough property to be called even moderately well off in these days of colossal fortunes, it will be found that his success in this is based upon the value of service which he has contributed to the general welfare, and those who envy such a man or who would limit his activities or his prosperity are very few and have practically no effective influence.

We can think of some engineers who have acquired what, until recently at least, would have been considered wealth. But they have usually done this in strictly competitive lines of business, engaged in by others, and open to all comers upon equal terms.

Compare the case of Corliss, for instance, and his early practice of selling his new improved engine upon the basis of a stated share of the saving in fuel for a limited time, with those usually absentee owners of the anthracite coal fields of Pennsylvania who create no coal, dig nor transport any coal, do nothing about coal except to own the ground in which it was placed by Nature long ages ago "for the benefit of mankind," as we are fond of saying. Some of these owners are reported to have been recently doubling and more than doubling the royalties they collect for the mere privilege of employing engineers and miners and using machinery in the production of coal; the state of Pennsylvania helping them to retain their grip upon the coal supply by taxing their as yet unused coal lands at ridiculously low valuations as "unproductive farm lands," if you please; while real farmers in that state are taxed heavily not only upon their lands but upon everything else that can be found in their possession.

Herein is the backbone of the anthracite monopoly which is a grievous burden upon industries and upon coal users, and obstructs not only the production of coal but of all manufactured things in the production of which coal is a prime necessity. Being a citizen and a taxpayer in the state of Pennsylvania gives me the right, if nothing else does, to criticise its unenlightened and grossly unjust taxing system.

There are those, however, who will say that Corliss made his money by the possession of a monopoly, because his engine improvements were patented. But engineers especially need to clearly perceive the vital difference between such a monopoly as Corliss enjoyed and those monopolies of the earth's resources to which I have referred. Corliss had, for a limited time only, a monopoly of that which he had himself created; and any one who chose to do so went on using the plain slide-valve engine after the Corliss appeared, and all surely would have done so had not the terms offered by Corliss for the use of his improved engine been advantageous for the purchaser as well as for Corliss. In other words, the benefits arising from the improvements of Corliss were shared, first, by the purchaser of the engine, then by users of power, and finally by everybody.

Nearly always the same is true of any invention. It will not come into use unless purchasers of it share the benefits of what, before the invention, did not exist. Have we equal freedom of choice in the matter of fuel, not invented by man but created by Nature and an indispensable necessity of modern civilized existence?

Some thirty years ago a member of this Society invented a steam-engine governor from which he received a fortune in royalties based upon a certain charge per horsepower of engines to which, it was applied. It is safe to say that that governor was never applied to one engine without benefiting both the engine builder and its purchaser by an amount far exceeding what was paid for its use. We are playing directly into the hands of those who advocate the socialization of all property and of all human activities when we for one moment admit that the inventor of any such thing is to be classed with those who monopolize that which is the common heritage of all of us — that which no man created and without which human existence cannot be maintained.

It would of course be wrong to condemn individuals who simply take advantage of a system that has long been tolerated by their fellow-citizens, and is generally looked upon as respectable and proper. It is not individuals but the system that is wrong, and wrong because it permits and even entices men to take steps to secure large rewards without doing anything of real value in return for them; particularly to make themselves a heavy burden upon the industries which the engineer is trying to develop and upon which he depends for his opportunities to render service.

All clear thinkers, I believe, support and will maintain the just rights of property; but it is becoming clearer every day that a necessary condition for safe-guarding property rights is, first and above all, the safe-guarding of human rights. In fact, security of property rights rests, finally, upon the maintenance of human rights.

It is not true that "the world owes every man a living," but the world does owe every man an exactly equal opportunity to earn a living.

And let us not be deceived by the newly invented factor of production called "ability," which, it is asserted, rests somehow upon an economic basis different from that of labor. Labor, in the broad sense — the economic sense — is any kind of useful endeavor, mental or manual. There are only three factors in production of wealth: land or Nature's resources; labor, mental or physical; and capital. The invention of so-called "ability" as a fourth factor has no basis in reason or experience and its only function is to confuse our reasoning about industrial and economic problems and to bring us to erroneous conclusions concerning them. Especially is it being invoked in attempts to justify exorbitant gains by those who have rendered no service in return and are often mere gamblers. A burglar or a pickpocket may exercise ability in his profession, and even the right kind of ability is of no use unless exercised, and then it is labor.

If it is true that any considerable portion of our American workers are being led astray by ultra-radical teachings, then such teachings may be most effectively combated by such teaching as the engineer is in a position to give. But it will not answer to simply assert that there is nothing the matter and that we should all be content with things as they are, or as they were in a time now past and gone. There must be, in such teaching, evidence of a real understanding of what conditions really are and of a sympathetic attitude toward any reasonable and fair ambition to improve them.

It must be shown, as it easily can be, that profits, in normal times and in competitive lines of business and manufacturing, are usually and on the average not greater than they need to be to attract to them capital and men capable of carrying them on; and that business men and manufacturers as well as workers, often suffer from the power of special privilege and inadequately regulated monopolies.

The fact is that our economic and social science has not kept pace with the work of the engineer; for he, in his own work, is by temperament and training an innovator. He lives to persuade other people to do new things, or to do old things in entirely new ways. This being so, it would seem to be logical that he himself should at least maintain a sufficiently open mind to avoid resisting a new thing concerning which he has not cared to, or has had no opportunity to, investigate.

Whether it be a new idea in industrial management; revision of our tax system so as to remove burdens from beneficial activities and to discourage monopolies; a League of Nations to preserve the peace of the world, so that the results of the work of the world can be wholly useful instead of largely wasteful, destructive and pernicious; we engineers, especially, ought not to remain bound by mere custom, which, as Guizot said:

" . . . contracts our ideas with the circle it has traced for us; it governs by the terror it inspires for any new and untried condition; it makes us believe the walls of the prison within which we are inclosed to be the boundary of the world, and beyond all is undefined, confusion, choas, where, it makes us feel, we should not have air to breathe."

Customs and usages have their places and their values, but when new problems present themselves we must not shrink from the contemplation of what are, or may seem to be, new remedies, so long as they are founded upon established principles of justice and fair dealing.

No. 1756

JOHN ALFRED BRASHEAR

HUMANITARIAN AND MAN OF SCIENCE

By Henry S. Pritchett,[1] New York, N. Y.

Non-Member

I AM asked to speak before this body of trained engineers touching the scientific work of John Alfred Brashear. For nearly forty years I had the pleasure of his friendship. It is not easy under such circumstances to differentiate between those qualities in a man's life that pertain to his science or to his profession and those deeper and simpler relations that belong to friendship. My purpose in speaking before this body of scientific men is not so much to tell the story of his life as to call attention to his contributions to optical science and research and his relation to some of those great forces in our social order which went to the making of a character like Brashear.

Born at a period of profound national peace he lived to see the end of a world war of unexampled fierceness and magnitude. The foundations of the social order have been shaken by the violence of these forces. Our own country with all its advantages of personal freedom and of equal opportunity has been drawn into these currents. Today it is interested not so much in the personal story of a man's life, however fine and fruitful it has been, but rather in the question of how his personality arose out of the social and intellectual forces that operate in our national life. How did such a character come to be? What was there in the movements of his time and of his environment to produce a man of his quality and in what way do the forces of the social order operate today in the growth, development, and perfection of a human character such as was John Brashear's?

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An address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

He was born almost exactly eighty years ago in what was then the little village of Brownsville, about thirty miles south of Pittsburgh. In the same village was born ten years earlier James G. Blaine, afterwards secretary of state; and a dozen years later Philander Knox, likewise to become a secretary of state. His father was a mechanic and the boy grew up under surroundings that intensified his natural aptitude for machinery. His mother's father was a watchmaker. From him the boy acquired not only his taste for the technique of fine tools but he also imbibed from him early in life that love of astronomy which was perhaps his ruling passion.

Brashear's formal education was obtained in the public school of Brownsville which he attended until he was about thirteen years old, when it became necessary for him to earn his own living. Accordingly he came to Pittsburgh, learned the patternmaker's trade, and sought employment in one of the steel mills where he remained for twenty years, developing into an expert mechanic and becoming, toward the end of this period, the superintendent of a rolling mill.

During this period of his life as a practical mill mechanic, he was under two strong influences, the one appealing to the spiritual side of his nature, the other to his intellectual side. Having had from early life a deep devotion to humanity, he became impressed with the notion that it was his duty to become a preacher of the gospel. He was a devout member of the Methodist Church and having a facility in public speaking he came to take an increasing part in the religious exercises of his denomination until finally he came to believe that his path of duty lay in the ministry of the Methodist Church.

Side by side with this absorbing and insistent motive, there grew up in his life another equally insistent and equally influential source of action. Early in his mill days Brashear had acquired a small refracting telescope mounted on an ordinary tripod. With this modest instrument he observed the moon and the principal planets, some of the star clusters and nebulæ. Those views merely whetted his thirst and he began to long for something better than this small and inferior objective. Having no money there was but one way to get it and that was to make it himself; and so in the little shop at the back of his house, he and his wife began the grinding of a five-inch refractor. The story of this refractor has often been told. Brashear and his devoted wife worked for some three years before the lens was brought to anything like perfection. Then he took it to the one man in Pittsburgh whose sympathy and help was necessary for his own development. Brashear himself has told the story in very fitting words: "I first met Professor Langley," he writes in 1913, "nearly forty years ago when I visited the Allegheny Observatory and showed him the results of many years of labor by my wife and myself in the making of a lens. I remember how my knees trembled when I stood before him. As he gazed, as it seemed to me, into the inner recesses of that lens upon which we had spent so much thought and time and labor, he showed me how to proceed in order to improve the lens and then took me to view Saturn through the great telescope of the Observatory. I carried away recollections of that visit that are as fresh in my mind today as if it had happened yesterday. I did not think then that I should ever be closely associated with Professor Langley in his profound studies of astronomical science."

The lens which Brashear showed to Langley in 1875 was a five-inch refractor that had occupied all the spare time of himself and his wife for three years. At that time he was a man of about thirty-five years of age.

These two motives, a deep and fervent humanitarianism and a passionate devotion to science were to be the dominant influences of his life. He never gave himself wholly to either the one or the other. Always he was the humanitarian, and always he was the man of science, but at certain epochs he was preëminently the humanitarian and only in a secondary way a man of science; while at other times of his life he was essentially a man of science and secondly the humanitarian. Doubtless the impulse toward both of these lines of action arose from the same thing, — the possession of a keen and ardent imagination. We do not always realize that the vision of human destiny and human progress that brings men with complete devotion into the service of their fellowmen is closely akin to that other vision of the forces and operations of nature, whether exercised upon the molecule or upon the stars, which also brings men irresistibly into the service of knowledge. Imagination, keen, far-sighted, sympathetic, may equally well lead its possessor into the one path as into the other. With Brashear it was a question hard to decide whether he should travel his life journey by the one way or by the other. Out of this duality he was sometimes the humanitarian, sometimes the scientist, and always partly both.

It seemed at one time of his life that he had definitely turned toward the ministry. His intention to be a Methodist minister was publicly announced. He took part as a lay speaker in various gatherings of his church. On one or two occasions he undertook to conduct formal service, but in the end he convinced himself that he was a better mechanic than he was a preacher, that he could serve God better as a layman than as a minister, and in the meantime the lure of the telescope was drawing him more surely into the service of science. It needed only the association with Langley to make him a lifelong worshiper at the altar of science, equally inspired by the beauty of the planets and the perfection of a prism.

Brashear was always a little reticent as to the actual experiences which finally brought him to retire from the ministry. I suspect that his efforts as a preacher fell below his expectations and that the simple faith of his earlier years came in time to conflict in some degree with his widening scientific knowledge. I have heard a story told in his presence which he never denied and which may or may not have been true. It was to this effect. On one occasion when he had made a visit to a farmer in the neighborhood of Pittsburgh and had commended to the good farmer and his wife the consolations of religion, the latter called his attention to the parched condition of the ground and the fact that the cabbages, which formed the sole crop, were in danger of dying by reason of the drought. Before taking his departure, as was the custom, Brashear in his capacity of lay preacher held family prayers in the course of which he prayed fervently that the Lord would send rain to the aid of the cabbages about to perish. Having done his duty, he went peacefully on his way, and in a few hours the heavens opened and the rain descended in such torrents that it entirely swept away the cabbages. The unfortunate farmer and his wife complained afterwards that "These Methodists always overdid it." Whether this story is apochryphal or not, Brashear after a few such efforts decided that the life of a mechanic was better suited to him than that of a minister, and before he had finished the first lens all thought of taking up the formal ministry of his church had vanished, but the deep devotion to humanity that had drawn him toward the life of a preacher remained one of the dominant principles of his whole life.

Encouraged by Langley's praise of the five-inch lens, Brashear immediately began in 1875 a twelve-inch reflecting telescope which he completed in 1877 and which he used for some three years afterwards in a study of the lunar crater Plato, — seeking to determine whether any change had taken place since the carefully made sketches of Bayer and Mædler in 1838. During these years also Brashear became more and more the mechanical and optical assistant of Langley in his researches on radiant heat.

The latter part of his life Langley's name became identified in the public mind with the study of aeronautics, but in the days when Brashear first knew him as Director of the Allegheny Observatory he was engaged upon a most important study of the infra-red portions of the solar spectrum. In 1881 on his Mount Whitney expedition, working at an altitude of 12,000 feet, he discovered an entirely unsuspected extension of the invisible red portion of the spectrum which he called "the new spectrum." These researches, delicate in the extreme, were dependent absolutely on the prism and lenses used in the measurements. Brashear became Langley's coadjutor on the optical and mechanical side. It was a fortunate chance that brought Langley, the astronomer, and Brashear, the mechanician, to the solution of a common problem, a process that also knit two great souls in a lasting and devoted friendship.

To the men of our day, living amid the engrossing and insistent demands of civilization as it has developed in the last fifty years, it is not easy to picture the scientific enthusiasms that made possible in that day an astrononomical observatory in the City of Pittsburgh, and which furnished at the same time the inspiration of a technician like Brashear and of an astronomer like Langley. The rise of the Allegheny Observatory is a part of the romance of the astronomy of the middle half of the nineteenth century. One can scarcely imagine the Pittsburgh of today alive with an enthusiasm for astronomy, its people attending in great numbers astronomical lectures, and talking and speculating in every moment of leisure over the wonders of the heavens, debating whether the moon and planets were inhabited, and anticipating with eagerness the discoveries which the telescope was expected to bring forth. Between 1840 and 1860 there was a popular passion for astronomy that was wide-spread, which took hold upon the imagination of the people of the United States as perhaps no other science has ever done. It led to the formation of astronomical societies and to the building of many telescopes, a large number of which, alas, are today idle.

This popular enthusiasm for astronomy in the United States had its origin in two things. The improvement of the telescope and of the optical means for the manufacture of telescopic lenses and mirrors had resulted in the interesting discoveries of Herschel in the invention of the spectroscope and had aroused the general expectation that the telescope would soon reveal new wonders of the heavens to the eyes of men. Furthermore, in that day most of the information in regard to science was disseminated through the medium of public lectures. The all-pervading newspaper had not come to dull the thirst for knowledge and to satiate the curiosity for news.

The lecturer on astronomy through the eastern and central states was able to appeal to the imaginations of whole communities. Of these perhaps the most influentital was Mitchell, director and founder of the Cincinnati Observatory, author of the popular works, The Planets and Stellar Worlds, and The Orbs of Heaven. Mitchell had a remarkable life, which began as a clerk. He obtained an appointment to West Point. Subsequently he resigned from the army and was called to the bar, but forsook the law and became the director of the Cincinnati Observatory. Later on he was director of the Dudley Observatory at Albany, but his great influence lay in the eloquent and attractive way in which he described the wonders revealed by the telescope. Popular imagination was fired by the notion that the world was upon the verge of great discoveries which might perhaps lead to a knowledge of the inhabitants of other worlds. In that day views concerning the planets were strongly tinctured with the hope, and oftentimes with the belief, that ultimately through the telescope we might learn to know a universe peopled with creatures like ourselves. The popular thought of that day pictured our universe as created for the sole use of men. The notion of planets rolling through space age after age uninhabitated by human beings was a distinctly repugnant conception to the audiences of the period between 1840 and 1860. Long and serious discussions were held as to whether the death of Christ served as atonement for the inhabitants of Mars as well as for the inhabitants of the United States. The imagination of intelligent men was fired by the notion that the world was about to realize great and far-reaching discoveries in the heavens, and the eloquent pictures of a man like Mitchell produced upon his audiences impressions which in the fleeting and confused life of our day are apt to be effaced by the next morning's paper.

This popular interest in the science of astronomy was greatly quickened by the appearance of Donati's comet in 1858. This was one of the most glorious heavenly visitors that have ever come within human ken and the drawings made of it by the Bonds of the Cambridge Observatory remain even in these days of photography wonderful pictures of this heavenly visitor. At its brightest the comet reached from the zenith to the horizon and the nations of the world gazed at it with awe and admiration.

The establishment of the Allegheny Observatory was a part of the romance of this movement. In February 1859 in the glow of the enthusiasm created by Donati's comet, a few citizens came together to consider the purchase of a telescope which in their own language "should have the magnifying power which would bring the heavenly bodies near enough to be viewed with greater interest and satisfaction." The association was called in its earlier days the Allegheny Telescope Association, but it did not attain its object until after the expiration of ten years when it found itself in the possession of a telescope of thirteen inches aperture, at that time an instrument of extraordinary power. The Association was a stock company but the stockholders voted to convey their interests in the telescope to the University, retaining the privilege of making occasional use of the instrument. The first director, Professor Philotus Dean, who served without salary, began his labors in the latter part of the year 1863, and was succeeded four years later by Professor Samuel Pierpont Langley, whose work was destined to bring to the Observatory lasting fame, although its earlier years were marked by a series of happenings worthy of the troubled days through which the country was passing. Among other adventures was the theft of the object glass of the telescope which was held by thieves for ransom at an enormously exaggerated value. The trustees, like the American Congress confronting the Barbary pirates, were ready to spend millions for defense but not one cent for tribute. The negotiations which finally led to the return of the lens were conducted by Professor Langley. In his final interview with the bandit who stole the lens, the matter was put upon a high plane. "You," said the thief to Professor Langley, "are a gentleman, and I am a gentleman; we must trust one another." Starting upon this comfortable basis, it goes without saying that the lens was finally returned uninjured and without tribute money.

I have dwelt with some length on the astronomical enthusiasm of the period between 1840 and 1860 and of the founding of the Allegheny Observatory because in this movement and in the founding of this observatory lay the forces which transformed John Brashear from a skilled mechanic into a maker of great lenses and the perfector of the most delicate instruments of spectrum analysis. The immediate causes which brought him into this life lay in the enthusiasm for astronomy, the result of the eloquent words of Mitchell and others. His opportunity came in the acquaintance of Professor Langley, an acquaintance which could not have happened save through the existence of the Allegheny Observatory. Both Brashear and Langley were the products of forces of which perhaps neither was conscious but which profoundly affected the public opinion of their day.

When Brashear brought to Langley in 1875 his little five-inch lens, Langley was engaged in a research into the problem of the selective absorption of radiant heat. Astronomy at that day was just beginning to turn from the old astronomy of position, which concerned itself with the right ascension and declination of stars, their proper motion, and their distances, to the study of the physical nature of the sun and of the stars made possible by the invention of the spectroscope and the gradual advance of photographic art. To carry on such researches as Langley was engaged in, there was needed not only a telescope but also the most refined and delicate mechanical instruments. The telescopic lens was only a single element in the apparatus for stellar and solar investigations. More important were the delicate and refined instruments which should sift out the rays of different wave lengths. To carry on such researches there was needed a combination of astronomer and mechanician. Stellar physics could only be advanced by such a coöperation. When Langley gave his kindly welcome to the mechanician Brashear with his imperfect lens, he had little reason to suspect that he was to find in him a coadjutor whose coöperation was to make his work glorious in the annals of astrophysics. Brashear became his indispensable aid.

Between 1875 and 1880 he had made a number of pieces of apparatus for Professor Langley to be used in connection with his study of radiant heat. By that time he had become so interested in these problems that he gave up his work in the steel mill and began making telescopic lenses and mirrors as well as other optical apparatus in his own shop established in 1880.

His first great success was attained in the silvering of the mirrors for Professor Langley's Mount Whitney expedition in 1881. Out of this grew Professor Langley's famous instrument the bolometer for the measurement of the direction and intensity of the most delicate heat rays. So delicate is this instrument that the heat radiated from a human hand at the distance of sixty feet can be detected in the readings of the instrument.

His next great mechanical achievement in Langley's researches was the invention of a method for polishing rock-salt prisms. Rock-salt crystallizes in cubes. It has the remarkable property of transmitting heat rays and was therefore specially adapted for Langley's researches in radiant heat. On account, however, of its softness and its dull color it is difficult to obtain prisms or lenses that will take a high polish. Brashear overcame these difficulties and produced rock-salt prisms and lenses which were not only used in Langley's researches but were immediately adopted by European astrophysicists. His description of these methods was given in a paper read before the American Association for the Advancement of Science in 1885.

About this time also he became interested in the work of Professor Rowland of Johns Hopkins. The spectrum studied in the spectroscope is produced by a dispersion of rays of light to passing through a series of prisms due to the varying refraction of light of different wave lengths. Soon after the discovery of the spectroscope and the adoption of the undulatory theory of light, it was found that a spectrum resembling the prismatic spectrum could be produced by the diffraction of light waves from a ruled surface, the difference in the light phase being proportional to the number of rulings. The spectra obtained in this way with sharp diffraction bands or shadows which characterize them when the lines were parallel afforded great advantages over the ordinary prismatic spectrum in determining the wave length in different parts of the spectrum. Professor Rowland in the effort to perfect a machine capable of making a great number of lines on a hard surface so as to obtain gratings suitable for determination of the lengths of the light waves invoked Brashear's assistance in making the plate. After a year of hard work he succeeded in overcoming the difficulties and Professor Rowland was able to rule gratings on the surface containing many thousand lines to the inch. The gratings from Rowland's engine are still considered the best in America or Europe although the researches of Michelson and others, have shown that for the determination of absolute wave lengths the gratings will in time be displaced by the so-called "interference" methods.

In 1885 Brashear completed his first large spectroscope and in 1887 he made what was then the greatest spectroscope of the world for the Lick Observatory, with which Professor Keeler made his famous investigations on the motion of the nebulæ in the line of sight. Other great spectroscopes followed in the years immediately succeeding. About this time also Brashear became interested in the work of Professor Michelson on the determination of the velocity of light and also on the measurement of velocities as existing between long and short waves. His aid was enlisted in the first refractometer which was used to measure the meter of the International Bureau in terms of light waves. He succeeded in reducing the limiting error of the optical surface to less than 5/100 of the wave length, a marvellous achievement in optical construction.

In the early nineties Brashear constructed for Professor George Hale the first spectro-heliograph for the automatic photography of the surface and surroundings of the sun. This work has been epoch making in the realm of solar photography.

Meantime in his optical works Brashear was constructing objectives and mirrors for reflecting telescopes and for refractors of larger and larger aperture. Among the largest of these are the 16-inch refractor of the Carleton Observatory, the 18-inch refractor for the Lowell Observatory, and finally the 30-inch Cassegrain for the Allegheny Observatory, completed some ten or twelve years ago, and the 72-inch reflector for the Dominion Observatory at Victoria. He succeeded the Clarks as the great lens maker of America, but unlike them his marvellous mechanical knowledge and skill extended to almost every field of solar investigation. As the Clarks were the representatives of the old astronomy in the manufacture of great refractors, so Brashear became the representative of the new astronomy in the realm of mechanical and technical advancement. There have been in America three manufacturing firms who have had a notable part in the advancement of astronomy and stellar physics — Alvan Clark and Sons of Cambridgeport, the makers of the great refractors; Brashear and McDowell of Pittsburgh, also makers of lenses and of mirrors, but preëminent in the making of delicate prisms and tools needed in the new astronomy, and, finally, Warner and Swasey of Cleveland, the engineers of both the old and the new astronomy, whose designs and constructions have made possible the telescope tubes and the great installations necessary to operate the modern telescope with the spectroscope and photographic attachments. I have sometimes wondered whether there was any significance in the fact that none of them came out of a college.

Among Brashear's many achievements in aid of astronomy was the manufacture of photographic doublets designed to photograph in a single exposure a wide field. In this he had the coöperation and advice of his colleague Dr. Charles Hastings of Yale. Brashear always insisted that a large share of the success of his optical shop was due to the investigations of Dr. Hastings and to the patient and skilful work of his associate Mr. James McDowell.

One of the most ingenious of these instruments were the great cameras made for Dr. Max Wolf of Heidelberg which were used for the photographic discovery of asteroids. The world has almost forgotten the asteroids, but in the middle of the last century they used to be on the front page of the newspapers. In the early part of the nineteenth century the civilized world was thrilled by the discovery of a little planet between the orbits of Mars and Jupiter. The space between these two orbits had seemed to astronomers for many years an inexplicable gap. The distances of the planets from the sun had shown a remarkable adherence to an empirical formula known as Bode's law. Bode's law was obtained in this way: If you write down the numbers

0 3 6 12 24 48 96

which it will be noted are in arithmetic progression after the second number, and if we add 4 to each one of them, we have another series viz.:

4 7 10 16 28 52 100.

Now the terms of this series represent very closely the distances from the sun of the planets known at the time the law was announced, the planets being Mercury, Venus, the Earth, Mars, Jupiter and Saturn but there was a blank between the orbits of Mars and Jupiter corresponding to the number 28. When Uranus was discovered in 1781, and it was found that the distance of the new planet was precisely represented by an eighth term, a strong hope was created that a planet would be discovered between the orbits of Mars and Jupiter. When, therefore, in January, 1801, Piazzi at Palermo discovered a planet afterwards known as Ceres, occupying precisely the vacant number 28, it was hailed as absolute proof of the truth of this empirical law. A little more than a year later however when the astronomer Olbers discovered a second planet, which received the name Pallas, the theory was badly shaken, but Olbers suggested an ingenious explanation which for the moment satisfied the doubters. He argued that the two new bodies were fragments of a single body which had been torn asunder by some internal convulsion. Now the laws of mechanics show that after any such castrophe the fragments in whatsoever direction they might be turned must pass, on each of their revolutions around the sun, through the point in which the catastrophe took place. The orbits of Pallas and Ceres very nearly filled this condition and the third planet Juno, discovered soon after, seemed to confirm this view.

In the days between 1840 and 1860 when Brashear received those deep impressions which centered his interest upon astronomy, asteroid hunting was a favorite sport of the astronomer, and the discovery of an asteroid brought fame to the finder. What safer title to immortality could a man have than a planet of his own circling century after century through this great vacant space of the solar system? Two American astronomers were famous catchers of asteroids, — Professor Watson of Ann Arbor, and Dr. Peters of the Hamilton College Observatory. Professor Watson discovered over twenty of these little planets and Dr. Peters more than forty. He had the rare distinction of discovering two new planets on the same night.

By the '80s the number of these asteroids had grown to well over a hundred and the computing of their orbits had become a matter of enormous labor. It was the custom of the young astronomer of that day to take on one of these newly found planetoids, compute its orbit and follow it for the first few returns until the orbit was brought to a sufficient state of accuracy so that the planet could be identified for many years to come. I remember in the early days of my own study of applying to Dr. Peters for one of his newly discovered planets and receiving from him the naive reply that he had given that particular planet away but that he would be happy to discover another for me within a short time, a promise which he very easily kept.

In that day the discovery and the pursuit of these little planetoids was purely the work of the mathematical astronomer searching with the ordinary refractor. It would have been a bold prophet who could have foretold fifty years ago that a highly trained mechanician and technologist like Brashear would have anything to do in the discovery of these little planets between Mars and Jupiter, but the wonderful photographic cameras which he made for Dr. Wolf of Heidelberg were so effective that they quickly superseded all other means of asteroid hunting. Within a short time Wolf had gathered in more than a hundred new planets and the number has now grown until something like seven hundred little worlds varying in diameter from 10 to 100 miles have been identified as circulating in the wide space between Jupiter and Mars. Their orbits have no common point of intersection and Bode's law has long since been consigned to the limbo of discarded theories. The number is now so great that names are no longer attached to them nor are there enough astronomical students to compute the orbits, so that today the great majority of the flock circle through this wide vacant space of the solar system as stellar mavericks, unsought by the terrestrial telescopes and unbranded by the yoke of the astronomical computer.

They could all be rediscovered by a single instrument such as was constructed by Brashear in a fraction of the time devoted to their discovery by the old-time astronomer of the nineteenth century.

In view of the great success realized with the Brashear cameras Dr. Wolf invited Brashear to name one of his planets. Brashear conferred upon the new planet the name of "Pittsburghia." The orbit was computed by Dr. Berberich. At opposition Pittsburghia is visible in the telescope as a 12½ magnitude star, its mean distance from the sun is 249 millions of miles and it requires four years, four months and twenty days to travel around its orbit.

Whether Juno and Pallas and Ceres and the other classical divinities have undergone any perturbations of an unusual sort by the introduction of little "Pittsburghia" into their select company does not yet appear. Most of us do not associate the name of Pittsburgh with the celestial places, but it always pleased Brashear to recall that through all the ages of the future a little planet would travel its appointed orbit round the sun, just as regularly as the big fellows like Saturn and Jupiter, bearing the name of his beloved city.

Brashear as a scientist, stands as the great technician of the new astronomy. His lenses, his mirrors, and his prisms, his gratings and his stellar cameras made possible those refined observations which have opened a new chapter in our knowledge of the physical conditions of the stellar universe. Astrophysics is not different from other physics except that it studies the physical properties of highly heated bodies millions of miles away. The spectroscopic and photographic studies that are made of these rays are exactly the same kind of spectroscopic and photographic studies that are made of the light rays and the heat rays in the physical laboratory. The lens of the telescope is only an incidental part of the apparatus and the great results are gained by the perfection of those delicate instruments which are able to recognize a small displacement of an absorption line in the spectrum of the sun, of the stars, and of the nebulæ. Brashear's scientific fame rests on these great achievements. He stands in the history of science as the master mechanic of the new astronomy.

The last fifteen years of Brashear's life saw him turn once more to the passion of his earlier years. With him the humanitarian and the scientific imagination had lived side by side and in his last years, and particularly after the death of his faithful wife, the old passion for human brotherhood once more became the dominant factor in his life. His work as Chancellor of the University of Pittsburgh, as administrator of Mr. Frick's gift for the teachers of Pittsburgh, and his untiring service to the Carnegie Institute and the educational interests of Pittsburgh was the expression of the devotion which had always been a part of his life and which became in his last years the moving force of his existence. He traveled over the United States, lecturing to many hundreds of audiences, reviving once more among people the enthusiasm and interest which had been in his own younger days aroused through the lectures of Mitchell and his colleagues. Into all this work he threw a devotion and a sweetness of personality which made him a power in increasing and deepening those forces in education that make for good will and right living and earnest devotion. He did not pretend to be an expert in the organization or direction of education, but he was a missionary of kindliness and human service to those who wrought in formal education. After one of his astronomical lectures to the West Point cadets a very discriminating listener, himself an astronomer, said to Brashear, "These boys will soon forget what you have said to them about astronomy but they will never forget you." It was a true word. Brashear's service to education was the gift of his own personality alive with the twin passions for science and for humanity.

At the end of Morley's Life of Cobden, when the historian has finished the story of a human life spent in public service yet remarkable chiefly for its intensely human qualities, he closes with this significant statement: "Great economic and social forces flow with a tidal sweep over communities that are only half-conscious of that which is befalling them. Wise statesmen are those who foresee what time is thus bringing, and endeavor to shape institutions and to mould men's thought and purpose in accordance with the change that is silently surrounding them."

There is in this statement a profound truth not only for political leaders but for all men who are in some sense leaders. Brashear came to be what he was first of all through his own personal character and ability, but he was made into that force which is formed out of character and ability by the sweep of great underlying movements of which he reckoned little. The wide-spread interest in astronomy which occupied the public mind of his younger manhood and which touched his imagination was the result of a great spiritual and intellectual movement that embraced both Europe and America. Out of that novement grew the Allegheny Observatory, and the Allegheny Observatory brought Samuel Pierpont Langley to Pittsburgh and made the opportunity through which two men, who mutually supplemented each other coöperated to accomplish great and splendid results.

And there was yet another who needs to be always spoken of when the work of Langley and Brashear is remembered, and that is William Thaw of Pittsburgh, who died in 1889. In his day he was probably the wealthiest man in the community and the last twenty years of his life were devoted in large measure to the support and encouragement of science and art. It was his intelligent appreciation that made Langley's work possible, and his help and sympathy also made Brashear's optical shop a possibility. When Brashear came to the parting of the ways, when he had to choose between remaining a skilled worker in the manufacture of steel or a designer in his own shop of telescopes and spectroscopes, it was the interest and help of William Thaw that made possible the solution which he desired to make. For many years he bore the financial loss arising out of the development of Brashear's shop so as to make it the most perfect optical factory in the world. But more than all this, he gave both to Langley and to Brashear the encouragement and support of a large-minded, far-seeing, generous man of the world. His name deserves to be remembered when the achievements of Langley and Brashear in the field of astrophysics are praised.

As we recall the service of these three men — so fortunately associated in the service of science and of humanity — as the years bear them farther into the mist of the past, we realize that their association was not wholly accidental. All three were borne along in those great currents of social and intellectual activity which marked the mid-years of the nineteenth century. To many these forces were invisible, but each of these had the imagination to conceive somewhat of the direction and intensity of these great forces. It was not wholly by accident that the astronomer, the technician, and the man of business became co-workers in the fields of science and of human betterment. For science and human sympathy and finance and education are not separate things. They are all parts of one thing — they are expressions in human activity of that same infinite and eternal energy through which the stars move in the heavens, the light waves race through the immensities of space, and the earth itself travels age after age its annual path. These three men were human transformers of that same infinite energy, brought together by the deep unseen currents of their day and generation.

HENRY LAURENCE GANTT

DIED NOVEMBER 23, 1919

MANAGER OF THE SOCIETY, 1908-1911

VICE-PRESIDENT, 1913-1915

No. 1757

BIOGRAPHY OF HENRY LAURENCE GANTT

IN the death of H. L. Gantt on November 23, 1919, the engineering profession has lost one of its foremost engineers, a pioneer in industrial management and a keen student of the human element in its relations to industry.

Henry Laurence Gantt was born on May 20, 1861, in Maryland. He was graduated from Johns Hopkins as a Bachelor of Arts when only 19 years of age, and in 1884 he received the degree of Mechanical Engineer from Stevens Institute of Technology. He was associated with Frederick W. Taylor in his early work at the Midvale and Bethlehem Steel Companies, and with this as a basis and his personal ability as an organizer he later established and successfully conducted his own consulting practice as an industrial engineer.

Mr. Gantt was active in The American Society of Mechanical Engineers and contributed many valuable papers to the Society, among which his early paper on the bonus system brought prominently to the front new features of management developed during his early work with Mr. Taylor. He was also the author of three books: Work, Wages and Profits: Their Influence on Cost of Living; Industrial Leadership (Yale Lectures); and Organizing for Work. In addition, he contributed many articles to the technical and daily press.

During the war Mr. Gantt acted in a consulting capacity for the Ordnance Department. His production charts, well known among engineers and constituting an important development in recording the progress of work, were used by him in the Ordnance Department and later also by the U. S. Shipping Board and Emergency Fleet Corporation in routing ships and in following up construction work.

Mr. Gantt's career was marked by original and thoughtful work, progressive in viewpoint and effective in its accomplishment. He developed a broad conception of industry as a national problem in which he regarded it essential that the man at the top should have the same close scrutiny and careful direction that has in the past been given his co-workers in the lower ranks.

Mr. Gantt's vision of a better spirit in industry is well expressed by the following quotations from his latest book, Organizing for Work: "The community needs service first, regardless of who gets the profits, because its life depends upon the service it gets"; and "The business system has its foundation in service, and, as far as the community is concerned, has no reason for existence except the service it can render."

No. 1757 *a*

MR. GANTT'S CONTRIBUTION TO INDUSTRY

By Fred J. Miller, Center Bridge, Pa.
President of the Society

FOLLOWING a liberal and a subsequent technical education Mr. Gantt's industrial experience began and for many years continued under conditions which tended to magnify the importance of methods and the minute study of minute details of doing work. Though he never denied the intrinsic value of these minutiæ, he grew finally to attach relatively much more importance to the human element in an industrial organization.

2 It was my privilege to be, for about ten years, in a position to coöperate with him, a period during which this evolution was taking place. The extent of that evolution is indicated by the fact that, whereas at one time he had believed and had declared that minute studies of details of mechanical operations, based upon stop-watch observations, were an absolute necessity as a beginning for the improvement of such operations, for some time before his death he had practically discontinued the use of stop-watch time studies.

3 This was not because he had come to believe them to be of no value, but because he had become convinced that so much could be done for both management and workers without such minute studies, and that such excellent results could be obtained so much more quickly and with so much better effect in securing the immediate coöperation of all interested — executives and workers, that the stop watch and all that it connotes in time-study work was in his practice and teaching forced into the background, to be used,

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if at all, only as a help in the final perfecting of mechanical manipulations.

4 In other words, Mr. Gantt, being a philosopher as well as an industrial engineer of the first rank, had the breadth of mind which permitted him to be guided by the course of events; and seeing that in almost every industrial organization there were men who already possessed a great deal of knowledge concerning their work which for one reason or another they did not and often could not utilize, he set himself the task of helping these men, both workers and executives, to make the fullest possible use of this accumulated knowledge acquired by experience, that is, to remove obstacles in the way of its fullest utilization — obstacles usually traceable to the fact that industrial management has not generally kept pace with the growth of industrial enterprises and that engineers concerned in management have too much confined themselves to study of methods of handling and transforming materials and have given much too little attention to study of the human beings who must maintain and operate our mechanical devices used in production.

5 He still used, of course, many of the methods and mechanisms he had been accustomed to use and which had been proven valuable by long and wide experience of his own and of others; but his first concern in undertaking a piece of work was to find out what particular department or group of machines or of workers was believed to be most unsatisfactory in point of quality or quantity or cost of work done; then to find out what those most competent to judge believed ought to be fairly expected or attained by that department or group. He then ascertained by methods of his own not only how much short of that standard the actual performance was day by day, but also why and just who was responsible. He never allowed himself to prejudge the case or to assign this responsibility either to workmen or management until records had been made that plainly showed where the responsibility lay.

6 In the vast majority of cases he found and made plain the fact that in one way or another the responsibility for correcting the trouble was upon the management; then he stood ready to show by thoroughly tested and proven methods how to correct it. Thus very often blame that had been placed upon workmen was shown not to be theirs; but when their methods or practice were found to be at fault, teaching was resorted to, always upon the assumption that workmen will respond to fair treatment and are as anxious to be successful and to "make good" as others.

7 Mr. Gantt's influence upon industry cannot be accurately or adequately appraised now and possibly never will be. But as the evolutionary process now going on is further developed his influence will be more clearly perceived and appreciated; and no one familiar with his work can doubt that his name will stand very high among the group of men who have not only insisted that things were wrong in our industries but have proposed definite remedies and demonstrated the practicability of those remedies.

8 Mr. Gantt was always entirely candid, sincere and honest. No one needed ever to be in doubt as to what his opinions were concerning those matters with respect to which he considered himself qualified to have definite opinions. He was not always appreciated, for there are those who for one reason or another do not like plain, clear statements of facts and equally object to being faced with the necessity for making clear-cut, definite decisions and carrying the responsibility for them, when all the facts are known and there is little or no chance for evasion or shifting.

9 But Mr. Gantt's ideas are very prevalent in one form or another in the current literature of industrial management, although they are put forth sometimes by those who have only a very limited understanding of them or of the philosophy that underlies them. There are establishments, however, in which the methods and principles he taught are being followed with great success and satisfaction to all concerned, workmen, executives and owners; a result not attained by mere "stunts," as Mr. Gantt called them, but by thoroughly coördinated and harmonious organization and methods, making it easy for each person to do the right thing at the right time and in the right manner.

10 While he did not deprecate so-called welfare work per se, he did insist that it could never be a satisfactory substitute for simple, plain justice and fair dealing based upon a definite knowledge of ascertained and plainly recorded facts. I have faith to believe that his work and teaching along that line will continue to have a very great influence upon the final development of the science of industrial management. Especially do I think this will be the case as the world generally is more and more forced to recognize the simple fact that industrial establishments are created and maintained for the production of goods and therefore must be under the direction of those competent to direct production and to secure the real interest and coöperation of the workers.

11 To this plain proposition Mr. Gantt dedicated himself

and his talents; and when he found that a concern was dominated by those who had other objects or who wished to utilize his work in taking unfair advantage of either workmen or the public, his interest in the development of that concern was gone.

12 He himself played fair and he insisted that all with whom he was associated or in whose service he was employed should do the same.

13 His books teach, above all other things perhaps, the importance of the square deal in all that relates to industries or to the work of the industrial engineer; and no one, unless he has entirely lost faith in human nature, can doubt that in so doing he contributed in an important way to that which is and always will be a fundamental requirement for genuine success in industrial enterprises.

No. 1757 *b*

THE LIFE WORK OF H. L. GANTT

By E. A. Lucey, New York, N. Y.
Member of the Society

A GREAT voice has been stilled; but the true and forceful sayings of the late H. L. Gantt will live forever, for such a life cannot die. Not only the engineering profession but the country at large lost a truly great man when Mr. Gantt passed away on November 23, 1919. He was in the prime of life and doing his best work when he was called. Few men have so successfully rounded out their life's work, for he spent the last three years in consolidating and tying together the results of his former efforts.

2 After leaving college he tried drafting and teaching school with very little success, and it was not until he started work in the shop and decided that he would learn things from the bottom up that he found himself. He found that his greatest power was in his ability to understand people, to analyze operations, and to develop better methods of doing work. In the early days of his shop experience he met and worked with the late Frederick W. Taylor, and they started together some of the methods for which both became famous. In speaking of his own work and of that of his associates, Mr. Taylor in one of his most memorable works stated that Mr. Gantt was the greatest manager of them all.

3 An untiring worker, daring in the work that he attempted,

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steadfast in his decision to use only facts and never opinions, his influence was felt over a very large field. Of strong character, he imbued others with the spirit of his work. Young men and old were drawn to him and he was able to see some good in every individual.

4 He believed that people should be taught things instead of words and that any amount of instruction was worth while, but that skill could not be acquired by simply repeating an operation. A man must put his soul into his work and have the feeling and the will to do; therefore Mr. Gantt tried to make work interesting and was repaid for his efforts.

5 In the early years of his experience in management work at Midvale, and at Bethlehem, things were done in a very autocratic manner. For example, nothing could be moved except by a move order; work was assigned to a definite machine rather than to a class of machines, and the foreman was not allowed to decide which machine to use. Mr. Gantt believed that this was wrong, although perhaps necessary at the start because the people in authority did not understand other methods. He realized long before the others the strong part played by the human element and that executives could do more by leading than by driving.

6 It was the realization of these things that made him a good manager and that induced him to write his paper entitled Training of Workmen in Habits of Industry and Coöperation. His former associates and many of his friends tried to have him withdraw the paper before it was published, claiming that the time had not come for any such preaching, but he had tried out this work for many years before writing about it and had gotten such wonderful results that he was sure of his ground. The paper proved to be very timely, and more good was accomplished by this method of working than by anything that had been done before.

7 His book Work, Wages and Profits was written shortly afterward and was a further attempt to explain his ideas on the subject of training workmen.

8 It was not until he was forty-four years old that he saw the real solution of the problem of management. At forty-four most men are so set in their ideas that they cannot change their habits of work. Yet he set about a task which was to change many of his own ideas and which had so many difficulties in the way of its solution that it took ten years to solve it and three more to put it in its best form.

9 Mr. Gantt was always a strong advocate of the task idea in management and he found while pushing his methods of training

workmen that setting tasks to measure the work of employees was of relatively small importance as compared with setting tasks for executives. He found also that setting tasks for employees was one of the last things to be done in a plant, rather than one of the first, as most people considered. The amount of profit made was usually taken as the measure of an executive's work. But this was not a good measure as market conditions and other things beyond his control had as much to do with profits as the executive. For instance, one year in which a small amount of output was produced might show more profit than the next year in which everybody worked harder and produced more. Mr. Gantt claimed that the true measure of an executive's work is the product turned out as compared with the total possible product, and that productive capacity is the true measure of the value of an industrial property. He found, however, that while the managers and owners who sought his aid were glad to have him measure the work of the employees, they were not so anxious to have him measure their own effectiveness. It was not until he developed his methods to show idleness expense that he had any great success along this line. This method was welcomed because it gave facts in a compact, understandable form rather than anybody's opinions. It not only showed a concern how it was running its plant, but gave to the executives a mechanism for correcting most of the troubles that existed. His paper entitled The Influence of Executives and the book Industrial Leadership are both helpful contributions to the art of management.

10 In the few years preceding the war he worked out his method of straight-line charting in which he based everything on the element of time. During the war these charts were used by many of the departments of the Government to keep track of their progress, and by thousands of manufacturing plants to plan their work and to increase production. This method of charting is being used quite generally throughout the country at the present time and by it many complicated situations which had never been visualized before are easily controlled. It should be emphasized that as this method of charting was based on the element of time, it was really an improved and simple method of scheduling. This bit of work brought out once more Mr. Gantt's strong belief in the task idea in management.

11 His path was not always an easy or a pleasant one to travel, for like any man who is ahead of his time Mr. Gantt was opposed and misunderstood by a large number of people. He himself ex-

plained this very well in his paper read before the 10th Annual Convention of the National Machine Tool Builders' Association, when he said, "The greatest obstacles to the introduction of our methods have not in the past come from the workmen, but from the foreman or others in more or less authority. Those offering most objection have, as a rule, either not understood what was being done, or felt their inability to hold their jobs if they were asked to perform them in accordance with the high standards we set." It is a fact worth mentioning that workmen have never objected to his work and that there has never been a strike or other labor trouble in any plant in which he had introduced his methods. Part of this may have been due to the type of clients for whom he worked, but it was helped by his insistence that no tasks for workmen be set until the functions of management were straightened out, and until it was understood that no task would ever be increased.

12 The manner in which Mr. Gantt held the respect and friendship of those with whom he came in contact can best be shown by the way in which his associates and helpers stuck by him in all of his work. For fifteen years he had as his principal associates, Wm. E. Pulis, R. A. Wentworth, C. E. Volkhardt, W. L. Conrad, J. F. Mullaney, and the writer, in addition to a score of helpers and assistants. They so valued his leadership and he their suggestions that during all this period not one of them ever gave a thought to separating from their principal and working independently. In fact, it became a habit with a large number of men who had had dealings with Mr. Gantt and understood the value of his impartial advice and judgment, to undertake no new project in their business without first consulting him. Working for such a man was a privilege and an education and there was always a large number of men seeking a chance to join his happy family. He often said that his success was due to the work of his associates, but it can be shown that their good work was made possible by his guidance.

13 Mr. Gantt spent many years in devising methods for the training of executives, and for measuring their effectiveness. His conclusion was that the influence of executives on production can be determined only by a careful record of how the facilities at their command are being used. As it is one function of the executive to keep the plant in operation, we can find out how he is performing that function by asking why his machines are idle. To those who have not figured the expense of maintaining machinery in idleness, the amount of such expense in the ordinary manufacturing plant

comes as a great surprise. He was convinced that a study of idleness is much more effective in increasing the output of the plant than the study of what has been called "efficiency," and that the one should always be completed before the other was attempted.

14 He assumed that industrial plants were built to produce goods and therefore directed his efforts:

(1) "To finding out how they were performing the function for which they were created."

(2) "To finding out the reasons why they are not doing as well as they should."

(3) "To removing obstacles which hamper them in the performance of their functions."

15 To aid in the above work Mr. Gantt developed Machine Record Charts, Order Record Charts, and Man Record Charts, which were of great assistance in his work. All of these charts are based on his straight-line method and aim to show what was done as compared with what should have been done. These charts will be found to be of great help in any plant and can be used under any type of management or method of wage payment. His last book, Organizing for Work, explains these methods in the clear, straightforward manner in which he could so well express himself. Many of us believe this book to be the greatest one ever written on the subject of management, as it shows a real constructive way to oppose socialism or bolshevism. Ever since the start of what has been known as "Scientific Management," Mr. Gantt has been its most active exponent. His writings have always been constructive and his work more lasting than that of any other man in the field of management.

16 It is worth while to again call attention to Mr. Gantt's strong belief in the task idea. Through all of his work and writings this idea can be traced, for it was indeed the foundation that he built on. What was done he measured by what should have been done. His Man Record Chart and Bonus Record Chart show this for the workmen, and his Idleness Expense Chart shows this for the management.

17 Many men outside of his own organization gave Mr. Gantt valuable assistance in his general work and it would be difficult to name all of these. Too much credit, however, cannot be given to Mr. Horace Cheney and Mr. Fred J. Bendall for their share in the development of the idleness expense work, and to all of those manufacturers who made it possible for him to put his ideas into practice.

18 The underlying purpose of Mr. Gantt's life was to benefit the community; and his method to secure that benefit was to increase production. He knew that the responsibility for increasing production lay far more with the executive than with the workman, and he felt very strongly that since idle labor is not entitled to compensation, then to an even lesser degree is idle capital entitled to a return. His biggest problem was to measure this idle capital, and it was the solution of this problem which was his biggest service. But his whole life was a life of service, and if service is to be the measure of a man's usefulness, then Henry L. Gantt's life was well worth while, for he lived to serve and gave his life to that end.

No. 1757 *c*

HENRY LAURENCE GANTT

AN APPRECIATION OF HIS WORK FROM GREAT BRITAIN

By James F. Butterworth,[1] London, England
Non-Member

When the sad and unexpected news of the death of Henry L. Gantt reached his former friends in Great Britain early in December 1919, they were overcome with consternation and regret; and all of them felt that the world was much the poorer by the untimely loss of such a gifted man. To those who had the privilege of his personal friendship, and to whom the cause of industrial management meant much, the feeling of irreparable loss was overwhelming.

2 That the comparatively new discovery of scientific management should suffer such disaster as the unfortunate deaths of Dr. F. W. Taylor, its discoverer, and his friends and colleagues, James Mapes Dodge, and Henry L. Gantt, within the short space covered by five years, was truly misfortune heaped upon misfortune. To be cut down before their work was finished was hard, but the work accomplished by them, and the records they left behind, are a sure indication of the lines to be pursued by their devoted followers and

[1] 12 Hawthorne Terrace, Notting Hill, London W. 11.

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admirers in order to achieve economic prosperity for humanity a large. This task would have been nearer fruition today had it not been for the great war, and the effects of its aftermath.

3 Fortunately the industrial truths enunciated by these deceased pioneers, and some of their surviving and accomplished successors, are at length attracting the careful and increasing attention of industrialists and others on this side of the Atlantic, many of whom are giving practical effect to their teachings, with beneficial consequences to employers and employed.

4 As yet we have much to learn in Great Britain and our overseas possessions about industrial management, but I am happy to say we have an ever-increasing number of earnest students of this complex problem, and most of them turn with avidity to the lucid writings of our late friend. "I like Gantt's books" is a sentiment often heard over here. When his Work, Wages, and Profits appeared in this country, many of the leading newspapers in Great Britain published appreciative reviews of its contents, and a well-known economist in the Manchester *Guardian* summed up his criticisms by stating: "All works managers and trade-union officials should study this book."

5 His later work, Industrial Leadership, is widely read and valued, and his loyalty and tribute to the memory of his former chief, Dr. F. W. Taylor, admired.

6 His last and as some think his most important book, Organizing for Work, has not been available on this market sufficiently long to be as fully esteemed as its undoubted merits will cause it to be. That H. L. Gantt's writings are of great value there can be no two opinions, and generations yet unborn will benefit from their teachings.

7 When my late friend Gantt sent me a presentation copy of his last book in October 1919, I loaned the same to the editor of *Engineering and Industrial Management*, of London. He considered the work so important that he devoted five pages of the issue of the Dec. 18, 1919, to a notice of it, reproduced four of the charts, and gave the book unqualified praise. By a sad coincidence the announcement of Mr. Gantt's death appeared in the same issue, accompanied by a short sympathetic editorial appreciation of his labors, together with an acknowledgment of his worth, and concluded by saying: "His services to the cause of humanity and efficiency will be an imperishable monument to his memory." This journal has also published several of Mr. Gantt's papers, many of his trenchant

sayings, and a month before his lamented death, his portrait, together with a brief sketch of his career.

8 The July issue of *The Industrial League and Council Journal* contains a review of the recently published English edition of Organizing for Work, which says:

> The late Mr. H. L. Gantt . . . not only did wonderful work, but he also published certain books which are recognized to be amongst the most important contributions to the literature on the subject of Management. Any one who has read his Work, Wages, and Profits knows how thoroughly Mr. Gantt understood the whole subject, and one of his most important books published just prior to his unfortunate death is Organising for Work. In this book Mr. Gantt deals with numerous points in masterly fashion on his principal theme, that Scientific Management is not a question of getting the most out of workmen, but is really designed to obtain efficient Management, and through that, the proper use of all factors of production. It is a very common saying nowadays that workmen must not expect to take more out of industry than they put into it, but this is equally true of the business man, or, in other words, the Manager. If the Manager fails to provide the proper facilities for the workman, then the workman cannot be expected to do his best. . . .

9 The textile journals of this country have made frequent flattering references to his work in the bleaching and cotton industries, and have reproduced some of his charts for the guidance of their readers. The textile, bleaching, and dyeing industries have installed many of the Gantt continuous piling machines in their works, and have found that this ingenious device has solved some labor troubles, many of their past difficulties, and materially reduced their working costs.

10 In conclusion, I would add that most people in Great Britain can only judge of Henry L. Gantt by his published writings, and I feel certain he would have been content to be so judged, for they are full of counsel and wise words, and bear ample evidence that he was a man who could have truly said: "I pray thee, then, write me as one that loves his fellow-men."

No. 1757 *d*

H. L. GANTT — IN APPRECIATION

By S. Marshall Evans,[1] Chicago, Ill.
Non-Member

H. L. GANTT was born in Maryland, educated at Stevens Institute and Johns Hopkins University. His first work as mechanical engineer was at the Midvale Steel Company in collaboration with Frederick W. Taylor, the first great protagonist of modern manufacturing method and organization.

2 Gantt was soundly trained. Sincerity, human sympathy, rugged honesty, clear thinking, fearless advocacy of the right as he saw it, indefatigable at work — these finest of natural endowments, combined with his sound engineering training, gave to his profession in H. L. Gantt not only one of the greatest engineers of production, but also as many of us believe, a prophet of a new order in industry.

3 Previous to the war our people, young and old, found their chief pride in those leaders who achieved largest financial success. We were proud of our country, because such achievement was possible in our democracy to a greater extent than elsewhere in the world.

4 The ascendency of the engineer was the result of the inexorable necessities of the war. That our democracy should have first given heed to its financial and business leaders was natural. We were ignorant of warfare itself, and even more ignorant of the industrial technique and the national inter-organization of all industry by which alone modern armies may be supported, supplied and carried through to victory.

5 We entered the war in April, 1917. By the end of that year those who knew how little had been accomplished, were forced to forego the plans for our 1918 campaign and to plan for our offensive in 1919. Complacently enough these plans were regarded as inevitable,

[1] Eagle-Picher Lead Co., Chicago, Ill.

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until one day a certain general landed in New York, hurried to Washington and jolted the management with the grim facts of the war. Bluntly enough the President and the War Secretary were told that by 1919 our allies would no longer be able to fight; that if our allies and ourselves were to achieve victory, it must be by the advent of our American armies fully armed and equipped in the fighting zone by the middle of that year, 1918.

6 Great national need upon us, we finally turned to those men who knew how. They were there in Washington in considerable force. Vainly had they been pointing to the simple ways of hard work by which our vast requirements could be understood and procured. These men were those who had spent their lives in construction and production, and the greatest and most potent of them all were the engineers. Of the engineers who volunteered their services, many of us believe, Mr. Gantt was the foremost.

7 The outstanding requirements — ammunition, artillery, airplanes, small arms, ships — all were accelerated by the touch of the masterhand of him whose memory we are met to revere and commemorate.

8 His effectiveness in meeting these great emergencies is simple enough to understand when one knew the man. His whole life up to this time had been a study in preparation for exactly this emergency. His experience was almost unbelievably broad. His service had been sought by our greatest industries — electricity, steel, mines, textiles and fabricating plants of almost every description. The development of the principles underlying his accomplishment may be readily enough sketched in a paragraph. Associated as he was in the beginning with Mr. Frederick W. Taylor, these two men, with their associates, comprised an engineering laboratory force, their study the fact study of modern industrialism.

9 Shortly before Mr. Taylor's death, Mr. Gantt became more and more impressed with the appalling waste due to the fact that man power had always been regarded exactly as machine power, to be handled with a like degree of objectivity. His experience had already revealed to him the enormous potential for increased production, where conditions were such that men gave of themselves enthusiastically rather than grudgingly to their work, and, very early in his writings, we find the fact emphasized that men do want to be interested in their work, and that men will take pride in work well done.

10 Thus, the problem of industry became to him essentially the problem of management, and the problem of management, as he

saw it, was that management be led to the creation in industry of conditions whereby each man would know what he had to do, the extent to which he did it, the relations and importance of his task to the purpose or object of the industry, — in short, the conditions wherein and whereby each human unit in an enterprise might render the greatest service of which it is capable.

11 Mr. Gantt's vision was such that any particular shop in which he was working encompassed the whole circle of commerce. He never lost sight of the horizon. Small things were great to him and great things were small. Or, if you prefer, nothing was either small or great, everything was endlessly significant. That is why he developed an extraordinary talent for exactitude. He taught the industrial world how to measure its tasks to a hair's breadth, but he would not have these measurements applied as a substitute for thought. He insisted upon exactness as a stimulus to thought. It was his way of featuring the job. He made the details vivid, in order that every workman might have definite views day by day of the part he was playing in the moving drama of a creative process.

12 It is difficult to obituate about a man like Gantt. He was so completely absorbed in what he was doing that what he did is the thing to talk about in an hour set aside by his friends to refresh the memory of it.

13 It is difficult for anybody who knew him to imagine his not going on. Indeed, I find it hard to think of the pressure of present events — the swift and accelerating advance of the world movement in which he was involved, without half expecting to hear his voice. Indeed, this man was so deeply involved in the undying world of reality, in the study and development of sound principles of industrial management, in the blazing of trails, which we, all of us, are bound to follow, that we are, from the very fact of this meeting in the contemplation of his work and in our acceptance of it, making a concrete fact of the idea of immortality. In other words, this man cannot die.

14 And the integrity of the man — he was absolutely all of one piece. You could not distinguish his egotism from his altruism. There was no difference between the two. He was arrogant in self-assertion, because of his complete objectivity. He never entertained the notion that his private fortune could be separated from the fortune of the world. He was busy improving the general estate and naturally assumed that he was in the middle of the improvements.

15 He had faith. Not the faith of a passive pietism. His faith

was dynamic and substantial — the very substance of hopeful things. It was foreknowledge — like that of an astronomer who foretells a transit of Venus, because he has no doubt of the integrity of the universe. Gantt had no doubt of the recuperation of the working world, because he understood more profoundly than smart and cynical people do, the nature of men and the nature of things.

16 His insistence that the first thing to do in a factory is to stop the waste did not proceed, as one might suppose, from an impulse of frugality. That was not the point. The point was that you cannot make a beginning in the improvement of any process, until you know what the process is in its frictionless and perfected norm.

17 It is not necessary to go into the methods of his work, here among his friends. Suffice it to say, that wherever he worked both management and men were heartened to their tasks, perceived them more clearly, and the industry in question vastly benefited. His gospel was that of fact rather than opinion; of fact rather than prejuduce; truth instead of error.

No. 1757 *e*

APPRECIATION OF H. L. GANTT'S WORK IN FRANCE

By Charles de Freminville, Paris, France
Honorary Member of the Society

NOTHING could impress me more than to be invited among the friends of Mr. Henry Laurence Gantt to say how the work to which he consecrated his life has been appreciated in France. The remembrance of the cordial attentions I always received from him would not permit me to decline this invitation, but I fear I shall not do justice to the importance of the occasion. Because to speak properly of a man possessed of so many qualifications, whose activity was stamped with such ardor in the most varied and vast fields, it would be necessary to possess his own power of penetration, in order to encompass at a glance not alone the processes of industry, but also

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the mentality of the men who set them in operation; it would be necessary to follow him in his appreciation of the unprecedented economic evolutions of industry which we now witness; in a word, it would be necessary to visualize, as he did, the social idea of a superior order which should correspond to the material progress of our civilization, an ideal which Mr. Gantt conceived by the eminence of his mind and the superiority of his genius. I cannot entertain the thought of fulfilling all of these requirements, but I am convinced that our friends in America will be satisfied if I succeed in demonstrating simply how Mr. Gantt has been understood in France and how his benevolent influence has been exercised there, and this is what I shall endeavor to relate.

2 The United States has been from its infancy united to France by the bonds of a friendship which is strengthened more and more every day, and the thinkers of the two countries have always associated in the study of those great problems on the solution of which depends the future of our civilization and the defense of our common heritage of liberty. Upon this field an interchange of ideas has continuously taken place.

3 It was thus that the French heard speak of H. L. Gantt in the first years of the century. They learned then that he formed part of a group of Americans working assiduously among themselves, whose works were revealed by Frederick W. Taylor, the distinguished engineer who was the first to set forth the fundamentals of industrial management.

4 These works were enthusiastically received in France from their appearance, in a great measure by reason of the scientific method with which they had been conducted, but it was soon realized that the pioneers who were interpreting them were engaged in nothing less than establishing the needs and aspirations of a new era, and that in analyzing them they were endeavoring to orient them.

5 This was the occasion for the establishment of more cordial relations between the engineers of the two countries. More intimate friendships soon developed and it is to pay a tribute of friendship that the writer is prompted to bring here his testimony of admiration and affection.

6 In less than fifty years industrial tools were developed to such a point and acquired such power that past achievements were eclipsed; with the new tools the exploitation of natural resources developed more wealth than the imagination of former generations ever conceived; transportation facilities increased with such rapidity

that all the markets of the world were open to industry and distances were immeasurably shortened.

7 Profound modifications resulted in the operations of industry and in the mutual relations of all those concerned with it. The new machinery permitted the utilization of unskilled workers who, to do their work successfully, had merely to show a suitable spirit of coöperation. This was difficult because, according to the expression of Mr. Gantt, "it is easier to make men work skillfully than to make them work harmoniously."

8 Besides, this new machinery required the application of capital, formerly engaged in commercial operations, to the necessities of industry. Those who contributed capital assumed first place in the direction of industrial affairs and brought with them a new spirit in industry. "They had bought in the cheapest market and sold at the best price they could get. Their natural tendency, therefore, was to apply to the purchase of labor the same rules they had applied to the purchase ofmaterials, namely, to buy it as cheaply as possible." [1]

9 Finally, production in enormous quantities and distribution of products in all markets changed completely the economic and even the social equilibrium of the past.

10 Economists applied themselves, it is true, to the study of the new situation, but in an effort to secure a basis of common agreement among themselves, they tried in vain to establish their laws of the new evolution by mathematical reasoning based on global statistics in which they placed implicit confidence. The law of supply and demand, whose operation was no longer interfered with since the removal of the obstacles formerly introduced by the regulations of ancient corporations, must, according to them, suffice to the establishment of a stable equilibrium. It is not an exaggeration to say that in bowing low before this law and accepting all its consequences without reserve, the economists did nothing more than to enthrone empiricism and individualism and the craving for an immediate profit. They ignored that the new processes of industry demanded important scientific knowledge and a coördination of efforts all the more difficult to realize by the diversified character of industrial activities. Recent evidence has clearly demonstrated the imprudence of trusting implicitly to the law of supply and demand whose normal, or rather, ideal operation is never realized. The imperfect equilibrium which it had established has crumbled and it shows itself incapable of reëstablishing it.

[1] Industrial Leadership, p. 3.

11 As Mr. Gantt points out, this law is no other than the law of profit. Its imperfections seem to have become manifest more prominently in countries of great industrial production and particularly in the United States where groups of producers have been prosecuted for having made a fraudulent use of such law to the detriment of the interests of the public. It is against such that Mr. Gantt writes so forcibly in his last book, Organizing for Work.

12 The school of scientific management, in which Mr. Gantt occupied so prominent a place, undertook also the solution of the economic production, but it followed a method altogether new.

13 Instead of finding the explanation of industrial crises in vague and uncertain theories and rigid formulæ, it developed a truly scientific spirit and its methods rested on psychological principles. It applied itself to the study of the simplest elementary facts, and demonstrated that industry suffered from evils whose importance had been underestimated and which it would not be possible to remedy without effecting important modifications in the industrial organization.

14 Contrary to all expectations, the researches of the new economic school put in evidence that in the workshop itself a very imperfect knowledge of industrial technique predominated, and often even a complete ignorance of the most elementary scientific data on which industrial operations are based.

15 It discovered also the existence of the strange lack of coördination in efforts which were directed to the same purpose.

16 Finally, it did not satisfy itself with indicating the trouble but it pointed out the means to overcome it.

17 The work of this new school was followed with interest in France by H. Le Chatelier, the well-known scientist, who was struck particularly with the scientific spirit of the Art of Cutting Metals, and with the realization of the marvelous results which it was possible to obtain by systematizing industrial methods.

18 The publications which aroused interest in scientific management in France were those of Messrs. Taylor, White, Barth and Gantt whose coöperation had been so intimate and so long that on this title alone their work well deserved to be classed as extraordinary.

19 Taylor told us that his co-workers were endowed with very different qualities. He said that White was among them the best metallurgist, Barth the best mathematician, and finally Gantt, the best manager of men, the best psychologist. He relegated himself to an obscure place, but his colleagues have done him justice, and Gantt pays him a sincere and well-deserved tribute in the following

words: "Industrial leaders who have most prominently attracted our attention in the past are those who have by their inventions or their direction of activities accumulated large fortunes; but none of them is as great as the man who by force of his intellect leads people throughout the civilized world to benefit themselves and others. Such a man was the late Frederick Winslow Taylor, who, in his determination to eliminate error and to base our industrial relations on fact, set an example which will have an effect all over the world." [1]

20 It is also to H. Le Chatelier to whom the readers of *Revue de Métallurgie* owe their acquaintance with the works of Mr. Gantt and in particular with Work, Wages and Profit.

21 When the ideas of the promoters of scientific management first appeared in France, their novelty fascinated, and the first discussion held about them demonstrated clearly how little the labor of the modern workman was understood, not alone by economists, but by industrials as well.

22 The operations of industry have three phases which it is well not to separate when they are studied: *Conception, Realization* and *Production.*

23 Industry must first conceive objects upon which it will concentrate its activity and then these objects once created must be carried to the degree of perfection which is compatible with the means available. Objects will subsequently be produced in quantities to a greater or lesser extent. The relative importance in industrial processes of each one of these three phases, Conception, Realization and Production, characterizes the different stages in industrial development.

24 For a long time Conception and Realization held the first place in industry. It was then the age of the artisan, and this latter, who has possessed remarkable skill in France, has left a footprint of his methods in the French industrial ideal.

25 The development of America has taken place when the element Production was already assuming the upperhand in industrial development. For this reason the problem of Production has occupied the first place here more than in any other country.

26 This is the reason why the French viewpoint in industry was so different from the American viewpoint. It is also the reason why the new ideas on the organization of work were at first understood with difficulty in France.

[1] Industrial Leadership, p. 27.

27 It was feared that the organization proclaimed by the new school placed a constraint on the worker, which seemed incompatible with the French ideal.

28 It was necessary, however, to recognize that Production had become in France as elsewhere a predominant factor and that the ancient ideal must conform itself to the new conditions.

29 The question, therefore, seemed to be whether the necessities of production would compel the substitution of the artisan of former times for the man reduced to the condition of an automaton.

30 Such was the point on which it was necessary to give precise information to the French public and it was Gantt who supplied it.

31 We first knew him by his Work, Wages and Profit. From the very beginning of this work, he dissipated the fear with which a rigid industrial organization inspired us. He says: "The law of development is evolution. Revolution is justified only when evolution is impossible. . . . A system of management specially designed for economical production is a mechanism which is successful only when all parts work in harmony. The men who form part of this mechanism must be trained individually and collectively. . . . Form and blanks are simply the means to an end. If the end is not kept clearly in mind, the use of the form and blanks is apt to be detrimental rather than beneficial."

32 We were astonished at the way in which he groups all the new notions around the task idea, the psychological side of which he so admirably explains, insisting on the usage which must be made of it in the formation of workers (training workmen in industry and fixing habits in industry), and also for the analysis of results. He demonstrates that "the setting of a proper task for the workman also imposes obligation on, or sets tasks for, the management." The few examples which he gives in this work are easily assimilated by the French mind and are so presented that they form a deep impression in the memory. This book has been for many of us the revelation of a new spirit.

33 Later on it was Industrial Leadership which attracted our attention. There Mr. Gantt explains to students the ideas which he had formerly explained to the general public. It is striking to note the importance he attaches to being understood by his new audience. He treats his object in a more concise and still more remarkable manner, and endeavors to inculcate in the minds of those who wish to become managers of industry, the principle "that in all problems in administration, the most important element is the human ele-

ment." He illustrates his doctrine with a variety of new and appropriate examples. Then, at the end of his chapter on production and sales, he discloses methods of accounting in direct relation with the utilization of the workshops, setting forth "what expense we are under day by day for that portion of the plant and equipment which are idle." He says that he did not invent these methods. But even if he did not invent them, it was by his genius that he brought them to the attention of others.

34 That which particularly gained the sympathies of the French engineers for Mr. Gantt was his constant endeavor to show that in order to successfully conduct organization work it is necessary above all to make certain the coöperation of all, and in order to do that, to make everyone understand the purpose which is aimed. It was with this view that he imagined the "charts" published in his last book, Organizing for Work, which have also been a revelation to many. He made himself understood admirably by resuming all his doctrines on organization with the word "preparedness."

35 But Mr. Gantt did not limit himself to the study of the means of organizing the workshop, utilizing the machinery and remunerating more or less justly the labor of the workman. His mind soared to still greater heights; he realized that the most beautiful mission of a leader is to form men, to produce men worthy of the name, and he would have wished to see enthroned in the industrial world, the world of today, the ideal towards which he impels the readers of his last work and which he so truly expresses with the word "service."

36 It may be said that civilization consists in a state of equilibrium more or less stable between the material wealth possessed by men and the moral ideas by which they are grouped. Gantt saw clearly that in our days the only ideal which can be found consists in the immediate satisfaction of individual interest, that is, Profit, and this ideal, he reasoned, could give but an extremely doubtful equilibrium.

37 Gantt searched through history and found that nothing had been accomplished with such a meagre objective. Public interest, "service," must guide the future operations of industry; it must furnish the element of necessary coördination to the development of the mentality and the morality of all. It was certainly to demand a great change and so he appealed to great men, hoping to find in them emulators of the great enthusiasts who headed the liberal movement in France toward the middle of the last century.

The enormity of such a task did not discourage him and he was ready to sacrifice his whole self to it when death put an unhappy end to to his activities.

38 It was in 1913 when I had the pleasure of meeting Mr. Gantt. I saw him in New York. I was seduced by his benevolence and struck by the superiority of his mind. He showed me his charts and explained to me his ideas on the organization of industrial work. I was then studying Work, Wages and Profit with special care.

39 A few months later he paid me a visit in France and I could show him that while I had not realized much as yet in the matter of organization, I had, nevertheless, a few charts capable of interesting him and that I was ready to profit by his ideas and perhaps also to make others profit by them.

40 We spoke repeatedly on the importance which a systematic interchange of ideas between American and French engineers would have in the development of the qualities and aptitude of both.

41 Since that time we always remained in constant communication, often interchanging letters and some times friends. All those persons whom I sent to him returned filled with enthusiasm and inspiration. I had anticipated the pleasure of seeing him in my last trip to America, but unfortunately I could not see him at my first call and when, a few weeks later, I returned to New York, I learned with sorrow that he had departed.

42 But the ideas which Gantt has sown among us will fructify. The application of the methods of scientific management is making rapid progress in France. The name of Taylor serves as watchword, and that of Gantt is constantly associated with it, and is no less familiar to all those who have made a study, however brief, of industrial questions.

He understood and loved our country.

He helped us powerfully.

He will always inspire us.

No. 1758 *a*

RAILWAY TERMINALS AND TERMINAL YARDS

By William Barclay Parsons,[1] New York, N. Y.
Non-Member

TERMINALS form the largest item in railway construction cost, said the speaker, and the most expensive item of cost in railway operation. On the average, 30 per cent of the valuation of the railroad is represented in its terminals and terminal yards. Similarly, it might be said that the average cost of terminal operation is 30 per cent of the total expense of transportation. The cost of delivery from railway terminal to consignee's warehouse is unbelievably high, being in New York about 80 per cent of the whole railway carriage from Buffalo to New York.

In view of the tremendous expense due to terminals, rates could be reduced by the pooling of terminal resources and avoiding duplication of resources. In certain sections of the country the savings in 28 cities by these means have already amounted to $5,750,000 annually.

The speaker then raised the question of how the cost of terminal operation can be reduced and discussed some considerations of the problem.

He considered the New York terminals as really the affair of the entire country. In the metropolitan district, the greatest manufacturing district of the United States and embracing one-twelfth of the population, 40 per cent of the country's exports and imports are handled. These terminals occupy valuable waterfront space which could be relieved for other purposes by a relocation and pooling of terminals. The freeing of the waterfront property would make it available for ocean-steamship terminals of which New York is in great neeed. If properly developed, the port of New York could be almost indefinitely extended and with advantages vastly superior to those now existing or obtainable elsewhere.

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Abstract of an address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For a more complete account see Mechanical Engineering, January 1921.

No. 1758 *b*

THE TRANSPORTATION AND TERMINAL PROBLEM OF NEW YORK AS AFFECTED BY THE HUDSON RIVER

By Gustav Lindenthal,[1] New York, N. Y.
Non-Member

THE Hudson River is a barrier which must be crossed daily by 700,000 passengers, 3000 freight cars and 8,000 to 10,000 vehicles. Of the passengers about one-half come through the tunnels; all the rest of the passengers, all vehicles and all railroad cars come over as they did 80 years ago, on floating equipment along a river front of 12 miles.

If the traffic crossing the Hudson River were accommodated in the same proportion as it is across the East River at least 20 additional subfluvial tubes would be required, costing around $400,000,000. The cost of a bridge erected at or below 59th Street, Manhattan, where shores are favorable, is estimated at $100,000,000.

One of the most effective means of distributing the passenger traffic at the Manhattan end of such a bridge would be a moving platform which would intersect every line, in subway, on elevated or on surface, running north and south, to which a passenger might change. Such a moving platform, at a moderate speed of nine miles per hour, would have a carrying capacity of 40,000 passengers per hour in each direction. The moving platform would be a most important and indispensable feature of the bridge across the North River.

The eight heavy railroad tracks connecting with the seven large railroad systems on the New Jersey side can be used alternately for rapid-transit cars or for through trains. If these eight tracks were connected with an elevated marginal railroad extending along

[1] Consulting Engineer, 7 Deg St.

Abstract of an address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For a more complete account see Mechanical Engineering, March 1921.

the west side of Manhattan and looped at both ends, and if this West Side railroad were connected by a subway two-track tunnel with a similar East Side marginal railroad, the transportation capacity would be 200,000 passengers per hour. There would be on the bridge alone a potential transportation capacity of 500,000,000 passengers per year, not including highway passengers.

The present number of freight cars crossing the North River averages 3000 a day or 15,000,000 tons of freight per year, including coal. This freight traffic as well as the passenger traffic is growing at the rate of over 10 per cent per year. At least 60 per cent of the food for a population of six million is brought on these railroads from the western and southern states. The capacity of the bridge and stations for freight would be sufficient to handle 2000 freight cars per day each way, which leaves a large margin for growth.

From a thorough investigation for many years, the conclusion has been arrived at that the interests of the public would be best served by a separate terminal organization acting as an agent and trustee of the Federal Government. It is unquestionable that private capital can build and operate this undertaking quicker, cheaper and more efficiently than it can be done with public funds, but the ultimate title to the structure after amortization should be held by the United States Government.

The time for preparing agreements and plans and for the construction of this gigantic work is estimated at seven years. The trustees of the undertaking should comprise the best managers of railroads, of shipping, of warehousing and of finance.

The gross revenue is estimated at over $45,000,000 which would leave after expenses of operation a net income big enough to meet all fixed charges and a substantial margin for amortization and profit. The incidental benefits in the increase of real-estate values would be experienced on both sides of the river, but the bridge would undoubtedly make of northern New Jersey a new country.

No. 1758 *c*

RAILROAD TRANSPORTATION

By Daniel E. Willard,[1] Baltimore, Md.
Non-Member

THE speaker voiced the ideal that means of transportation should be as universal and all-embracing as the means of communication, but unfortunately no single agency of transportation had been developed of such universal adaptability as the telephone. A complete and well-articulated national transportation system under present conditions can be had only by the coördination of a number of different transportation agencies. Of these the railroads are undoubtedly the most important.

The new Transportation Act provides, among other things, that in times of emergency the Interstate Commerce Commission shall have authority to treat all the cars, engines and facilities of all the railroads, regardless of ownership, in such a way as best to serve the interests of the public and the commerce of the country.

The law also lays down for the guidance of the Commission a definite rule in the matter of fixing rates and says that they shall yield a fair return on property devoted to public use, and further provides that for a period of two years from March 1, 1920, rates shall be fixed so as to yield 5½ per cent and may be fixed to yield 6 per cent. Rates have been fixed by the Commission to yield 6 per cent on a valuation of about $19,000,000,000.

If the country were fully developed, the railroad problem would be far simpler, but it has been demonstrated that at least $1,000,000,000 per annum must be provided for new equipment and facilities. Congress evidently expected that by the Transportation Act a plan had been provided by which the railroads could establish and maintain a firm credit basis sufficient to enable them to raise the new capital. It was his opinion that the Act would do this.

[1] President, Baltimore and Ohio Railroad.

Abstract of an address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For a more complete account see Mechanical Engineering, January 1921.

The speaker then discussed the provisions of the act whereby the resources of the railroads may be pooled in times of emergency and spoke of the Advisory Committee of the eleven railroad presidents formed to coöperate with the Commission.

He spoke of the poor condition of the railroad equipment at the termination of the Federal control and of the efforts of the roads subsequently to increase the miles per day of each car and the tons per car. In spite, however, of all that the railroads are doing, the equipment is inadequate.

It seemed clear to him that in the larger public interest in developing a transportation system adequate to meet the growing requirements of commerce, each suitable agency of transportation should be used to the extent that it is economically advisable. Railroad presidents, collectively or individually, are glad to coöperate with all other transportation agencies in such ways as will insure the greatest public good, because in the end all enterprises of an individual character must be tested by that rule.

No. 1758 *d*

FEEDERS FOR RAILROADS

By Charles A. Morse,[1] Chicago, Ill.
Non-Member

AFTER reviewing the burdens laid on railroads by the arbitrarily imposed rates, many times making it impossible for a road to make a sufficient return on its invested capital and pointing out the useless and expensive duplication of branch lines by competitive companies, the speaker said that the advent of good roads and the motor truck had warranted a serious consideration of the question of dismantling branch lines.

Branch lines, formerly built at an expense of $12,000 to $15,000 per mile are now built at an expense of $20,000 to $30,000 per mile, and with more expensive equipment, the cost of operation is considerably above the return in rates, so that they are now maintained

[1] Chief Engineer, Chicago, Rock Island and Pacific Railroad.

Abstract of an address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For a more complete account see Mechanical Engineering, January 1921.

by main line revenue. Where possible, such lines should be replaced by motor truck transportation, and where this is not feasible, the rates should be increased so that those using the branch railroad may bear their proper share of the burden of these lines.

The speaker then discussed the factors to be considered in abandoning branch lines and pointed out some of the accompanying benefits. This was followed by a consideration of rate making as affecting branch lines and a discussion of the extension of feeders in undeveloped territory and the rates which should be allowed in order to secure a fair return on the money invested in their construction and equipment. The construction of such lines was then considered by the speaker, who condemned the practice of the past of absorbing the expense of new construction by the parent line. Each branch line should carry all charges for labor and material used in construction, as well as all operation and maintenance charges. The speaker ended his address by discussing the operation of these branch lines.

No. 1758 *e*

THE GOVERNMENT AND INLAND WATERWAYS

By Frank T. Hines,[1] Washington, D. C.
Non-Member

FROM 1910 to 1916 but little was done by the Government for the improvement of waterway transportation aside from routine river and harbor work of the Corps of Engineers of the Army. In the latter year came the forerunner of the permanent Waterway Bureau of the Committee on Inland Waterways of the Council of National Defense.

Under the pressure of wartime conditions of 1917 this committee presented reports which were instrumental in later developing a definite governmental policy toward waterway transportation.

With the coming of the United States Railroad Administration and the actual coördination under Government control of all public

[1] Vice-President, Baltic Steamship Corporation of America.

Abstract of an address delivered at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For a more complete account see Mechanical Engineering for March 1921.

transportation agencies, the waterway problem was forced still further toward a permanent solution. The speaker then discussed the conditions existing under the régime of the Governmental administration of waterways, the extending of their usefulness, the chartering of all water craft and the construction of new equipment.

The routes are today serving as avenues for transporting thousands of tons of merchandise every month, at an average of 20 per cent below our rates for similar hauls. The most important things which the lines are doing are to awaken widespread public interest in the possibilities of water transportation, to demonstrate the classes of merchandise best suited for water haulage, to educate the shipper in the advantage of shipping by water, to indicate the necessity for waterway terminals and to pave the way for a mutually profitable basis of interchange between railroad and water carriers.

Congress has already appropriated $2,000,000 to be spent within the next year in building necessary terminals. These projects must stimulate similar construction by private capital. Once these individual lines have reached the stage where they are actually producing profits, the physical operation of the lines may well be given over to private ownership on much the same basis as the Shipping Board is disposing of its war fleets. Until that time, however, the money spent by the Government in pioneering the way toward profitable large-scale waterway transportation may be considered as an investment which will in the near future bring splendid returns.

No. 1758 *f*

MOTOR TRUCK TRANSPORTATION

By Francis W. Davis,[1] Buffalo, N. Y.
Non-Member

IN the short period of twenty years, motor highway transportation, in capital invested and diversified fields of application, has reached the point where it is warranted in seeking counsel with its elder brothers in transportation. Except for certain services, the

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motor truck will never replace waterway and railway transportation, nor be related to them in other than a supplemental capacity.

The problem has three main aspects: (1) the field of truck transportation; (2) the present status of truck transportation; and (3) the improvement of truck transportation.

It is in the short haul and at terminal points that the railways are at greatest disadvantage, and it is here that motor transportation comes as a help and releases cars for the long hauls where they are most efficient. Another field is the so-called combination service. The truck engine, used primarily for transportation, is available for auxiliary purposes and increasing use is being made of it for loading and unloading operations, the driving of winches, cranes, pumps, etc., and for miscellaneous power requirements.

An estimation of the total tons moved by truck per year is 1,200,000,000. Truck operating costs vary too widely to give figures. It is improbable that increase in size beyond the present 5 or 6 tons capacity will result in the future.

Improvement of truck transportation is dependent upon improvement in road conditions, and figures are given to show the savings due to good roads. Proper surfaces, foundations, grades, bridges and reasonable legislation for the protection of highway and motor transportation are all necessary for advance.

No. 1759

FOUNDATIONS FOR MACHINERY

BY N. W. AKIMOFF, PHILADELPHIA, PA.
Member of the Society

The purpose of building foundations is clearly defined and differs radically from that of providing substructures for machinery. Considerable contradiction can be found in blindly applying the principles of the former to the latter, and the results are sometimes so unsatisfactory as to necessitate the use of cushions and other yielding means, thus entirely undermining the very theory on which the foundation was supposedly built.

Vibrations arise from either lack of balance or from other causes. The proposed theory contemplates the latter, the problem of balancing being considered as capable of complete solution by suitable treatment. After briefly considering the nature of possible displacements of the foundation as acted upon by various causes leading to vibrations, the author introduces, by way of illustration, a double pendulum, a few experiments with which form the basis of his theory. Means for localizing the expected vibration and of controlling the resulting periods are then illustrated in a working sketch of the proposed arrangement.

THE weight of the Great Pyramid is approximately 5,274,000 tons, its base is 764 ft. square, and its height about 486 ft. It is built on leveled rock. The Washington Monument rests upon a bed of fine sand, 2 ft. thick. The piers of Brooklyn Bridge are founded 44 ft. below the bed of the river upon a layer of sand, 2 ft. thick which rests upon bedrock. The massive St. Isaac's Cathedral, Petrograd, is built on a swamp, and the piling has been so carefully proportioned that the exceedingly heavy doors of the cathedral swing easily, whereas the slightest lack of uniformity in settling would doubtless lock them. These well-known structures are here mentioned by way of illustrating the obvious purpose of the foundations upon which they rest. The object of such foundations is double: (1) To distribute the load in as nearly uniform a manner as possible; and (2)

NOTE: Patent protection controlled by the General Machinery Foundations Co., Philadelphia, Pa.

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to secure uniformity in settling, it being a well-known fact that all heavy structures settle, some to a considerable extent, 6 to 12 in. and sometimes even more.

2 Just how all this applies to foundations for all sorts of machinery, and in particular to rotative machinery, is not easy to say. Indeed, the weight, say, of a large pumping engine or of a turbo-generator outfit is generally much lower per square foot of floor space occupied than the limits prescribed by municipal laws or building ordinances, and furthermore, uniformity of settling of a relatively small volume of this nature (substructure plus engine, etc.) can be secured without going to the extremes usually observed in designing footings for buildings.

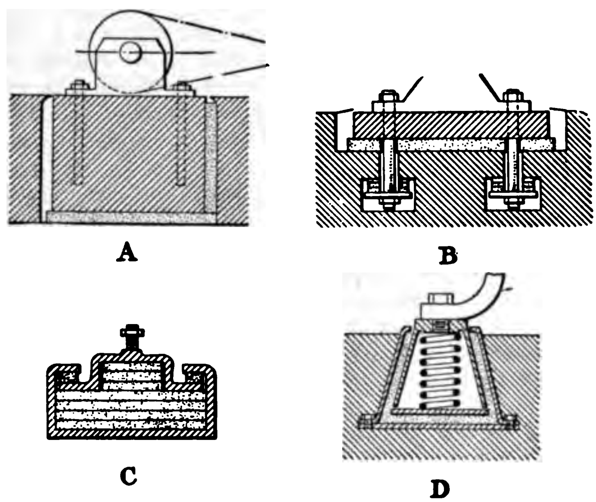


Fig. 1 Some Common Methods of Neutralizing Vibration

3 What, then, would be the general governing idea in proportioning a foundation, say, for an engine of a given type? Should it be heavy or light? Should it be deep, resting on rock or sand if possible? Should it be independent of the footings of the building, or would it be desirable to tie it to the latter?

4 By examining the existing records we can find a great variety of rather contradictory answers to each question, but the predominant idea in the mind of the designers appears to be somewhat as follows: Since the engine is likely to vibrate, let us tie it as firmly as we can to the earth itself. The mass of the earth being practically infinite, the amplitude of the resulting vibration will probably be

zero. The designer may be utterly unconscious of this reasoning, but he applies it through instinct or "horse sense" and gets results which sometimes are satisfactory and sometimes exceedingly poor.

5 It is especially interesting to see how the same designer, having decided to provide as solid a foundation as practical considerations allow, will often unconsciously neutralize his whole theory by such means as: (*a*) resting the foundation upon a layer of rubber, cork, felt, or other yielding material (see Fig. 1, *A* and *B*); (*b*) providing a space between the foundation and the walls of the building and filling it with sand (if the foundation is so heavy and so deep as to secure immunity from vibrations, why fear its contact with the walls?); (*c*) using various pads, buffers, cushions, etc. (see Fig. 1, *C* and *D*), which, if at all yielding, of course tear to the ground the very idea of solidity of the foundation with the earth.

CAUSES OF VIBRATION

6 A rational basis on which to work is thus seen to be lacking and it is accordingly the object of this paper to point out some definite lines along which a rational theory of substructures for engines and moving machinery can in general be built up. To begin with, it is important to realize that vibrations are caused by two distinct orders of agencies: (1) Those due to unbalance, or, more correctly, lack of running balance; and (2) those due to causes other than unbalance.

7 As regards unbalance, it may be said that this can be so easily corrected in the construction of machinery that all specifications should invariably call for perfect running balance at all speeds, that is, complete absence of tremor or of "periods" under all conditions. The writer feels that such a complete practical solution of this problem of balancing has been offered by him for bodies of all kinds that there is now absolutely no excuse for the manufacture of unbalanced machinery. However, there are many causes quite independent of balance, each of which is likely to result in vibrations, as for instance, "whipping" of a slender body (crankshaft, armature, turbine rotor, etc.); water in a steam turbine; peculiarities of the reciprocating mechanism, — for instance, a 4-cylinder or an 8-cylinder V-type engine where certain forces do not cancel out and where running balance alone is not conducive to perfect results; or torsional vibrations, which under certain conditions produce an effect very similar to that of unbalance. Our problem, then, is to analyze the effect of these various causes, with the view of designing

a substructure for a given machine that will be *least responsive* to these causes, for this is what the "relative" freedom from vibrations really means.

8 But whatever may be the cause of vibrations, it is safe to say that in general they are always due to *forces*, acting in a plane or planes, perpendicular to a certain axis; also to centrifugal *couples*, located in a plane, rotating about a certain line, usually the axis mentioned just above. We know from elementary mechanics that any motion of a body can be resolved into six distinctly separate motions: three *along* the three mutually perpendcular axes drawn through any point, within or without the body, and three *about* these same axes. A free body, capable of a displacement in any of these six modes, is said to possess six degrees of freedom.

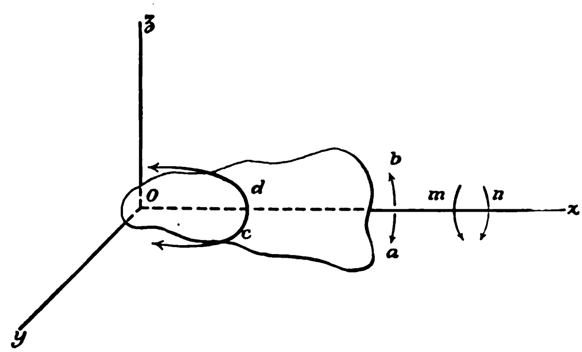


FIG. 2 DIAGRAM OF A BODY WITH ONE POINT FIXED

VIBRATION AS AFFECTED BY DEGREES OF FREEDOM

9 If the body is rigidly locked so that no displacement of any kind is possible, we say that all six degrees of freedom have been suppressed. By fixing two points in the body we have the effect of rotation about an axis, and only one degree of freedom, that is, the angular displacement about the axis, characterized by these two points. By fixing one point we suppress all bodily motion along any three axes through this point, but we still have three degrees of freedom, that is, freedom of angular displacement about any or all three axes. Fig. 2 shows a body whose point O is fixed. Such a body can have only three kinds of displacement: (1) about the axis y, as shown by the arrows a or b; (2) about the axis z, as shown by arrows c or d; and (3) about the axis x, as shown by arrows m or n.

It makes absolutely no difference what the forces are which act on the body, no other motion is conceivable.

10 On the other hand, placing a body upon a thick sheet of yielding material, or for that matter on four springs, means freedom in all six directions; and of course the same applies to cushions or pads. For this reason it appears to be of extremely questionable value to interpose layers of such material between a massive subfoundation and the foundation proper of a machine, even if isolated examples are on record where such an arrangement actually *happened* to give satisfactory results.

WHAT IS STABILITY?

11 With the foregoing in mind, let us digress for a moment. Stability with regard to our subject is a somewhat relative term. Why was the Great Pyramid built upon leveled rock, even, we are told, dovetailed therein? To insure stability. Why is a ship's compass or barometer mounted in gimbals? To secure stability. Why build a massive foundation for an engine? To secure stability. Why provide a layer of yielding material or springs or rubber pads? To secure stability. In other words, it is quite necessary to define in a more rational way the purpose of a foundation for an engine of a given type. We shall attempt to do this, first stating, however, the well-known effects of vibrations on various types of apparatus.

12 In large power plants where the main units are of the modern turbo-generating type the steam mains have been known to burst, and subsequent investigation has often revealed no defect in either material or in general arrangement of piping. The accident can thus only be explained as due to "rough" running, that is, vibrations. What, then, would be the remedy? To anchor the unit still firmer, or to place it on more yielding substance, thus encouraging the effect commonly known as "flopping around"?

13 The operation of a printing plant or of a leather-working factory is often extremely unpleasant for adjoining dwellings, and sometimes even for buildings located at a rather considerable distance. Inspection often reveals that machinery in such plants is firmly secured to extremely massive foundations and the owners are at loss as to how to remedy the trouble.

14 Within the building itself, irrespective of the effects of neighboring properties, the action of machinery is often felt and has been known to cause a great deal of annoyance and dissatisfac-

tion. In addition to the printing press and the leather-surfacing machine, the types of machinery usually apt to cause trouble are as follows: triplex pumps, ice machines, air compressors (direct-driven), and various types of purely rotative machinery, such as blowers, centrifugal pumps, etc.

15 An internal-combustion engine often exhibits a certain amount of vibration which can be felt all over the understructure. In fact, owing to violent vibrations some of the tie rods, lamp brackets, etc., on automobiles have been known to snap in two, and in aircraft some of the instruments to drop off the board, yet these were parts of the understructure, to which they, as well as the engine itself, were firmly secured. The point we wish to emphasize is that in an understructure too much rigidity is as harmful as too much freedom to yield.

STABILITY DEFINED

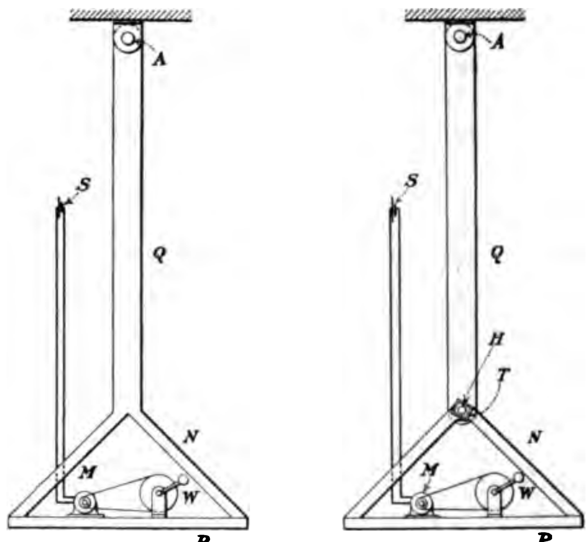
16 We are now ready to formulate the new criterion of "stability" for foundations. Stability is here characterized by remoteness of the operative speed from any one of the several synchronous speeds at which the frequency of the operative speed would be nearly, or exactly, equal to the frequency of the free oscillation of the system, if displaced from natural state of rest and let go.

17 How many distinct synchronous speeds a system is capable of having depends upon the number of degrees of freedom. An absolutely free system, for instance, placed upon an elastic sub-foundation, may have six independent synchronous speeds, or "critical" speeds, as they are sometimes called. A massive foundation resting upon rock is likewise often apt to be, in the larger sense, free in all six degrees, since it occasionally *does* vibrate and propagate the vibration to other buildings, etc. A system with one fixed point may have only three such synchronous speeds, while a system mounted to rotate about an axis can have only one such speed or "period," as it is often termed. If we could control these synchronous speeds, so as to make sure that none comes anywhere near the actual speed of operation of the machine, we would then have a fairly complete solution of our problem.

VIBRATION IN A BODY WITH ONE DEGREE OF FREEDOM

18 In order to understand clearly the foregoing as well as the broad methods here proposed, let us consider the following experiment and the general consequences manifestly derived therefrom:

Imagine a pendulum (Fig. 3-A) consisting of a platform P rigidly connected to the member Q by means of the side members N. The system is free to swing about the axis A in the plane of the figure. A small motor M fastened on the platform P operates a countershaft carrying an off-center weight W. The motor is fed through a suitable flexible connection and it is always possible to adjust the speed of the countershaft carrying the weight W so that the number of revolutions per minute of the former will be equal to the number of double oscillations per minute of the pendular system, if the latter is slightly displaced from its vertical position of equilibrium and let go. The



A B

FIG. 3 A — SYSTEM WITH ONE DEGREE OF FREEDOM; B — SYSTEM WITH TWO DEGREES OF FREEDOM, ONE OF WHICH MAY BE SUPPRESSED

effect of this adjustment of speed will be the so-called "synchronism," and the extent of swing (amplitude) of the pendular system, in general very slight for arbitrary values of the rotative speed of the weight W, will now become violent, in fact, out of all proportion to the magnitude of the weight W itself.

19 This phenomenon of synchronism of cause and effect has been well studied. The amplitude at the exact condition of synchronism should theoretically be infinite, but of course in practice resistances of various kinds are always present, so that instead of *infinite* we have *large* amplitudes. The most curious fact is that in the vicinity

of synchronous speed, both above and below, the amplitude drops down to a value almost insignificant, so that if the weight W is small the system appears to be practically at rest. Furthermore, any further increase of speed will not produce any effect, contrary to the current opinion of those not very well versed in the matter. For the sake of illustration let this synchronous speed be 100 r.p.m.

VIBRATION IN A BODY WITH TWO DEGREES OF FREEDOM

20 As a modification of the experiment let us now provide another system (Fig. 3-B) identical with the first except that the platform P is not solid with the member Q but is pinned thereto at H, the pin used being both frictionless and at the same time so arranged that it can be tightened up by means of the nut T, so as to lock the joint, thereby securing the exact effect of the rigid pendular system of Fig. 3-A. Providing the pin H in the language of dynamics is the *introduction of an additional degree of freedom*, thus securing a system of two degrees of freedom; while the tightening of the nut T, amounts to *suppressing one of the degrees of freedom*, thus converting a two-degree into a one-degree system. In experimenting we shall first deal with the system of one degree of freedom, tightening the nut T and thus converting the pendulum into a system exactly similar to that discussed above, the synchronous speed being, say, 100 r.p.m. The pendulum will oscillate violently. We now loosen up the nut T, introducing an additional degree of freedom, with the apparently surprising result that the amplitude decreases practically to zero. If we reduce the speed considerably, say to 50 r.p.m., violent oscillations of the whole system will reappear, as will likewise be the case in speeding up the countershaft, say to 150 r.p.m. These figures are purely illustrative; whether they will actually correspond to exact facts will of course depend upon the characteristics of the system.

21 In other words, by introducing an additional degree of freedom we have accomplished this double result: (1) What was synchronous speed for a system with a single degree of freedom is no longer synchronous speed for the same system provided with an additional degree of freedom; (2) the new system has two frequencies of oscillation, at which it is sensitive to disturbing influences (such as centrifugal action of the weight W), one being below and the other above the value corresponding to that of the same system with the additional degree of freedom suppressed.

22 It should especially be observed that the oscillations were thus reduced practically to zero, not by steadying the system by something without it, but by some sort of an adjustment wholly within the vibrating system itself. Furthermore, what we did was to increase in a measure the flexibility of the system by breaking it in two; and although at first glance this might have increased the effects of the disturbing agency (rotation of off-center weight W) the actual effect was practically to bring the system to rest. It is still more important, however, to note the fact that we introduced the additional degree of freedom precisely in the sense of action of the disturbing agency, that is, in the sense of the plane of the figure and not at right angles thereto; or, say, in an up-and-down sense, as for instance by providing a coil spring instead of joint H.

THE BASIS OF FOUNDATION DESIGN

23 This, then, will be taken as basis for our further discussion: In contemplating the design of a foundation we shall always separate those directions (or axes of instantaneous rotation) about which the system cannot (or at least is not likely to) oscillate from those directions (or instantaneous axes) about which the system is more or less certain to vibrate. We next shall select a "steady" point from purely practical considerations, and finally devise such means of controlling the "free periods" of the system as will secure the desired degree of remoteness from synchronism under the actual operative speed. Such means of course will be springs, exceedingly heavy, and not in the least calculated to allow of any free wabbling of the system. They will also be adjustable so that the desired periods may be readily varied within wide limits; and in general structurally arranged to introduce as few changes as possible in the arrangement as a whole.

24 The author feels that to submit too many particulars as regards the detailed designs of such an arrangement would certainly defeat the purpose of this paper, which is merely to introduce the broad idea and not any one of the scores of individual designs which might readily suggest themselves to the engineer confronted with the problem of designing a foundation for a given machine. Considering, therefore, only one type of apparatus, a turbo-generator (Fig. 4), we must start out with the selection of the steady point. We will naturally place it as near the steam main as possible (not to the exclusion, of course, of a suitable expansion joint), as under all conditions, should there be a choice of position, preference should be

given to that point as far as possible from the center of gravity of the system, so that any static unbalance (whipping, etc.), would be made to act not as a force *upon*, but as a moment *about* that steady point. Such point should actually be made as steady as possible and no trouble should be spared in providing suitable piling or digging down to the solid ground and constructing suitable footings.

25 The next problem is to design a substructure adapted to receive the bedplate of the apparatus and made stiff enough so as to eliminate any "periods" of its own. This bedplate may be made of structural steel or of reinforced concrete, in which latter case the ends thereof may be made of cast iron. The substructure is supported upon the steady point either by a ball-and-socket arrangement, or is simply bolted at that point to the floor plate underneath

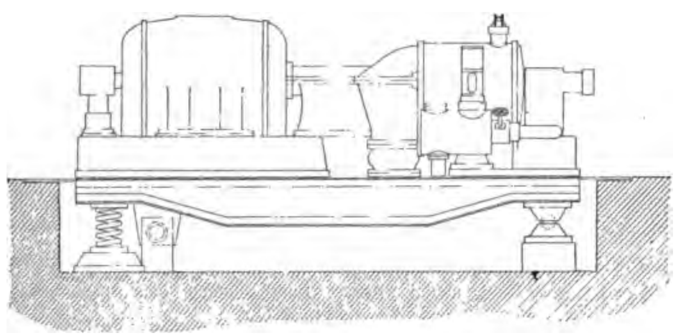


Fig. 4 Application of Author's Design to a Turbo-Generator

by a bolt which need not necessarily be very light but which must be arranged in a manner to secure the minimum area of actual contact. Remembering that in apparatus of this sort the tendency to oscillate about the axis x (Fig. 2) is always rather negligible, we have practically only two degrees of freedom and only two periods to adjust so as to have them well out of the limits of the operative speed. Hence the two sets of springs, one to take care of the period corresponding to oscillation in the vertical plane, the other to control motion in the horizontal plane. It should not be imagined, however, that these springs will necessarily be very light; they will always have considerable stiffness, but their function is that of being the only members that *can* yield and the whole situation is controlled by the proper choice of these yielding elements.

DISCUSSION

W. E. S. Dyer.[1] The paper seems to indicate a radical but likewise welcome departure from the existing views on the subject of Foundations for Machinery. The precautions as to settling have not been underestimated and apparently the foundation is considered as a part of the machine it supports and not solely an extension of the immovable understructure. The author's scheme conceives a method of providing controllable flexibility, with the opportunity to adjust the flexible conditions in order to secure desirable results.

In the customary foundation of the usual type, there is no opportunity for an adjustment, but there is a tendency to design the foundation as massive as possible and to spread it over the greatest available area.

It should not be forgotten that the vibration of a turbine must not be restrained within itself, as is the case when the turbine is erected on the usual foundation structure. Turbine operating difficulties may be traced in some instances to the rigid attachment of the turbine casing to the foundations, although the same turbine when on the shop testing table had a very satisfactory record as to vibration. Testing tables do not provide as rigid a means of anchorage as the usual concrete foundation at the final point of installation.

The author's suggested method of foundation construction and attachment provides the means of eliminating acknowledged difficulties, and the turbine manufacturers may justly demand their equipment be erected upon a support, yielding in character, but controllable as to the amount or degree of flexibility.

The design as submitted by the author would seem to be easy of installation upon existing foundation. His work in the field of vibration in general is sufficiently well known to encourage a careful study of his paper by all designers and engineers.

Maurice Deutsch.[2] Since 1906 the writer has been frequently consulted with reference to the effect of vibrations on various structures, as to cause and remedy, as well as to furnishing designs for reducing vibrations in certain structures to a minimum.

In the writer's experience there have been many cases in which the mere placing of a machine to operate along a certain axis has

[1] Mill Engineer and Architect, Land Title Building, Philadelphia, Pa.

[2] Architect and Engineer, Hudson Terminal Building, New York, N. Y.

minimized or altogether eliminated complaints of serious vibrations. On the other hand very little has yet been done in the way of placing machines or buildings on such foundations as will tend to reduce vibrations to a minimum. One of the first cases in which steps were taken, at the time of the construction of a new building, in order to prevent transmitting vibrations to adjoining property, which vibrations would arise from the operation of many machines in the new building, was that employed in the construction of the Hill Publishing Building at 10th Ave. and 36th St. In this case aside from the suggestions that were made by the writer in regard to the structural design of buildings, the mere expediency of setting the building back about one inch from the building line next to the adjoining building has avoided any complaints and would avoid any legal grounds for complaint in the future.

One of the first attempts that has ever been made in order to reduce vibrations in buildings adjoining or built over railroad tracks is that being used in the Grand Central zone. In this case the building columns were protected from vibrations by placing them on a vibration mat consisting of layers of lead, asbestos, galvanized steel, in proper relation to each other. This has given very satisfactory results.

The writer has been consulted recently in connection with two different buildings which are to be built directly over New York subway tracks. In these cases a mat properly used will tend greatly to reduce the frequency of vibrations transmitted to the steel columns. In this case vibrations resulting from the operation of trains are generally of very small amplitude but high in frequency. These frequencies vary from 19 to 120 per second, this variation depending upon whether trolley cars, heavily loaded trucks, or railroad trains are the cause. Also the condition of the roadbed and rolling stock has much to do with the nature and extent of this vibration.

In the operation of printing presses, textile, or other machinery, vibrations vary considerably. Sometimes an oscillation will be produced in the building containing the machinery as well as in adjoining buildings. These vibrations generally have amplitudes varying from 0.0001 millimeter to one millimeter, and the frequencies will vary from 6 to 50 per second for every high-speed machinery depending upon the rotative speed, and other factors. Generally speaking and comparing them with machinery and traffic vibrations, earthquake vibrations are higher in amplitude and lower in frequency, these latter generally running about a half vibration per second and less.

The writer considers the paper a valuable contribution to engineering science and one that merely scratches the surface of a subject that has far reaching possibilities in applied science.

F. G. Hechler. So far as the writer knows, the author is the first to put the subject of foundation design on a rational basis.

Foundations have usually been proportioned by rule of thumb methods, with a tendency to make them as massive as conditions would permit. Every one knows that the results were often unsatisfactory and that, in many cases, annoying vibrations were transmitted to surrounding structures, even when a considerable distance away. The fact that the earth transmits such vibrations should make it evident that mass alone can never furnish the ideal solution of the problem. This is clearly shown in a valuable article on the Transmission of Vibrations through Soil which appeared in *Zeitschrift des Vereines deutscher Ingenieure*, in the issue of September 18, 1920, an abstract of which is given in the December 1920 issue of Mechanical Engineering.

Furthermore in many cases massive foundations cannot be used, for instance where machines are placed on upper floors of buildings, or motors are mounted in the fuselage of an airplane, or in an automobile chassis. In these cases we know that, no matter how rigid we attempt to make the structure, there will in most cases be an annoying vibration at certain speeds, and in some cases at all speeds. In other words some part of the structure is in synchronism with the disturbance and responds thereto by vibrating.

Stability, as the author points out, is a relative term, and when applied to foundations, means freedom from the transmission of annoying vibration to the foundation at all usual operating speeds. This result is secured not by attempting to fix the body immovably, but by giving it perfect freedom to move in certain prescribed direction. As soon as this freedom is permitted, the body no longer exhibits any tendency to transmit vibrations to the supporting structure.

This new method of foundation design is applicable to every kind of machine from the tiny phonograph motor to the largest steam turbine. In some cases its adoption will be of importance chiefly from a mechanical standpoint, because of the increased life of the bearings, better lubrication, reduced wear and tear with consequent reduction in upkeep cost. In other cases its special appeal will be to the comfort and pleasure of the users of the machine.

A notable example of the latter is the automobile where its adoption will relieve the occupants absolutely from the annoying "periods" to which most cars are subject, and which usually occur at the speed at which one would prefer to drive.

The necessity of careful balancing of rotating parts must not be overlooked, the foundation should not be expected to overcome the harmful effects of poor balance, however much it may minimize them. Methods of obtaining perfect balance were outlined in a paper [1] presented to this Society by the author several years ago and are now in general use by the U. S. Navy and by large commercial companies.

Balancing alone will not always entirely eliminate vibrations, but good balance in conjunction with properly designed foundations will give ideal results.

C. V. Kerr. Referring to expedients for neutralizing vibration mentioned in Par. 5, an interesting case came into the writer's experience. A turbine-driven centrifugal boiler-feed pump was set on a concrete steel floor. At full speed, the resonance of the floor emphasized by the hollow bedplate produced a vibrant hum that could be heard a block away. Just filling the bedplate full of clean, dry sand so reduced the noise as to be almost negligible a few feet away; and there was doubtless a gain in efficiency with the reduction of noise. In another case it was necessary to set a turbine pump unit weighing about five tons on a concrete steel floor rather lightly supported. The bedplate was filled solid with concrete and although some noise persisted, water standing in the gutter around the cast iron showed by reflected light almost no tremor. At the same time the floor vibrated in spots enough to be unpleasant to stand on. A final remedy for such a case might be to support the floor from beneath at the vibrating spots.

It is gratifying to an engineer to observe such a development as this in the difficult art of balancing and no one will quarrel with the hope here expressed that it may continue.

S. E. Slocum.[2] The author's principle of engine suspension applies with special force to automotive apparatus, although this particular application of it is hardly mentioned in his paper.

[1] Trans. Am. Soc. M.E., vol. 38, p. 367 and vol. 39, p. 779.

[2] University of Cincinnati, Cincinnati, O.

In automotive apparatus there are at least four principal causes of vibration. The first is lack of balance of rotating parts, such as crankshaft and flywheel, which as the author points out, may be entirely eliminated by proper methods of balancing.

The second consists of inertia forces arising from reciprocating parts, which are difficult to counteract without making the motor too complex to be practical.

The third comprises torsional vibrations, which produce what Lanchester has called torque recoil. Any unevenness of torque results in torsional vibrations, and consequently such vibrations are present in every motor. Of course increasing the number of cylinders tends to smooth out the torque curve. On the other hand, any difference in the gas mixture delivered to the various cylinders tends to produce unevenness of torque, and the same effect results from unequal firing intervals such as occur in a 60-deg. or 45-deg. V-type motor.

The fourth cause is synchronism between the speed of the motor and the natural frequency of vibration of the chassis. This cause is undoubtedly the one most in evidence in automotive work, and accounts for the periods which are so noticeable in every automobile. It may be well to note here that synchronism is an intensifying cause and not an exciting cause, like unbalance and torque recoil. Whenever the speed of the motor synchronizes with the natural frequency of vibration of the frame or body, this part is set in violent vibration, and the result is one of the so-called periods which are so annoying to the driver and passengers. This effect is unquestionably strongest in a closed car, as the air confined in the body is set in vibration, and the effect of course is more pronounced than if the vibration were only transmitted through the seat.

Now if we dispose of the first cause of vibration by balancing, there still remain at least three other causes, one of which, namely synchronism between motor speed and the natural frequency of the frame or body, is mainly responsible for the unpleasant features to which the buyers of automobiles object.

The author's method of suspension, which consists in mounting the motor with three-point support, one point only being rigid and the other two resilient, evidently makes it practicable to destroy any possibility of synchronism between motor and frame. That is to say, if the resilient supports are so designed that their natural frequency of vibration will never correspond to any ordinary motor speed, it is evident that there can be no vibration of the frame due to synchronism with the motor, as this possibility is cut off at the supports.

Furthermore, by this same means, the other causes of vibration in the motor will also be prevented from transmitting their effects to the frame. In other words, this principle of suspension virtually short circuits the vibration of the motor by insulating the motor from the frame so far as regards any periodic disturbance in the nature of a vibration.

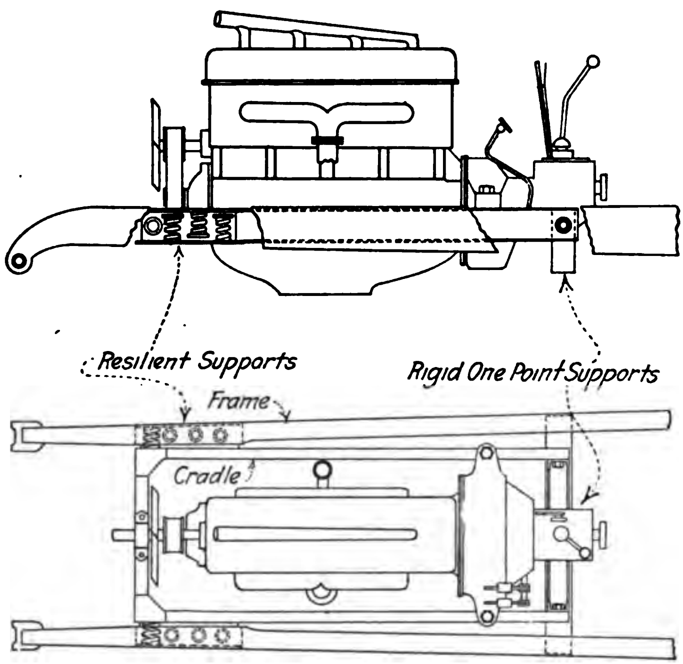


Fig. 5 Proposed Resilient Support for Automobile Motor

Fig. 5 is a suggestion as to how this may be applied in automotive work. Without making any radical changes in either crankcase or frame, the motor may be mounted in a cradle as indicated, and this cradle supported at three points on the frame, the single rigid point being some sort of a universal joint supported on a cross member and placed directly under the transmission, and the two resilient supports, consisting of both vertical and transverse springs, interposed between the forward end of the cradle and the frame. This method of mounting will take care of both vertical and horizontal vibrations due to inertia forces and unbalances, as well as to torque recoil, while by selecting springs such that their period does not fall within the limits of ordinary motor speeds, it will also be effective in preventing synchronism between motor and chassis.

With the increasing demand for comfortable riding qualities in pleasure cars, any feature which will increase the satisfaction of driver and owner is bound to commend itself to the manufacturer and dealer. The author's principle of suspension is entirely novel, and its practical possibilities are such that it should attract the attention of every one connected with the automotive industry.

T. A. Bryson. The author is to be congratulated upon his study of this problem for it is desirable that an effort be made to rationalize the design of machine supports and other structures with regard to the elimination of harmful vibration. With this in mind the writer is prompted to offer some criticism of the statements and conclusions of the author.

In Par. 9 is discussed the effect upon vibration of the number of "degrees of freedom." It is obvious that the six degrees of freedom refer to one set of instantaneous axes mutually perpendicular and intersecting at one point. With regard to the fixing of one or more points, it must be assumed when dealing with engineering materials that fixed means comparatively fixed, it being impossible to obtain a perfectly inelastic and immovable point of attachment. Therefore we must consider that every body, irrespective of the mode of support, has six degrees of freedom and will have six critical speeds. The term "degree of freedom" while probably suitable in purely mathematical discussion becomes misleading when dealing with the actual materials encountered in construction. Whether or not all or any of the critical speeds will be encountered depends upon the inertia of the body and the degree of restraint offered towards any one of the motions in question.

It is therefore suggested that the term degree of freedom be abandoned and in considering any particular critical speed we consider the degree of restraint offered to the resulting motion. In this term the word degree has its usual significance. It is also in accord with the theory that every element of a system is governed by the same condition, that is, it is subject to independent vibration and its periods of vibration depend upon the inertia of the element and upon the degree of resistance to motion offered by its supports.

We may now examine the systems illustrated in Fig. 3. Fig. 3A is a system of one element (neglecting the machinery from which impulses emanate) fixed to such an extent that at reasonable speeds only one critical speed may be encountered. Fig. 3B with the nut loosened is not, according to theory, a system with two degrees of freedom but a system of two elements each having a low degree of restraint in one direction. These elements are of different mass and shape from each other and from the original structure and therefore it is to be expected that there will now be two critical periods each different from that of the original structure. If point *A* of Fig. 3A had been changed to a ball joint there would have been additional freedom given in the sense of the author's definition. The writer is inclined to believe that in this case the synchronous or critical speed in the plane of the figure would remain the same. For this reason it would seem that the conclusions drawn in Par. 21, not only are not justified by the preceding demonstration, but are open to question.

Since the operative speed is usually fixed, let us invert the author's definition of stability and say: Stability is characterized by the remoteness of every one of the critical speeds or natural vibrations from the operative speed. It follows that by the proper selection of elastic supports the restraint with reference to all six possible motions may be made of such degree as to locate every one of the critical speeds sufficiently remote from the operative speed. The resulting structure will then, by definition, be stable. Apply this reasoning to Fig. 1B and consider the motion of rotation of the bedplate about the axis perpendicular to the paper and also the motion along a vertical line in the plane of the paper. The pressure upon the elastic members, or in other words, the amount of tension to which the bolts are subjected is a measure of the degree of restraint and therefore there is available means for varying this tension or restraint and for locating these two natural periods or critical speeds sufficiently remote from the operating speed.

It would seem then that any lack of success in the use of such materials as rubber, cork, felt, etc., or springs, is not due so much to the material but to improper application. Further, it seems possible to avoid the expense of providing even one relatively rigid point in the bedplate if the mounting illustrated in Fig. 1B is applied to control each of the six possible motions.

George W. Klinger.[1] The principle developed in this paper should be a valuable asset to the aeroplane designer. Vibrations in

[1] 1534 Winter St., Philadelphia, Pa.

a plane are terrific, in fact they are so violent that it is often impossible for the delicate instruments necessary to an aviator to function properly, aside from the snapping of metal and wood parts apparently due to vibrations.

The causes of vibrations in an aeroplane are numerous, the most disturbing of which are: lack of static and dynamic balance in the crankshaft and propeller; non-cancellation of the reciprocating parts in some types of engines; torsional vibrations; deflection or whipping of the crankshaft; and imperfect alignment. Most designers of crankshafts for aeroplane motors are too willing to sacrifice strength and rigidity for lightness in weight, whereas a few pounds more of metal properly placed would produce an ideal shaft. In some types of meters, if a fairly strong shaft together with its propeller are properly balanced, vibrations can be entirely eliminated.

The new principle set forth by the author is capable of suspending the motor in such a way that no vibrations can be transmitted to any part of the plane. The paper treats of various degrees of freedom, in the correct directions, with the use of springs, which, when applied to an object will make it stable. In other words, synchronism or the critical period between the motor and the rest of the plane would be at a speed so low that the plane would be incapable of operating, and this stability would be maintained throughout the entire working range of speeds.

Recently some interesting and important tests were made with an aeroplane, while resting with three points in contact with the ground and securely fastened so that it could not take flight, a certain maximum engine speed being developed. After this same plane, with the motor adjustments untouched, had taken flight, the maximum engine speed developed was several hundred revolutions per minute less. It is generally known that a plane is a very flexible object in itself, but when resting on its three points, on something rigid like the earth, it is not capable of vibrating to its greatest amplitude. Some of the vibrations are dampened and some do not appear at all. Now release the plane and permit it to take flight. It is no longer in contact with the earth, therefore its entire flexibility can readily respond to the disturbing forces and vibrate accordingly. This increased vibration naturally consumes some energy, and since the motor is the only source of power, the energy would be taken from the motor and thus the tendency to reduce its power and speed.

Webster Tallmadge. The tendency in turbine design is so to arrange the metal in the spindle as to bring the critical speed far above the normal running speed, but it is entirely possible to have a vibration at the normal running speed due to a critical speed, of such structures as the foundation or parts of the building The author gives, in other papers he has written, a formula of 188 divided by the square root of the deflection as being the critical speed. It would be interesting to have him explain the method of determining this deflection for columns that stand vertically.

A. A. Adler said that the paper interested him as he had made mathematical investigations of vibrations. He thought that the author had not clearly brought out the point that it was practically impossible to eliminate the causes of vibration in engineering designs. The author's solution of a single rigid point, all others being on springs was simple enough, but introduced new elements into the design which would have to be considered.

W. C. Marshall. Nothing has been brought out in the paper or in the discussion in regard to the application of the principles set forth for dampening the vibrations of marine engines. Such vibration is in many cases severe and the engine must be securely fastened to the boat in order that it be safe in a seaway. Movement in the foundation cannot be allowed in a marine installation on account of the pitching and rolling which occurs at sea. Have the principles by which marine engines may be supported on foundations of this kind been worked out?

F. A. Stanley. During the war considerable experimentation was done on a spring foundation for auxiliary machinery with a view to eliminating vibrations. Various degrees of freedom were had and various degrees of control; and it appeared that the more the freedom was restricted, the less was the vibration. From that it was argued that for the time the rigid foundation was the better. This is a question, however, which should be given much study by marine engineers, and probably the results will prove advantageous.

H. C. Dickinson. One of the most important phases of the question under discussion seems to be the nature of the forces which can be transmitted from a moving machine to its foundation. There are two elements which should be considered and which have not been brought out clearly in the paper. One is the motion of any particular machine with reference to its foundation and the other is the force which can be transmitted to the foundations. The author's method, if the amplitude be kept down, reduces vibration to a force which can be eliminated or limited by the stiffness of spring supports.

MAYO D. HERSEY. Is there any objection to having a damping attachment in addition to the spring adjustment? The use of a damping arrangement, such as a layer of felt or rubber, could hardly change the periodicity of the parts involved but might suppress some of the forces transmitted.

Has the inverse of this important problem been considered, not that of designing a foundation to suppress vibration arising from the machine, but of preventing the transmission of vibrations to the apparatus from its surroundings? Such a problem is presented in designing instruments for aircraft.

For example, even if the vibrations arising from such a machine as a steam turbine are not transmitted to the foundation, is there not danger of trouble through resonance of the springs with vibrations arising elsewhere?

THE AUTHOR. With regard to T. A. Bryson's remarks, the term "degrees of freedom" is sanctioned by universal usage, in Dynamics, for over 100 years. For this reason no one can seriously consider changing it to "degrees of restraint," or to anything else.

The author would also point out that unlocking the joint *H*, Fig. 3A, means *two* degrees of freedom, with *two* (normal) critical speeds, as Bryson will readily find out by building this very elementary model, or by reading up a bit upon this subject; a ball joint at *H* would mean *three* distinct "critical" speeds.

To Webster Tallmadge the author would reply that the formula referred to by him has reference to small oscillations of a *weight on a massless spring* only; for vibrations of vertical columns he should look up the first volume of Lord Rayleigh's Theory of Sound.

To A. A. Adler the author would point out that in some cases (steam turbine rotors or other stiff bodies) it is both possible and easy to eliminate all causes of vibrations; the trouble arises only if the rotating bodies are flimsy, like some of the crankshafts or electrical rotors, where windings shift under load, etc.

W. C. Marshall's point is well taken; of course it is of great importance to eliminate all vibration on board a ship. The author is working on this subject at present.

Professor Hersey's first question calls for an answer in the affirmative; the object of the proposed foundation design is not to eliminate the amplitude of vibration but to reduce it to a negligible amount by keeping away from synchronism as far as possible. But a frictional damper would eliminate even that reduced amount.

As regards the inversion of the problem, i.e., the idea of using the foundation as a means for preventing a vibration from outside sources reaching that which is mounted upon the foundation, this of course could be accomplished by making the period of the understructure very slow compared with that of the exciting vibration. An ordinary seismograph is built on this very principle; the stylus seems to vibrate and make a record upon stationary paper; as a matter of fact it is the paper that vibrates (together with the earth, the support, etc.) while the stylus itself remains practically steady.

No. 1760

RATIONAL DESIGN OF HOISTING DRUMS

By Everett O. Waters, New Haven, Conn.
Member of the Society

The main problem of hoisting-drum design resolves itself into two parts: determination of flange shape and thickness, and determination of the thickness of the drum body. Other factors, such as brake, clutch, and bearings, have already been investigated and are therefore not considered in this paper.

The correct proportioning of drum flanges is obviously dependent upon the side pressure brought to bear against them, by the coils of rope wound about the drum. In other words, flange thickness is a function of (a) rope tension and (b) depth of winding. Previous theoretical investigations have recognized these factors, but have failed to take account of the friction between adjacent coils of rope, and between rope and drum, which tends to hold the coils in position without the aid of the flanges, and have also disregarded the flattening of the inside coils under pressure from the outer layers. This flattening, which is quite appreciable on account of the soft core of hoisting rope, results in a shortening of the inner coils, thereby relieving them of a part of their tension, and reducing the pressure against the flanges.

A theoretical formula is deduced for the total pressure against a drum flange caused by the winding-on of rope to a given depth and under a given initial tension. Two other formulæ are then deduced, which relate this total pressure to the flange thickness and the maximum allowable tensile and shearing stresses in the material. By means of these formulæ a flange of the usual straight-sided or the mushroom type may be designed to withstand safely the pressure of the rope wound upon the drum.

The drum body is subjected to combined stresses, since it is under compression from the coils wound upon it, under tension due to the lateral pressure against the flanges, and under combined torsion and bending caused by the load which is to be hoisted. Formulæ are derived which may be used to obtain the correct thickness both at the center of the drum, where the normal stresses are greatest, and at the ends of the drum, where the shearing stresses are most prominent.

In order to check the theoretical formula for flange pressure, tests were made with small-size rope wound upon a special drum, which was so arranged that the side thrust against one flange could be directly measured while the rope was being wound on or unwound. These tests show conclusively that formulæ for flange pressure which do not take account of rope friction and flattening give excessively large results.

Presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

SOMEWHAT over a year ago the writer's attention was called to certain details in the design of rope-winding drums as commonly used in hoisting and conveying machinery. Upon investigation it appeared that little if any study of this subject had been made beyond the private researches of various firms engaged in manufacturing hoisting apparatus. Published matter dealing with this class of machinery makes scanty reference to the design of hoisting drums, and recommends proportions based on careful guesswork or previously accepted practice rather than upon theory or experiment.

2 Such being the case, it seemed well worth while to make a special study of the hoisting drum, with a view to deducing some logical rules for proportioning the important parts. Such rules, if applied in practical design, would serve on the one hand to prevent drum breakages with their disastrous consequences, and on the other hand would reduce needless weight and complexity of form. There are in reality only two such parts that require this analysis: the drum body or core, and the flanges. Other details, such as length of bearing, size of arms connecting the drum core with the hub at either end, and proportions of brake and clutch surfaces, present no novel problems; a discussion of these parts is accordingly omitted from this paper.

DESIGN OF DRUM FLANGES

3 In many installations of hoisting machinery the length of rope to be handled is not great and it is quite possible to wind it in a single layer on a drum of moderate diameter and length. This is of course the most desirable arrangement, since it insures uniform hoisting speed, obviates the danger of overwinding at either end of the drum if the rope is improperly guided, and prevents the wear on the individual wires which results when two or more coils are wound on the drum in contact with each other. However, it is often necessary to wind great lengths of rope on comparatively small drums, and in such cases the end flanges must be relied upon to support the rope at the ends of the layers, of which there may be as many as 14, 16, or more. In addition these flanges are usually subjected to a stress from brake, friction clutch, or both. It is therefore necessary to know

(*a*) the maximum tangential load on the flanges caused by the brake,

(*b*) the maximum axial or tangential load due to the clutch, as well as

(*c*) the load due to the axial thrust of the several layers of rope.

THEORETICAL DETERMINATION OF AXIAL THRUST

4 *First Method.* The first two of these loads can be found quite readily, as explained in treatises on brakes and clutches. The third is much more difficult to determine, and it has been suspected for some time that the theoretically obtained values for this thrust or pressure do not agree with the facts. The series of tests described in a later paragraph were made for the express purpose of throwing light on this point.

5 In deriving a simple formula for the axial thrust, it is assumed that the coils of rope are wound uniformly, the coils of each successive layer resting in the spaces between coils of the preceding

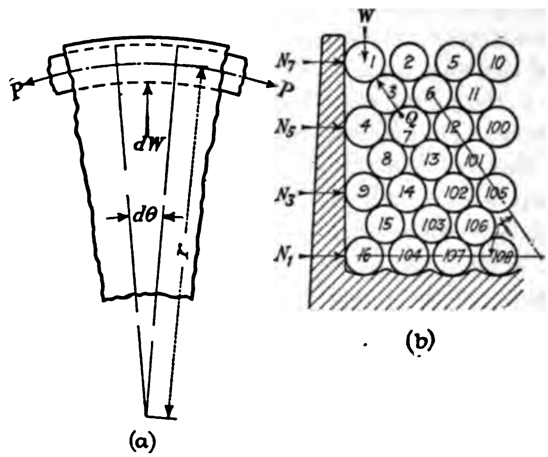


FIG. 1 DIAGRAM OF COILS WOUND ON GROOVED DRUM

layer, as shown in Fig. 1. Friction between contiguous coils and between rope and drum surface is neglected. If the drum is grooved, as shown in the figure, all coils in the central part of the drum (those numbered 100 and higher in the figure) will be self-supporting, while a certain number of coils, occupying a space of wedge-shaped cross-section at each end of the drum, must rely upon the flanges for their support. Thus in Fig. 1 the coils numbered 1 to 16, inclusive, are the only ones which will cause axial thrust against the left flange. This thrust, which is indicated by the forces N_1, N_3, N_5, etc., in Fig. 1, is computed from the following formulæ:

When there are $2m - 1$ layers of rope,

$$N_{2m-b} = bW \cot\gamma = 2\pi bP \cot\gamma \quad \text{[1a]}$$

$$N_1 = (m-1)W \cot\gamma = 2(m-1)\pi P \cot\gamma \quad \text{[1b]}$$

When there are $2m$ layers of rope,

$$N_{2m-b} = (b+1)W \cot\gamma = 2(b+1)\pi P \cot\gamma \quad . \; . \; . \quad [1c]$$
$$N_1 \quad = (m - \tfrac{1}{2})W \cot\gamma = (2m-1)\pi P \cot\gamma \quad . \; . \quad [1d]$$

6 The total axial thrust for an odd or even number m of layers is approximately

$$N_1 + N_3 + N_5 + \; . \; . \; . \; + N_m = N = \frac{m^2 - 1.5}{2}\pi P \cot\gamma \quad . \; . \quad [2]$$

7 *Second Method.* The foregoing expressions are easy to use but in the writer's opinion give results which are excessive, this being corroborated by the experiments described toward the end of the paper. Three important factors have been disregarded: friction; cross-over of the rope from a given space between coils on the preceding layer to the second space beyond, due to the fact that alternate layers are wound right and left hand; and compression of the internal layers. All three of these tend to lessen the axial thrust. An approximate expression for this thrust, as modified by these conditions, may be found by considering the end coils as a solid wedge *abcd*, Fig. 2, upon which forces are acting as indicated. The effective slope of this wedge is not the actual slope γ of the center lines of the coils, but is $90° - \gamma$, because that is the actual inclination of the contact surfaces between contiguous coils. The coefficient of friction μ is probably higher for rope on rope than for rope on drum. For example, some experiments performed by the writer gave 0.201 for the friction of rope on rope and 0.181 for rope on drum, these results having been obtained by wrapping a short length of $\frac{1}{4}$-in. rope around a drum, hanging a known weight from one end of the rope, and then suspending enough weight from the other end to just cause slipping. The rope was bright, but not lubricated, except with the grease applied in manufacture; the drum was finish-turned and unlubricated. In view of the closeness of these results, a single value for μ has been assumed in the following analysis.

8 The radial force R_1 is indeterminate; it is therefore assumed equal to $1/p$ times the vertical component of all forces acting along dc and cb, where p is the number of coils between a and b. For example, if there are six layers as in Fig. 2,

$$R_1 = \frac{1}{3\frac{1}{2}}\left[R_1 + Q(\sin\gamma + \mu\cos\gamma)\right]$$

$\Sigma(W)$ is the sum of the radial components of the tensions on all the coils in the wedge; in other words, $\Sigma(W) = 2\pi\Sigma(P - P')$, where P is the initial tension in each coil and P' the tension loss in the same coil due to the pressure from outer layers.

9 Taking the sum of the horizontal and vertical components of all the forces shown in Fig. 2, and eliminating R_1 and Q, the following equation for the end thrust against each flange is obtained:

$$N = \frac{(p - \mu^2 - 1)\cos\gamma - \mu p \sin\gamma}{(1 - \mu^2)p\sin\gamma + \mu(2p - \mu^2 - 1)\cos\gamma} 2\pi\Sigma(P - P') \quad . \ . \ [3]$$

This equation is to be considered as a refinement of Formula [2], in which no account was taken of μ or P'. It is interesting to note

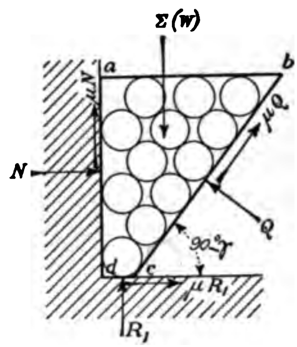


FIG. 2 DIAGRAM SHOWING FORCES ACTING IN ROPE WOUND ON DRUM

what widely divergent results the two formulæ give, especially when the number of layers is large.

10 The tension loss P', which to the best of the writer's knowledge has hitherto been disregarded or avoided in all published works dealing with the design of hoisting drums, deserves some discussion at this point. As each successive layer is wound on the drum, the preceding layers are flattened out to a certain extent; this reduces the actual length of each flattened coil and thus relieves it of a part of the initial tension P which was applied by the hoisted load at the moment when the coil was wound upon the drum. Suppose, for example, that a $\frac{1}{4}$-in. rope is being wound upon a 10-in. drum under a tension of 1000 lb. First, the bottom layer is completed, each coil having the initial tension of 1000 lb. Then the second layer is wound on, each coil of which has a tension of 1000 lb. But at the same time

the tension in each coil of the first layer drops to 700 lb. When three layers have been completed the outside layer is under the initial tension, while the tension in the second layer has fallen to 500 lb. and that in the first layer has dropped from 700 to 600 lb. With four layers the effect is still more pronounced, the respective tensions being 1000, 400, 250, and 500 lb. In general, the maximum compression occurs near the middle layer; the inner layers are prevented from much shortening by the unyielding surface of the drum, while the outer layers, being near the surface, are acted upon by comparatively small radial forces.

11 It is possible to determine the reduction in tension in each layer, and hence the value of $\Sigma(P - P')$, by a simple analysis of the mechanical principles involved. Unfortunately, however, as the number of layers of rope increases the number of unknown quantities increases correspondingly, and the equations which accumulate can only be solved after much laborious computation. But by making the assumption that the radii of all the layers are approximately equal, the formulæ are greatly simplified. Table 1, based on this assumption, gives values of P' for each layer, in terms of the initial tension P, for windings of a number of layers.

12 The quantities E' and m involved in the expression for h in Table 1 deserve a word of explanation, since they represent properties of the rope which determine the extent to which it will yield under lateral pressure. E' may be termed the "lateral modulus of elasticity," and defined as the ratio of lateral pressure per unit length of rope to the decrease in rope diameter measured along the lines of action of the pressure. On account of the non-homogeneous character of wire rope, E' bears no direct relation to the true modulus of elasticity E, and is best determined by test. The writer obtained the value $E' = 9260$ lb. per sq. in. for new $\frac{1}{4}$-in., 6 by 19 hoisting rope with hemp center. Obviously rope with steel center would give higher values of E', while cotton-centered or 8-strand rope would probably give lower values, on account of the greater softness of the core. The coefficient m is a factor of lateral deformation, or ratio of the increase in rope diameter along one axis to the decrease along a perpendicular axis, the lateral pressure causing this deformation being assumed to act along the latter axis. Its value is probably less than 0.3, the value for homogeneous steel, since the pressure which causes deformation is applied at isolated points instead of over the entire surface.

13 An example will best illustrate the use of Table 1 and

Equation [3]. Suppose that it is desired to find the lateral thrust caused by five layers of $\frac{3}{4}$-in. rope, 6 by 19, wound upon a 30-in.

TABLE 1 VALUES OF P' FOR VARIOUS LAYERS IN TERMS OF THE INITIAL TENSION P

	1 Layer					2 Layers				
h (See Note)	0.6	0.7	0.8	0.9	1.0	0.6	0.7	0.8	0.9	1.0
P_1'/P	0.00	0.00	0.00	0.00	0.00	0.30	0.35	0.40	0.45	0.50
	3 Layers					**4 Layers**				
h (See Note)	0.6	0.7	0.6	0.9	1.0	0.6	0.7	0.8	0.9	1.0
P_1'/P	0.13	0.13	0.13	0.13	0.13	0.05	0.04	0.04	0.03	0.03
P_2'/P	0.56	0.62	0.67	0.71	0.75	0.21	0.20	0.19	0.18	0.17
P_3'/P	....	....	....	....	....	0.62	0.67	0.71	0.75	0.78
	5 Layers					**6 Layers**				
h (See Note)	0.6	0.7	0.8	0.9	1.0	0.6	0.7	0.8	0.9	1.0
P_1'/P	0.02	0.03	0.01	0.01	0.01	0.01	0.00	0.00	0.00	0.00
P_2'/P	0.08	0.07	0.06	0.05	0.04	0.03	0.02	0.02	0.01	0.01
P_3'/P	0.23	0.22	0.20	0.19	0.17	0.08	0.07	0.05	0.05	0.04
P_4'/P	0.64	0.68	0.72	0.75	0.78	0.23	0.22	0.20	0.19	0.17
P_5'/P	....	....	....	....	.....	0.64	0.68	0.72	0.75	0.78

NOTE. — The value of h is given by the formula

$$h = \frac{aE}{E'r^2}\left(\frac{\gamma}{45°} - \frac{90° - \gamma}{45°} m\right)$$

where a = area of wires in rope
E = modulus of elasticity of rope
E' = transverse modulus of elasticity of rope
m = coefficient of lateral deformation of rope
r = average radius of coils under consideration

drum under a tension of 7000 lb. Pitch of coils parallel to axis of drum = $\frac{13}{16}$ in. Assume that $\mu = 0.15$, $E = 12{,}000{,}000$, $E' = 9000$, and $m = 1$. With the given pitch, γ is found to be 57° 1′; $r_1 = 15\frac{3}{8}$ in., and $r_4 = 17.26$ in., hence $r = 16.31$ in. The area of the wires in a rope of this construction is 0.199 sq. in. The number of coils in the "wedge" which bears against each drum flange is 9, and the number of those which lie in the outer layer is 3, that is, $p = 3$. Making the proper substitutions in Equation [3], we have

$$N = \frac{(3 - 0.0225 - 1)0.544 - 0.15 \times 3 \times 0.839}{3(1 - 0.0225)0.839 + 0.15(6 - 0.0225 - 1)0.544} 2\pi\Sigma(P - P')$$
$$= 1.53\Sigma(P - P').$$

In order to evaluate $\Sigma(P - P')$ we have recourse to Table 1. From the formula given in this table, $h = \dfrac{0.199 \times 0.533 \times 12{,}000{,}000}{9000r_a^2} = 0.6$, 0.58, 0.55 and 0.53 for layers 1, 2, 3, and 4. Then, when two layers have been wound on,

$P_1' = 0.3P$

For three layers .... $P_2' = 0.55P$

$P_1' = 0.13P$

For four layers .... $P_3' = 0.6P$

$P_2' = 0.21P$

$P_1' = 0.052P$

For five layers .... $P_4' = 0.6P$

$P_3' = 0.23P$

$P_2' = 0.085P$

$P_1' = 0.021P$

The total tension on the various layers of the wedge will be as follows:

Outer layer .....$3P$

Fourth layer ....$2(P - 0.6P) = 0.8P$

Third layer.....$2(P - 0.23P - 0.6P) = 0.34P$

Second layer....$P - 0.085P - 0.21P - 0.55P = 0.16P$

First layer ......$P - 0.021P - 0.052P - 0.13P - 0.3P = 0.5P$

Aggregate tension on all the coils in the wedge = 33,600 lb. (Compare this with 63,000 lb., the aggregate tension when flattening of the coils is not taken into account.) Hence the total end thrust = $1.53 \times 33{,}600 = 51{,}400$ lb.

14 Equation [3] tells nothing about the distribution of the axial thrust over the surface of the flanges. According to the first method for determining it, where friction and coil flattening are disregarded, the distribution over a small sector is as indicated by the line ABC, Fig. 3. The effect of friction is probably to reduce the abscissæ of the shaded area a constant amount, to such a line as DEF. But the aggregate effect of coil flattening is much more pronounced near the drum body, so that when both friction and coil flattening are considered, the pressure is in all likelihood distributed with a fair degree of uniformity over the sector of flange. This is indicated by the line GHI. The experiments described later point toward the same conclusion; and in the absence of more definite

information a uniform distribution of load over a given sector will be assumed in determining the strength of the drum flange.[1]

EFFECT OF UNGROOVED DRUMS ON AXIAL ROPE PRESSURE

15 Thus far it has been assumed that the surface of the drum body is grooved. This is undoubtedly the best practice, since it insures the proper spacing of coils, tends to make the winding of all the layers uniform, and reduces wear. In addition it makes the coils of rope in the middle of the drum self-supporting, without the aid of the drum flanges. A fair parallel exists in the case of a pyramid of smooth balls resting on a plate of glass. If the pyramid is to be prevented from falling, a support of some kind must be placed all around the lowest layer of balls; but if each ball in this lowest layer

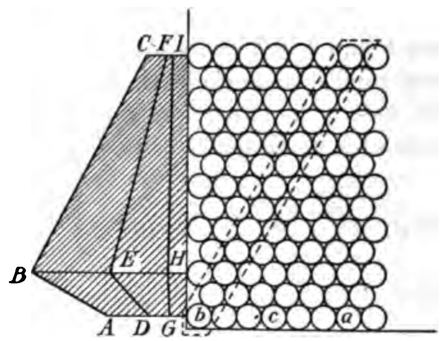


Fig. 3 Diagram Showing Distribution of Axial Thrust over Surface of Flange

rests in a slight depression or pocket, instead of the plane surface of the glass, the pyramid will stand without any support around its base.

16 With the ungrooved drum, the tendency of the bottom coils to side-slip is counteracted partly by friction, and partly by the fact that a number of coils in the middle part of the drum are "balanced"; that is, the pressure from the outer layers which tends to slide one of these coils to the left is counteracted by an equal and opposite pressure tending to slide the coil to the right. In Fig. 3 the coil *a* is thus balanced; but all the bottom coils to the left of *a*

[1] By "uniform distribution over a sector" is meant a distribution in which (load per unit area) multiplied by (distance from area to drum axis) is constant. This should not be confused with the common meaning of uniform distribution, as used in connection with beams, floors, etc.

receive an excess of pressure from the right which tends to slide them to the left. This means (remembering that all coils in a layer are in contact with each other) that an additional load in the nature of a shearing force is placed upon the drum flange.

17 Coil b, Fig. 3, presses against the flange with a force $(m - 1)\pi P \cot\gamma$, neglecting friction, compression of coils, and the pressure of the coil immediately to the right. This force is caused by the coils included in the oblique strip indicated by dotted lines. Denoting by $\Sigma(P')$ the total tension loss in these coils due to compression, and taking account of the friction between coil b and the drum body and flange, the net pressure of this coil against the flange is

$$N_1 = \pi \left[(m - 1)P - \Sigma(P')\right] \frac{\cot\gamma - \mu}{1 - \mu^2}$$

The coils between b and a will cause a pressure against the flange, of similar form but smaller magnitude, the decrease being due to the partial balance of these coils. Therefore it may be assumed that the total shearing force on a drum flange caused by non-grooving of the body is

$$\Sigma(N_1) = \frac{\pi n}{2}\left[(m - 1)P - \Sigma(P')\right]\frac{\cot\gamma - \mu}{1 - \mu^2} \quad . \ . \ . \ [4]$$

n being the number of unbalanced coils in the first layer.

RELATION BETWEEN FLANGE THICKNESS, LOAD, AND STRESS IN FLANGE

18 No discussion of the loads caused by brakes or clutches need be given here, since these are quantities which can be determined quite easily from the dimensions of the bands, links, levers, etc., by means of which these forces are brought into play. The important point to be borne in mind by the designer is that brake and clutch loads on the flange should be based on the maximum forces which the operator can obtain with the leverages at his command, rather than upon the forces required to start or stop an assumed hoisting load in a given period of time. Hoisting machinery as a class is subjected to sudden and severe overloads, and the limit of capacity is set, as a rule, by the muscular strength which the operator has at his command in the manipulation of his control levers. In general, there is a tangential force S tending to twist the flange around the drum body, and giving rise to a simple shearing stress; also an axial force C,

opposed to the axial force of the coils of rope, and producing tensile, compressive and shearing stresses in the flange.

19 The problem of deducing a relation between load and stress in the flange is no easy one to solve, owing to the diverse forms of flanges which are embodied in standard designs, and the use of stiffening ribs, wide rims, etc. After considerable study the writer selected the mushroom type illustrated in Fig. 4 as being the simplest and at the same time the most widely used. The load-stress-flange thickness formulæ were then derived by first obtaining a relation between the radial and tangential strains at any point in the flange and the coördinates of that point. This gives a relation between the radial, tangential, and shearing stresses at the point and the coördinates of the point. By a single integration the total shearing

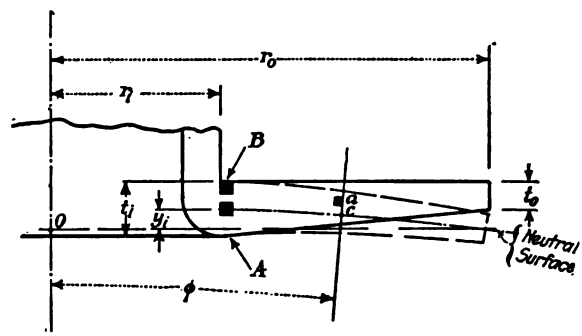


FIG. 4 DIAGRAM FOR STUDY OF STRESSES IN FLANGE OF HOISTING DRUM

stress on a cylindrical section of the flange containing the point was found; this was set equal to the external load acting on that part of the flange between the cylindrical section and the rim; and after successive integrations the deflection of the point was obtained. By proper substitution the true radial, tangential, and shearing stresses at the point could then be found in terms of known quantities, i.e., the flange thickness, inside and outside radii, and external load. By giving this general point certain specific locations, a number of formulæ were deduced.

20 In determining the constants of integration the following boundary conditions were assumed: The shear at the surface of the flange is zero; the radial stress at the rim of the flange is zero; the slope of the deflected flange is zero at the shoulder (rigid connection between flange and drum body), and the deflection of the flange at

the rim is a maximum. In order to simplify the integrations, the following relation was assumed between the thickness of the flange and its radius: $t_r = 1/\sqrt[3]{(\alpha + \beta r)}$, where t_r = thickness of flange at radius r, and α and β are constants. This gives a shape of flange closely approximating that in Fig. 4. A special case arises when $\beta = 0$; the flange is then of constant thickness.

21 The radial stress (tension and compression) at shoulder of flange, taking $K = r_i/r_o$, is:

$$p_r = \frac{0.884N}{(1-K)^2(4.7+2.7K^2)t_i^2}\Bigg[\left(1 - K\frac{t_i^3}{t_o^3}\right)(10.8 \log K + 4.25 - 4.7K + 1.35K^2 - 0.9K^3) - \left(\frac{t_i^3}{t_o^3} - 1\right)(2.58 - 4.7K + 2.35K^2 - 0.9\,K^3 + 0.68K^4)\Bigg] \text{ for cast iron} \quad [5a]$$

$$= \frac{0.866N}{(1-K)^2(4.3+2.3K^2)t_i^2}\Bigg[\left(1 - K\frac{t_i^3}{t_o^3}\right)(9.89 \log K + 3.92 - 4.3K + 1.15K^2 - 0.77K^3) - \left(\frac{t_i^3}{t_o^3} - 1\right)(2.34 - 4.3K + 2.15K^2 - 0.77K^3 + 0.58K^4)\Bigg] \text{ for steel} \quad [5b]$$

and the tangential stress (tension and compression) at rim of flange:

$$p_t = \frac{0.697N}{(1-K)^2(4.7+2.7K^2)t_o^2}\Bigg[\left(\frac{t_o^3}{t_i^3} - K\right)(-10.8K^3 \log K - 0.78 - 2.35K^2 + 3.13K^3) - \left(1 - \frac{t_o^3}{t_i^3}\right)(0.39 + 2.35K^2 - 3.13K^3 + 1.18K^4)\Bigg] \text{ for cast iron} \quad [6a]$$

$$= \frac{0.666N}{(1-K)^2(4.3+2.3K^2)t_o^2}\Bigg[\left(\frac{t_o^3}{t_i^3} - K\right)(-9.89K^2 \log K - 0.72 - 2.15K^2 + 2.87K^3) - \left(1 - \frac{t_o^3}{t_i^3}\right)0.36 + 2.15K^2 - 2.87K^3 + 1.08K^4)\Bigg] \text{ for steel} \quad [6b]$$

The shearing stress at shoulder of flange:

$$s = \frac{0.239N}{r_i t_i} \qquad [7]$$

22 The equations above give the stresses caused by the lateral rope pressure. By a similar analysis, it is found that the force C due to the action of the clutch causes stresses in the clutch flange, as follows:

$$P_r' = \frac{0.884C}{(1-K)(4.7+2.7K^2)t_i^2}\left[\left(1-K\frac{t_i^3}{t_o^3}\right)(10.8\log K - 1.35 + 1.35K^2) - \left(\frac{t_i^3}{t_o^3}-1\right)(5.6-4.7K-0.9K^3)\right]$$

for cast iron [8a]

$$= \frac{0.866C}{(1-K)(4.3+2.3K^2)t_i^2}\left[\left(1-K\frac{t_i^3}{t_o^3}\right)(9.89\log K - 1.15 + 1.15K^2) - \left(\frac{t_i^3}{t_o^3}-1\right)(5.1-4.3K-0.77K^3)\right]$$

for steel [8b]

$$P_t' = \frac{0.697C}{(1-K)(4.7+2.7K^2)t_o^2}\left[\left(\frac{t_o^3}{t_i^3}-K\right)(-10.8K^2\log K - 2.35 + 2.35K^2) - \left(1-\frac{t_o^3}{t_i^3}\right)(1.57-4.7K^2+3.13K^3)\right]$$

for cast iron [9a]

$$= \frac{0.666C}{(1-K)(4.3+2.3K^2)t_o^2}\left[\left(\frac{t_o^3}{t_i^3}-K\right)(-9.89K^2\log K - 2.15 + 2.15K^2) - \left(1-\frac{t_o^3}{t_i^3}\right)(1.43-4.3K^2+2.87K^3)\right]$$

for steel [9b]

$$s' = \frac{0.239C}{r_i t_i} \qquad [10]$$

23 These latter stresses act counter to the stresses p_r, p_t, and s. In other words, when the first layer of rope is being wound on the drum, the clutch flange is stressed in tension in the region A, Fig. 4;

but as the number of layers increases, this stress becomes less and less until a point is reached at which p_r and p_r' exactly neutralize each other, after which p_r becomes the greater and the flange is under tension in the region B. As a rule the net stress due to loads N and C combined will not exceed p_r, p_t, and s, unless the depth of winding is kept down to two or three layers.

24 To determine the effect of the turning force S, consider (1) a small cube in the region B, Fig. 4, and (2) a small cube halfway between A and B, on the neutral surface. The first cube is subjected to an apparent radial stress $p_1 = (p_r - p_r')q^2/(q^2 - 1)$, an apparent tangential stress $p_2 = p_1/q$, and a shear $S/(2\pi r_i t_i)$. Combining these in the usual manner to find the principal stresses, and combining the principal stresses to find the true stresses, we have:

$$p' = \frac{1}{2}\left[(p_r - p_r') + \sqrt{(p_r - p_r')^2 + \frac{S^2(q+1)^2}{\pi^2 r_i^2 t_i^2 q^2}}\right] \quad . \; . \; [11]$$

$$s' = \frac{1}{2}\sqrt{(p_r - p_r')^2\frac{q^2}{(q+1)^2} + \frac{S^2}{\pi^2 r_i^2 t_i^2}} \quad . \; . \; . \; . \; . \; . \; [12]$$

q being the factor of lateral contraction. The second cube is subjected to no normal stresses, but to two shearing stresses at right angles to each other, $S/(2\pi r_i t_i)$ and $0.239N/(r_i t_i)$. Hence the total stress is the algebraic sum of these two.

25 The writer must admit that these formulæ have a formidable appearance, and will be likely to discourage the designer who is looking for simple and easily applied rules. The chief difficulty lies with Equations [5], [6], [8] and [9]; when these have been solved it is a comparatively simple matter to apply Equations [11] and [12] as checks to determine whether the addition of a force S will still keep the stresses within safe limits. On the other hand, the equations for p_r and p_t may themselves be much simplified by omitting terms of negligible value and by assuming that the flange is of constant thickness. When the winding space is shallow as compared with the diameter of the drum, the flanges approximate short cantilever beams of depth t_i, width $2\pi r_i$, and carrying a uniformly distributed load N and a concentrated load $-C$. The tangential stress vanishes, the maximum radial stress becomes

$$p_r = 0.477\left(\frac{1}{K} - 1\right)\frac{N - 2C}{t_i^2} \quad . \; . \; . \; . \; . \; . \; . \; [13]$$

and the maximum shear (due to loads N and C) becomes

$$s = \frac{0.239(N - C)}{r_i t_i} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [14]$$

The additional stress caused by load S may be found as in Equations [11] and [12].

DETERMINATION OF STRESSES IN THE DRUM BODY

26 The obvious load in this case is one of compression, due to the crushing effect of the rope wound upon the drum under tension. There are, however, additional applied forces which should be considered in a careful analysis of the problem. Besides acting as a cylinder under external pressure, the drum body is also a beam sup-

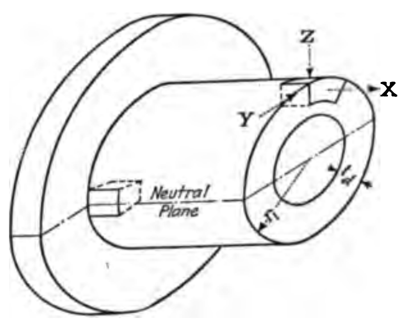


FIG. 5 DIAGRAM FOR STUDY OF STRESSES IN HOISTING-DRUM BODY

ported at the ends and carrying a moving load, a shaft transmitting a torsional moment, and a tensional member serving to tie the flanges together. These forces combine to produce definite stresses which vary in different parts of the drum, so that it is necessary to select certain points at which the stresses are to be determined, and to assume certain positions for the moving load P. The most suitable points are shown in Fig. 5 by two unit cubes, set "square" with the drum, it being assumed that the rope pull is acting vertically upward. One of these cubes is in the center of the drum body, on the side which is under tension, considering the drum as a beam; the other is alongside one of the points of support of the drum body, and in the neutral plane.

27 Consider the first cube. On each face there is a normal force, and in the x-y plane there is a shear due to the torque transmitted. X is a tension caused by the rope pressure on the flanges,

minus the clutch pressure, to which is added the tension caused by the bending effect of the load which is assumed to hang from the middle of the drum. Dividing by the section area for the direct tension, and using the common flexure formula for the bending, we have, approximately,

$$X = \frac{N - C}{2\pi r_i t_d} + \frac{Pl}{4\pi r_i^2 t_d}$$

where l is the length of the drum between bearing centers. Y is a pressure due to the rope tension; on the basis of a uniform distribution of this load over the entire drum body,

$$Y = -\frac{mP - \Sigma(P')}{vt_d}$$

v being the pitch or spacing of coils. Also

$$Z = \frac{Yt_d}{r_i} = -\frac{mP - \Sigma(P')}{vr_i}.$$

The shear due to the twisting moment is a maximum when the rated load P is acting on the outermost layer that can be wound on the drum. Denoting the radius to the center of this layer by r_m, and substituting in the usual torsion formula, the shear is equal to, approximately,

$$W = \frac{Pr_m}{2\pi r_i^2 t_d}.$$

The X and Y stresses can be combined with the shear W, giving the principal normal stresses and the maximum shearing stress

$$\tfrac{1}{2}\sqrt{(X - Y)^2 + 4W^2} \quad \ldots \ldots \ldots \quad [15]$$

From these are derived the true normal stresses

$$X' = \frac{q-1}{q} \times \frac{X+Y}{2} + \frac{q+1}{q} \times \frac{\sqrt{(X-Y)^2 + 4W^2}}{2} - \frac{Z}{q} \quad . \quad [16]$$

$$Y' = \frac{q-1}{q} \times \frac{X+Y}{2} - \frac{q+1}{q} \times \frac{\sqrt{(X-Y)^2 + 4W^2}}{2} - \frac{Z}{q} \quad . \; . \quad [17]$$

$$Z' = Z - \frac{X}{q} - \frac{Y}{q} \quad \ldots \ldots \ldots \ldots \ldots \quad [18]$$

28 To solve for t_d it is only necessary to make the proper substitutions in the equations for X, Y, and Z, then to substitute these in Equations [15], [16], [17] and [18], solve, and take the result which is numerically largest as the correct value for the thickness. When the allowable tensile and compressive stresses are different, as in the case of cast iron, it may be assumed that X' represents a tensile, and Y' a compressive, stress. The sign of Z' depends largely upon the relative values of X and Y, and in general Z' will be small if based

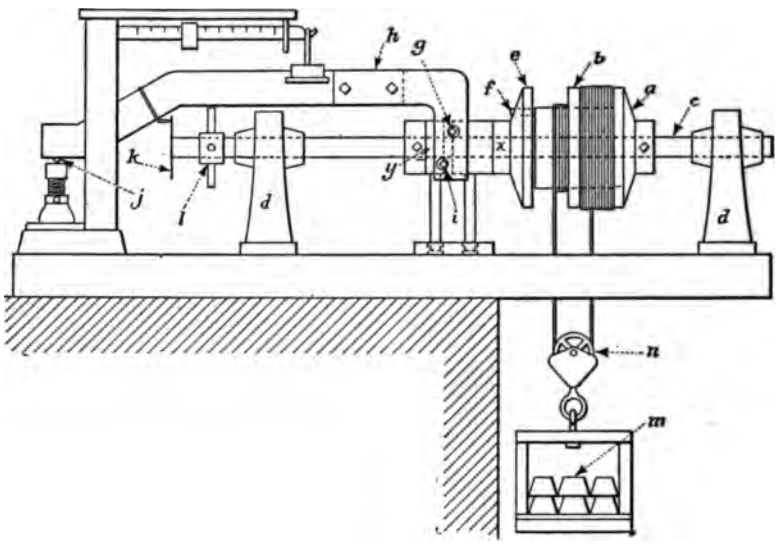


FIG. 6 DIAGRAM OF APPARATUS FOR DETERMINING AXIAL THRUST ON FLANGE OF HOISTING DRUM

upon a value of t_d obtained by solving Equations [16] or [17].

29 Consider now the second cube, located in the neutral plane of the drum body. The stresses Y and Z are the same as before, but X reduces to

$$X = \frac{N - C}{2\pi r_i t_d}$$

since the bending moment at this point is zero; and W becomes

$$W = \frac{P r_m}{2\pi r_i^2 t_d} + \frac{P}{2\pi r_i t_d}$$

where the first term represents the shear due to torsion, as before, and the second term the maximum shear due to bending. By making the same substitutions as in the previous case, the correct value of t_s may be found for the drum body at either end.

EXPERIMENTAL INVESTIGATION OF AXIAL THRUST

30 The apparatus used for this work is shown in Fig. 6. Here *a* is the drum, which is fastened rigidly to the shaft *c* supported in bearings *d*, *d*. An adjustable flange *b* divides the drum into two parts. The left end of the drum is without a flange, but close to it is placed the separate flange *e*, which has an easy sliding fit on the shaft but is made to turn with it by means of the pin *f*. Any axial load against

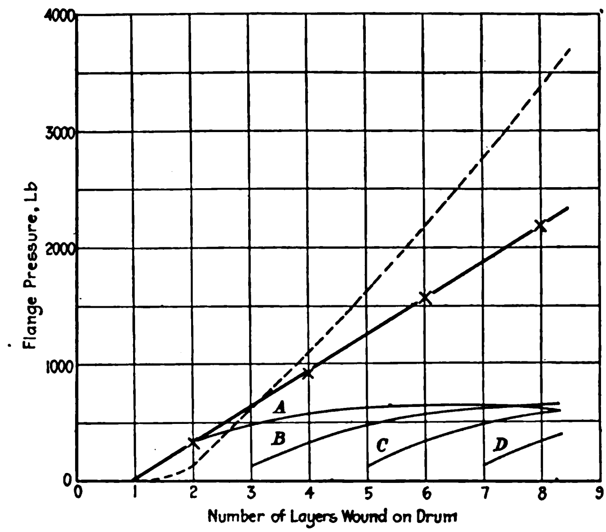


FIG. 7 CURVES SHOWING FLANGE PRESSURE

(Rope pull, 256 lb.; rope, ¼ in., 6 by 19; pitch of coils, $\frac{5}{16}$ in.)

this loose flange is transmitted through the knife edge *g* to the lever *h*, fulcrumed at *i* and bearing upon platform scales at *j*; thus by balancing the scales a quantitative indication of this side thrust can be obtained for any number of turns of rope upon the drum. The number of turns is shown by the dial *k*, and can be varied by turning the shaft by means of the capstan *l*. The platform and weights *m* produce whatever rope tension may be desired; and the necessity of a deep shaft in which these weights may rise and fall is obviated by passing the free end of the rope through the sheave *n* and back on the right-hand end of the drum, so that as it is wound on the testing end it is unwound from the storage end, and vice versa.

31 The following procedure was used in making the tests: The desired size of rope was attached to the drum, a sufficient length being used to fill the left end of the drum to the outside of the flange. One complete layer was wound on the testing end of the drum, the required spacing having been obtained by fiber liners placed between adjacent coils, at close intervals around the circumference of the drum body. It would of course have been possible to space the coils by grooving the drum, but this would have limited the apparatus to one size of rope. The desired weight was then placed on the hanging platform, and all was in readiness for a test run. In making a run

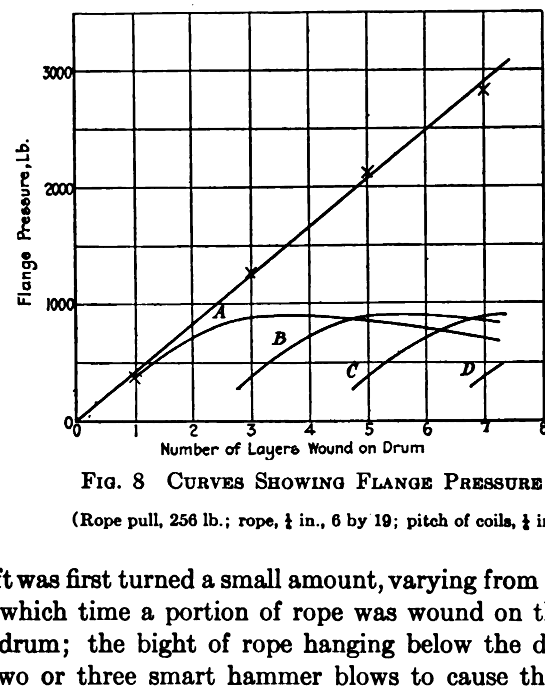


FIG. 8 CURVES SHOWING FLANGE PRESSURE

(Rope pull, 256 lb.; rope, ¼ in., 6 by 19; pitch of coils, ¼ in.)

the shaft was first turned a small amount, varying from 10 to 360 deg., during which time a portion of rope was wound on the testing end of the drum; the bight of rope hanging below the drum was then given two or three smart hammer blows to cause the new coils to settle well in place, and the scale beam was balanced and reading recorded. This was repeated until the testing end was practically filled with rope. The small turning intervals, 10 deg., 30 deg. and the like, were used only for the coils near the loose flange, as practically all of the increase in axial thrust occurred when these coils were wound on. For the remaining coils, readings taken for each revolution of the drum were considered sufficient.

32 A marked periodic rise and fall in the scale readings occurred during each revolution of the drum. This was undoubtedly

due to a slight camber in the shaft, with possibly some additional effect caused by untruth in the bearing surfaces x and y, Fig. 6, since the phase of this periodic change varied somewhat as the axial thrust increased. On account of its periodic nature, the error could be corrected by simply taking averages of the readings obtained during each revolution of the shaft.

33 It was also noted that, no matter how carefully the scales were balanced, the load upon them dropped off a few pounds during the course of three or four hours. Apparently a slight permanent set of some kind took place; possibly the rope contracted slightly in diameter, or the coils settled more closely together, and in so doing lost a part of their initial tension, thereby lightening the axial thrust. However, it was not thought necessary to investigate this action, as the recorded loads were always the maxima, and would therefore insure a better factor of safety than any values obtained after the apparatus had been allowed to settle.

34 Figs. 7 and 8 show the results of two of these tests in graphic form. In each figure, curve A indicates the flange pressure opposite the first layer, curve B the flange pressure against the third layer, etc. The heavy curve shows the total flange pressure, and the dotted curve gives the total pressure as computed by Formula [**3**]. A surprising difference in flange pressures is evidenced by a comparison of Fig. 7 with Fig. 8, where the rope pulls are the same, but the coils are wound close instead of being separated. The writer attributes this to a difference in the winding characteristics of the two cases: when the coils are separated to simulate the effect of a grooved drum, they are wound on in each layer in approximately helical form; but when they are closely packed, each coil is a true circle except for an arc of about 30 deg., in which space the advance to the position of the next coil takes place. This means that the arc of crossover, in which the coils are self-sustaining, is longer for the spaced than for the closely packed arrangement of coils; hence the flange pressure is less. Furthermore, the closely packed coils are in contact with each other at six points, instead of four points, per coil; consequently there is a less pronounced rope flattening in the interior layers, with a correspondingly higher flange pressure as a result.

35 This latter effect could be taken account of in Formula [**3**], by selecting the values of P'/P in Table 1 which correspond to higher values of the lateral-deformation factor m. As the writer did not have the time to obtain experimental values of m, he considered it best to omit the theoretical curves from Fig. 8.

36 At least two conclusions may be drawn from the tests: First, the actual flange pressure increases, roughly, in direct ratio with the number of layers, whereas the theoretical pressure (neglecting friction and rope flattening) is represented by a curve with rising slope; so that at eight layers the actual is only a tenth or less part of the theoretical pressure. The plotted points in Figs. 7 and 8 might be taken to indicate a drooping curve for the actual pressure, but Fig. 9, where four additional layers are shown, points rather to a straight-line relation. Second, the results obtained with Formula [3], and shown in Fig. 7 by a dotted curve, show a fair degree of correspondence with the test curves, and err for the most part on the side of safety. In plotting the dotted curve, E' was taken at 9260 lb. per sq. in., m at 0.3, and μ at 0.2, all of which are high values. Lower values would be more truthful and would make a still closer agreement of the dotted curves with the test curves.

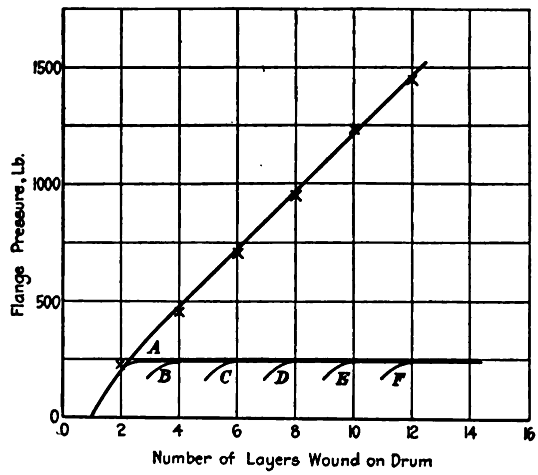


FIG. 9 CURVES SHOWING FLANGE PRESSURE

(Rope pull, 56 lb.; rope, $\frac{3}{8}$ in., 6 by 7; pitch of coils, $\frac{5}{32}$ in.)

37 The question naturally arises: If the tests indicate a straight-line relation between the flange pressure and the number of layers of rope, what value is there in a complicated formula such as Equation [3], which brings in several indefinite quantities like μ, p, E' and m? The answer is that the tests show merely a few results from a very wide range of possible conditions; whereas the formula, by taking into account a large number of variables, should be appli-

cable to all conditions, whether the number of layers, rope pull, coefficient of friction, etc, be large or small. The field is still open for further experimentation, using larger rope sizes and greater rope pulls, and the quantities E' and m should be accurately determined for different sizes and lays of rope. The writer hopes that this work can be undertaken in the near future; and that the results will stand as an additional check upon the theoretically derived relation between rope pull and flange pressure embodied in Formula [3].

DISCUSSION

W. C. Marshall asked why it was necessary to have flanges on the drums in question. If the axial thrust against the flange were reduced to zero, the flange would become unnecessary. Such a condition obtained with textile machinery in the universal winding cone. He asked whether or not this method had been tried, using a mechanical guide for the rope as is done with the universal winding cone.

The Author did not think it practical to wind rope on a drum without flanges. Manufacturers of rope would advocate winding it in a single layer, but manufacturers of drums would point out the impracticality of making drums of sufficient size to accomplish this end.

R. S. Bolger, asked if it would make any difference whether wire or Manilla rope were used, and the author replied that his paper dealt with ordinary wire rope of six strands with hemp core. The constants for Manilla rope would be different. Manilla rope might flatten so that there would be no tension except in the outer layer.

M. A. Stone, Jr., said the author had shown that as more coils were put on the drum the tension in the lower coils decreased. It had been his experience in the making of wire-wound guns thar the tension was constant. Did a wire rope act differently, he asked, than a steel ribbon?

The Author replied that if several layers of steel ribbon were wound under uniform tension the tension in the lower layers would be reduced considerably.

H. L. Doolittle asked what was the effect of the pressure of several layers of rope on the drum itself. The author answered that many practical designers took into consideration only the pressure from two layers of rope.

The Author had hoped that the presentation of the paper would bring out additional facts regarding flange pressure. However, the trend of the discussion indicates that such information has not, as yet, been gathered and arranged in systematic form. This gives additional weight to the statement in the closing paragraph of the paper to the effect that a further investigation of the subject, and on a larger scale, is needed.

No. 1761

EXPERIENCES WITH LARGE CENTER-CRANK SHAFTS

BY LOUIS ILLMER, PHILADELPHIA, PA.
Member of the Society

This paper is descriptive of some disastrous experiences with a number of large gas-engine shafts of the center-crank type. Investigations as to the cause of the repeated shaft failures revealed inherent structural weakness due to defective design. The reinforcements undertaken to remedy this condition were based upon careful stress analysis of actual sag determinations conducted upon the wheel shaft.

The center-crank shafts in question were mounted upon three-point bearing supports and carried a heavy flywheel between the intermediate and outboard bearings. The stress diagrams show that this mode of support is likely to set up a pernicious interaction of bearing load, culminating in excessive wear in the intermediate main bearing.

When the wheel shaft lacks adequate stiffness, the appreciable lifting action at the free end of the web portion of a center-crank shaft may exert a considerable thrust against the cap of the aligned outer main bearing, which in turn reacts upon the intermediate journal, causing it to become overloaded. The resulting rapid wear in the intermediate main bearing reduces the upward cap thrust and this drop in alignment gradually relieves the intermediate bearing of overload. Further wear causes a portion of the downward load on the intermediate bearing to be transferred to the outer main bearing.

It was found that when this readjustment is complete, the downward load upon the two main bearings becomes approximately equalized. The considerable sag required to attain this state of equilibrium as to wear, involved running under stress conditions so severe as to lead to ultimate breakdown of the wheel shaft.

Starting with newly aligned bearings, the observed wear in the intermediate bearing was found to exceed ¼ in. in less than thirty days of continuous operation, but after the intermediate bearing had dropped sufficiently to equalize the load upon the two main bearings, the wear became more nearly normal.

The stress diagrams corresponding to the estimated wheel-shaft deflection curves show the critical surface fibers to be subjected to a reversing stress ranging from 16,700 lb. per sq. in. compression to about 27,000 lb. per sq. in. tension stress, which is far in excess of the allowable range fixed by the Appendix.

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The life of these shafts, as expressed in observed running time required for breakdown, appears to be in fair accord with the expected number of revolutions as determined by Stromeyer's Law of Fatigue.

It is further shown that heavy counterweights are capable of exerting a detrimental influence upon the stress relations of a heavily loaded wheel shaft of the center-crank type. Under these conditions the best counterweight proportions are attained when its centrifugal force is made to balance that of the crank throw and the large end of the connecting rod.

The character of the reinforcements that were necessary to strengthen the defective wheel shaft is outlined, and the resulting improvements in stress and sag relations are recorded in the diagrams. In making replacements, both carbon and nickel-steel forgings were tried out, but the nickel-steel shafts did not come up to expectations.

Material specifications and acceptance tests are also tabulated, and the results of test bars taken from some critical sections of the defective nickel-steel shaft are compared with similar tests made upon a carbon-steel shaft.

In order to keep the wheel-shaft stress within desired limits under load conditions found in these engines, it finally became necessary to enlarge the intermediate main journal to $\frac{5}{4}$ of its original diameter, thus making its dimensions fully as large as required for side-crank construction.

The conclusion arrived at is that for large center-crank shafts carrying heavy flywheels with a long span between supports, the resulting stress relations are best determined upon the modified basis advocated in this paper. The maximum stresses so found are considerably higher than would be expected from simple crank-shaft formulae, such as are commonly used to determine center-crank-shaft stresses.

Before fixing upon the final dimensions of a large shaft of this type, it is therefore expedient to make a careful analysis of the underlying stress conditions so as to insure a proper margin of safety in both the web and wheel parts after allowing for the lengthening of the span between the wheel supports due to excess wear in the intermediate main bearing.

SEVERAL years ago a series of disastrous center-crank-shaft failures occurred in a 1200-kw. gas power plant, which the author investigated and found to be due primarily to defective design, the wheel shafts in question being entirely too light for the load they were obliged to carry.

2 The attempt to eliminate an inherent defect of this kind gave rise to a number of difficult engineering problems. It is the purpose of this paper to discuss these, to set forth the reasons for these shaft failures, and to record the behavior of carbon versus nickel-steel shafts, working under excessive stress.

3 The gas power plant comprised three 25-in. by 43-in. 500-b.hp. double-acting producer-gas engine units, each driving a 400-kw. 25-cycle three-phase alternator at 100 r.p.m. The engines were of the direct-connected horizontal type, provided with a massive flywheel. The shafts were of the single-throw center-crank type,

as indicated in Fig. 1, and were Krupp forgings of open-hearth carbon steel, having a tensile strength of about 65,000 lb. per sq. in. Three Krupp shafts, with characteristics as shown in Table 1, were delivered with the engines.

4 After being in 24-hour-a-day service for some four or five years, one of the original shafts unexpectedly gave way near the flywheel hub, allowing the armature to drop and thus completely wreck its generator and heavy wheel, and in other ways doing serious damage.

5 In order that the cause of this shaft failure might be intelligently arrived at, a series of sag determinations was conducted on

TABLE 1 TESTS ON ORIGINAL CARBON-STEEL SHAFTS

Average Chemical Analysis		
Carbon	0.26 per cent	
Manganese	0.59	
Silicon	0.18	
Phosphorus	0.042	
Sulphur	0.038	
Physical Characteristics	**Lengthwise**	**Crosswise**
Tensile strength, lb. per sq. in.	65,600	64,300
Elongation	26.7 per cent	21.5 per cent
Contraction	58 per cent	53 per cent

the wheel shaft and a thorough technical investigation made of the underlying stress conditions.

6 As will presently be shown, the three-point shaft support for the heavy flywheel was found to set up a pernicious interaction of load which led to excessive wear in the intermediate main bearing. This resulted in the lengthening of the effective shaft span, which in turn produced sag stresses of such magnitude as to cause ultimate rupture of the shaft.

7 It was decided to rebuild the first damaged unit and to purchase a new carbon-steel shaft for it. This forging was reinforced in the manner indicated in Fig. 5. In order to ease the load on this shaft, the wrecked flywheel was replaced by a lighter one of I-beam rim construction, running at considerably higher peripheral speed.

8 Upon further investigation it was found that the shafts of the two remaining engines were running out of round by about $\frac{1}{32}$ in. and that they also were seriously damaged by fatigue and partial

rupture. It thus became necessary to provide new reinforced shafts for these engines as well, and since their wheels were in good condition it was deemed advisable to utilize them by boring out the hubs 2 in. larger in diameter to fit the new shafts.

9 These two other shafts were ordered of nickel steel, as it was thought its higher tensile strength would compensate for the heavier flywheels carried by the second and third engines. By so doing, it became possible to try out carbon steel and nickel steel under almost identical conditions and ascertain their relative merits for this particular duty.

10 The results with nickel steel fell short of expectations since the crank web in one of the new nickel-steel shafts gave way and caused a second disaster in less than one year's operation. In this case the wheel shaft remained intact and the damage was confined chiefly to the engine parts proper, although the consequences were serious. The outermost main bearing was literally torn out of the bedplate, bending the connecting rod and letting the heavy piston come forward sufficiently to strike the head and pull off the entire end of the cylinder casting. The fact that the bedplate was built with a pillow-block section fortunately localized the frame damage.

11 In making the second set of engine repairs, the cylinder had to be entirely replaced and in redesigning the pillow block it was decided to effect a radical cure by going back to carbon steel and increasing the diameter of the intermediate main bearing in this one engine by 50 per cent, as shown in Fig. 7. With this change the journal of the fourth new center-crank shaft was made equal in size to that required for side-crank construction. These massive proportions were necessary to insure a proper margin of safety under the severe stress conditions found in these engines.

12 The inadequacy of a center-crank shaft for carrying a heavy wheel is made evident in the accompanying deflection diagrams. An excessive wheel-shaft deflection puts a large upward thrust against the outermost main-bearing cap. The reaction of this thrust upon the intermediate main bearing may become so large as to squeeze the babbitt lining out of the bearing shell, in which event the resulting drop in this vital support allows the shaft to run with an excessive sag between the outermost main bearing and the outboard bearing. Thus the shaft stresses may be increased far beyond those usually anticipated by the engine builder, and if allowed to continue can readily cause the metal of this shaft to undergo such rapid fatigue as to lead to shaft rupture.

13 The matters of fixed stress limits and requisite factor of safety underlying good shaft design are rather fully discussed in the Appendix.

CRANK-SHAFT-STRESS DIAGRAMS

14 The shaft outline and load conditions under which the original Krupp shafts operated are shown in Fig. 1, in which the line *A–A* indicates the location of the wheel-shaft failure, its relative position being almost identical in all three of these shafts. The journal portion of the shaft was carried straight up to the wheel hub but as indicated by the dotted outline in Fig. 1, it was entirely feasible to enlarge the shaft immediately after clearing the camshaft gear. Had this been done originally, it would have obviated the worst of the peak stresses shown in the diagram.

15 The original Krupp shafts were excessively loaded by a massive flywheel weighing about 73,000 lb., and by an armature of about 13,000 lb., to which should be added an assumed magnetic pull of about 10,000 lb., on the basis of $\frac{1}{32}$ in. armature displacement. The location of these weights is shown in Fig. 1, and their relative magnitude is indicated by varying line lengths, extending downward.

16 In making a preliminary stress analysis, in the conventional manner, it may be assumed that the wheel shaft is pin-supported between the intermediate main bearing I and the outboard bearing III. The resultant wheel-shaft load $P_r = 96{,}000$ lb. is readily found by taking moments about either of the end supports. In a similar manner, the reaction in the main bearing I becomes equal to 53,500 lb., while that in the outboard bearing III is 42,500 lb. To each of these forces approximately one-half the weight of the 5000-lb. wheel shaft should be added, as indicated in the diagram.

17 The bending moment may be found by considering the shaft as fixed at any given breaking point and subjecting this section to a moment equal to the nearest bearing reaction, multiplied by its leverage with respect to such section. This external moment is resisted by the internal moment produced within the shaft fiber.

18 The resultant wheel-shaft load $P_r = 96{,}000$ lb., acting downward, causes the shaft to deflect, thus putting the bottom fibers in tension and the top fibers in compression. The rotation of the shaft sets up a reversal of stress with each revolution.

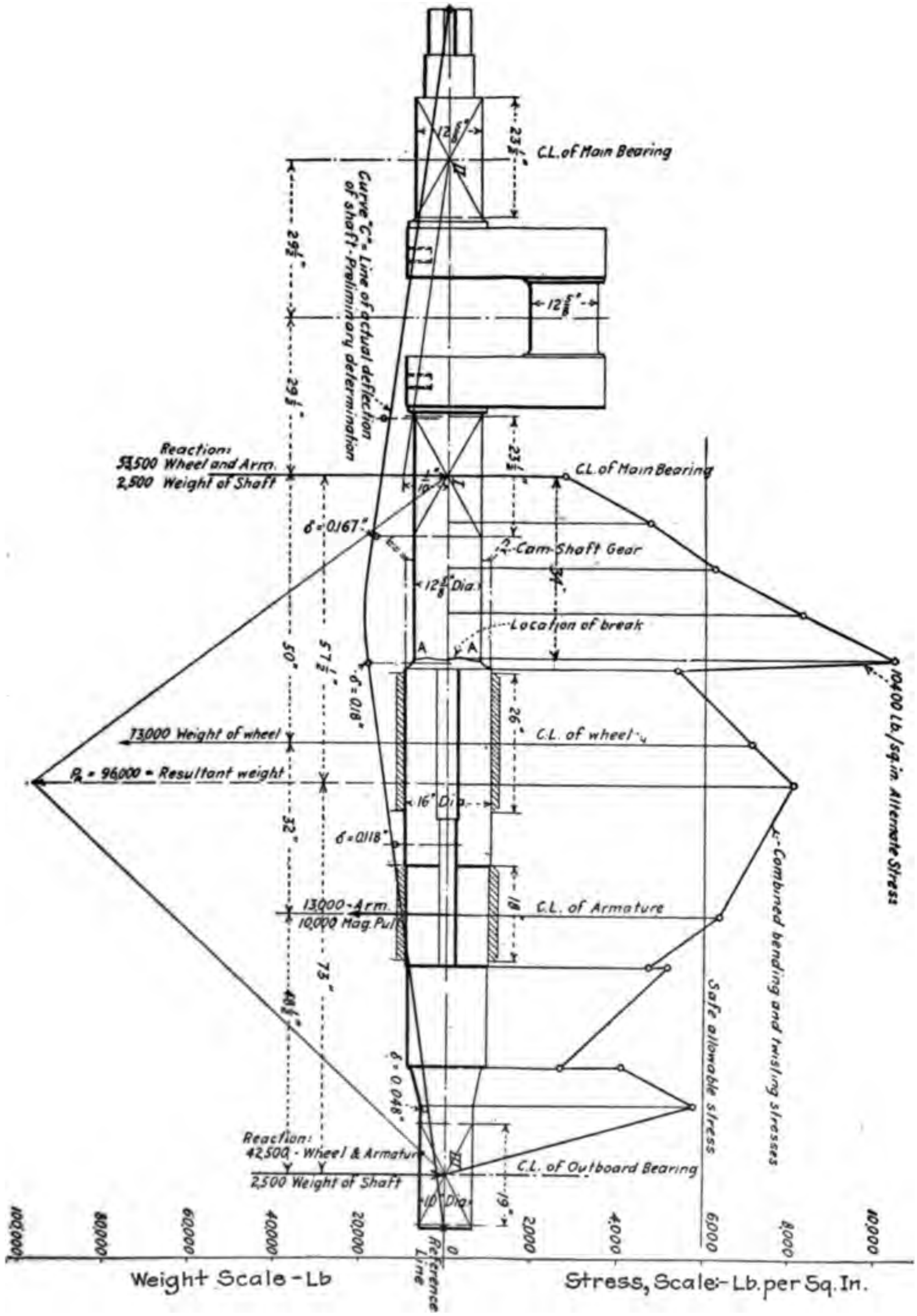

FIG. 1 PRELIMINARY STRESS DIAGRAM FOR PIN-SUPPORTED WHEEL SHAFT

19 In addition to these alternating stresses, the shaft is subject to a negligible shear stress due principally to the weight of the wheel. In a horizontal engine, the effect of the connecting-rod thrust in producing an increase in the wheel-shaft stress may be sufficiently allowed for by taking into account only the twisting stress resulting from the transmission of power to the generator. For the engines under consideration, the maximum twisting moment as determined from their crank-effort diagram, is approximately 50,000 lb., taken at full crank length, which corresponds to a piston pressure of about 100 lb. per sq. in.

20 The effect of this twisting action may best be combined with the bending moment as indicated in the stress curve in Fig. 1. It will be seen that while the original wheel-shaft strength lacks uniformity, the maximum stress estimated on this preliminary basis reaches a peak value of only about 10,000 lb. per sq. in. near the point of rupture.

21 Assuming the wheel to be perfectly balanced and the shaft to run true, the basic or net factor of safety for this shaft, as determined by Equation [D] of the Appendix, appears to be equal to about 2.2, as against a minimum stipulated factor of three. It is evident that this minor difference is insufficient to account for the wheel-shaft failure, but the diagram does show that this conventional mode of checking center-crank wheel-shaft stresses cannot be relied upon to bring out the cause of rupture.

EFFECT OF THREE-POINT SHAFT SUPPORT

22 In order to arrive at the cause of the present shaft failure, it is necessary to take into account the interacting effect of the three-point shaft support. Assuming all the bearings to be in perfect alignment, it will then be found that the excessive wheel-shaft deflection produced by the massive flywheel, puts a 20,000-lb. upward thrust against the cap of the outermost main bearing II, which in turn reacts upon and so overloads the intermediate bearing as to squeeze out its babbitt. As indicated in Fig. 3, the drop in this vital support causes sag stresses to be set up that more than double the peak stress previously determined by the preliminary stress check represented in Fig. 1.

23 The estimated stress relations shown in Fig. 1 assume the original wheel shaft to be pin-supported between bearings I and III only, a condition that is approximated when the cap is removed

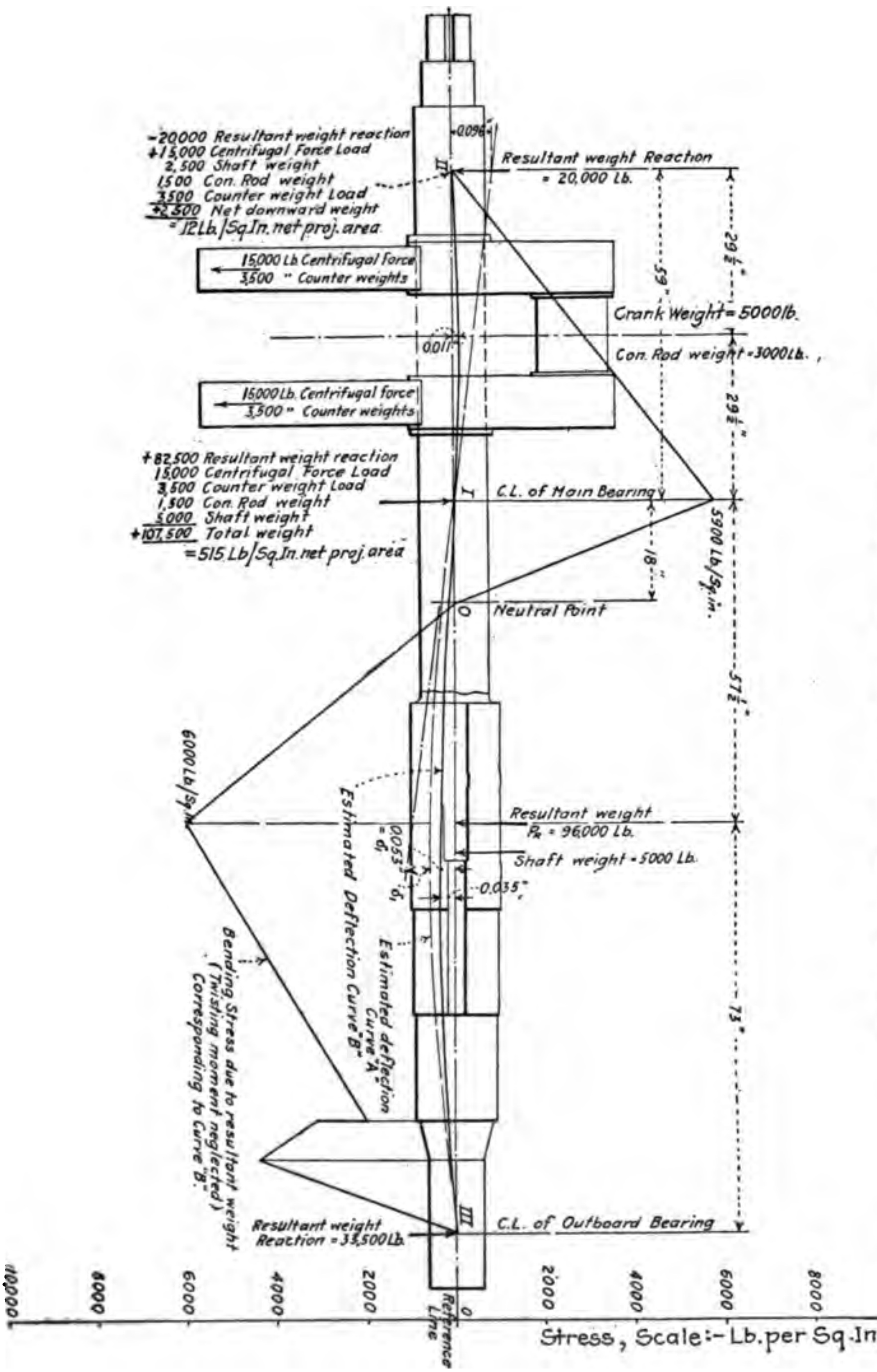

FIG. 2 STRESS DIAGRAM FOR ALIGNED THREE-POINT SHAFT SUPPORT

from the outer main bearing II and when the inertia effects of the crank parts are not taken into consideration.

24 Fig. 2 shows this same shaft placed in perfectly aligned bearings with the cap of bearing II pulled down into place and with the counterweights acting downward. In this figure the dotted curve A represents the estimated deflection that may be expected when the cap II is removed in which case the wheel-shaft sag would be about $\delta_1 = 0.053$ in. The corresponding slope is approximately 0.01 in. per ft. of length, or about twice the normal sag allowance.

25 As indicated by the projected deflection line, this sag tends to raise the free end of the shaft lying in bearing II by about 0.096 in. Approximately 20,000 lb. would be required to force the lifted portion of the shaft into place and this pressure acts against the cap of bearing II, as noted in the diagram. As a consequence, the wheel-shaft deflection is reduced to about 0.035 in., and the shaft is obliged to assume a sinuous deflection curve corresponding to the full-line curve B. The accompanying stress curve shows a peak stress of only 6000 lb. per sq. in. which, could it be maintained, would provide ample factor of safety against shaft rupture.

26 A detailed presentation of the method used in arriving at the sinuous deflection curve B for a three-point shaft support is beyond the scope of the present paper. Fig. 8 indicates the general method pursued in determining the stress and deflection curves. These rather complex relations are still further modified by the centrifugal forces stipulated in Fig. 2. The ultimate wheel-shaft stress has been taken by adding the bearing reaction of the vertical counterweight forces to those resulting from the wheel-shaft load and then finding the bending moment at any point of the wheel shaft on the basis of the gross load supported in the adjacent bearing. The bearing reactions are given in round figures and are only intended to trace the principles underlying the interacting forces that led to ultimate shaft failure.

27 Fig. 2 shows the extent to which the cap thrust of 20,000 lb. reacts to effect an increase in the downward load of the intermediate bearing. With the cap II forced into place under conditions of perfect alignment, the resultant downward pressure in the main bearing I is approximately 400 lb. per sq. in. and if to this the downward centrifugal force is added, a maximum pressure of about 500 lb. per sq. in. of net projected area is reached. Such a heavy bearing pressure fails to distribute uniformly and the consequence of excessive loading is to squeeze the babbitt out of the intermediate

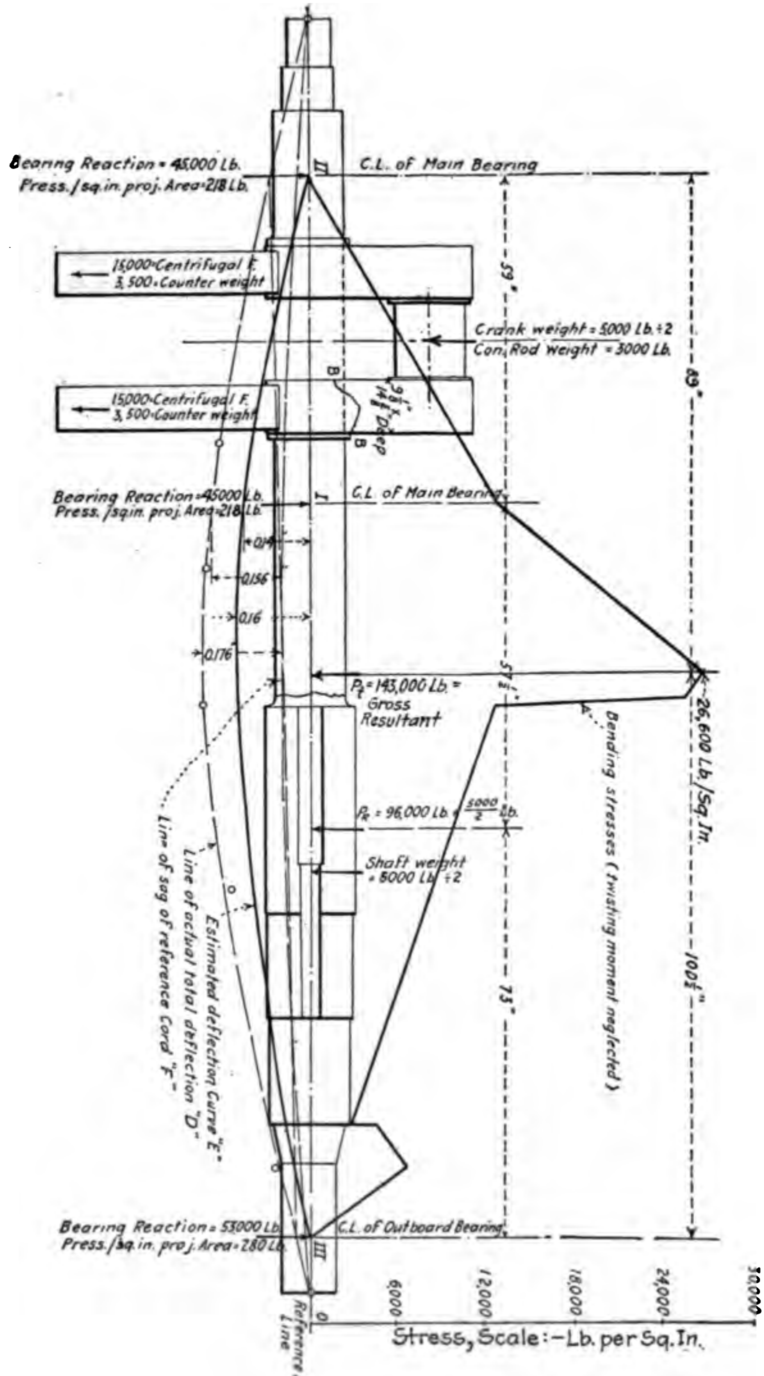

FIG. 3 DIAGRAM OF SHAFT WORN TO STATE OF EQUILIBRIUM

shell.[1] The resulting rapid wear serves to relieve the intermediate bearing of its overload by lowering its alignment with respect to the end supports of the shaft.

28 The dropping of the intermediate bearing support has the further effect of reducing the upward thrust against the bearing cap II, and as the downward forces undergo readjustment by continued wear, the outer main bearing II will gradually assume an increased portion of the downward shaft load. Experience shows that when the load upon each of the main bearings becomes approximately equal, a condition of stable equilibrium will have been established, after which the two main bearings continue to wear down together at a more or less uniform rate.

29 In order to attain such a state of equilibrium, the shafts must suffer undue deflection as indicated in Fig. 3. Experience further shows that when starting with newly aligned bearings this condition is reached in a remarkably short time, being approximately 30 days of continuous operation under the existing load conditions. The corresponding wear in the intermediate bearing was found to be about $\frac{1}{8}$ in. in this brief period, as against a normal wear of say $\frac{1}{64}$ in. to $\frac{1}{32}$ in. per year for continuous running.

30 Fig. 3 also shows the marked effect which the dropping of the intermediate bearing support has in enlarging the effective shaft span and thus largely increasing the bending stresses over those shown in Fig. 2 for perfect bearing alignment

31 These high sag stresses have been checked by comparing the estimated deflection curve E with the actual wheel shaft sag, plotted as curve D in Fig. 3. This curve represents the deflections taken off one of the original Krupp shafts. The deflections were determined by means of a pin gage and these measurements were corrected for the sag of the reference cord, which is plotted as curve F. Thus the actual shaft sag, measured at the center of the intermediate main bearing, was found to be about 0.16 in. lower than the end supports.

32 The calculations show further that after reaching the aforesaid state of equilibrium, the estimated maximum pressure acting in

[1] The above main bearings were of the quarter-box type for which the effective projected area of the bottom shell has been taken at 0.7 of the gross area. From the author's Bearing Design Constants, *Power*, Feb. 22 and 29, 1916, approximately 350 lb. per sq. in. of projected area is shown to be the limit of allowable pressure in babbitted main bearings.

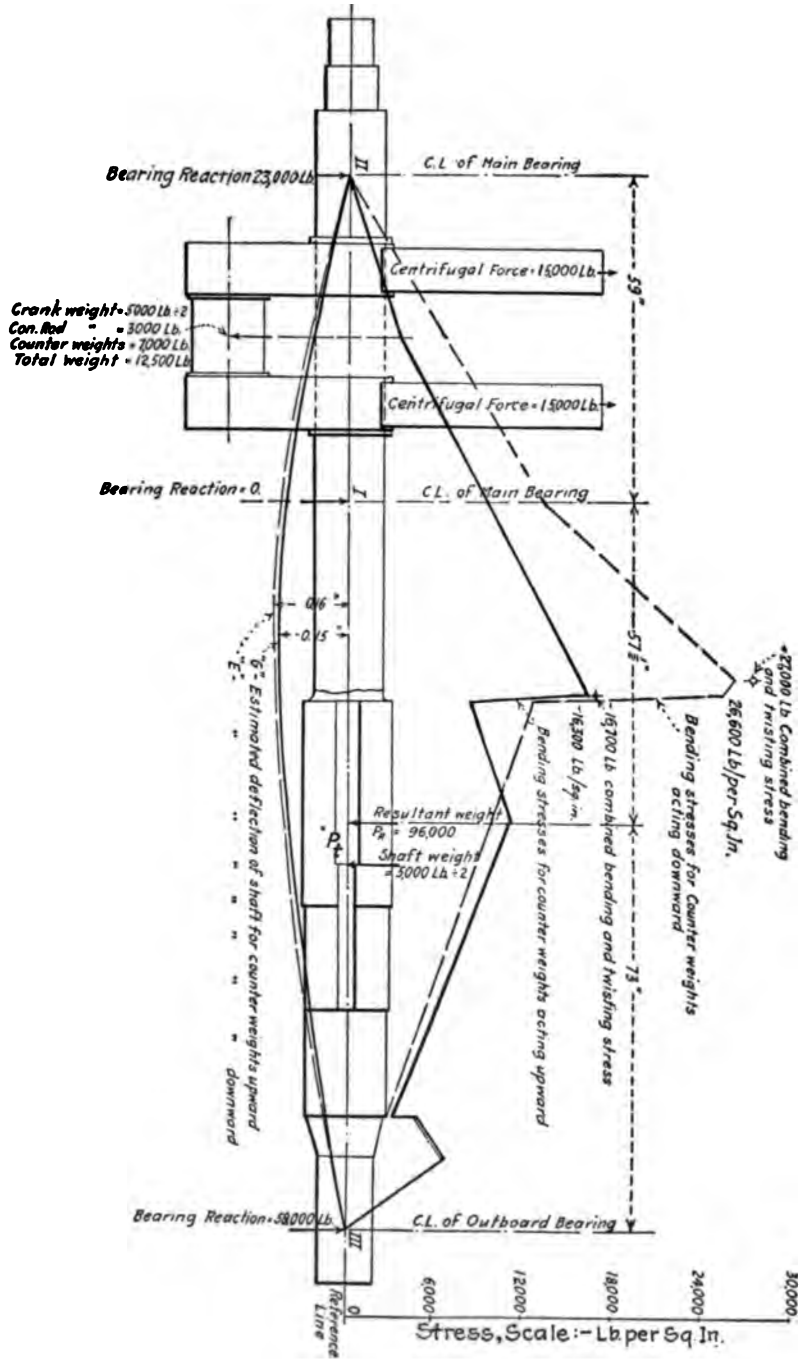

FIG. 4 DIAGRAM SHOWING LIFTING EFFECT EXERTED BY COUNTERWEIGHTS

each of the main bearings is reduced to about 218 lb. per sq. in. of projected area, which can readily be sustained without excessive wear. Under these conditions, the shaft suffers a maximum bending stress of about 26,600 lb. per sq. in., and it is found that this peak practically coincides with the point of rupture, as shown in Fig. 3. This extremely high stress is still further increased when combined with the crank twisting stress.

DETRIMENTAL EFFECT OF HEAVY COUNTERWEIGHTS

33 The massive counterweights with which these engines were provided were designed to balance the inertia effects of the heavy reciprocating parts, averaging about 56,000 lb. at the end of the stroke, i.e., about 115 lb. per sq. in. of piston area. The gross centrifugal force of the two counterweights proper is about 50,000 lb. at full speed, from which about 20,000 lb. is to be deducted for the counterbalancing effect of the crank throw and connecting-rod parts. Thus when the crankpin reaches its upward or downward position, the effective centrifugal force due to the counterweights amounts to about 30,000 lb. which was found to be in excess of the requirements for smooth running.

34 In a center-crank shaft the unbalanced centrifugal forces of the crank throw and connecting-rod parts acting without any counterweights are in themselves sufficiently large to modify the stress relations resulting from a heavily loaded wheel-shaft. When heavy counterweights are used they not only overcome these unbalanced throw and rod forces, but in turn are capable of exerting a detrimental effect upon the critical wheel-shaft stress. When acting downward, the counterweight pull adds to the shaft load and augments the high bending stress due to the long-span wheel-shaft support. When acting upward, the counterweight pull tends to reduce the critical bending stress in the wheel shaft, but the stress variation set up by the rotating counterweights is such as to increase the stress range through which the shaft fiber must work. Without any counterweight whatever the stress range in the shaft under discussion would have been reduced by about 10 per cent. For a heavily loaded wheel shaft of the center-crank type, the best conditions as to stress relations are attained when the counterweights are designed to balance the centrifugal forces of the crank throw and the large end of the connecting rod.

35 In the present instance, the effect of the heavy counter-

weights is made evident by the two stress curves plotted in Fig. 4. The dotted curve represents conditions when the centrifugal forces act downward, while the solid line represents conditions when the counterweight forces act upward. The corresponding shaft deflection curves marked E and G show a maximum estimated sag of 0.16 in. and 0.15 in., respectively.

36 In Fig. 4 it is assumed that the unbalanced counterweight forces when acting upward are sufficiently large to lift the shaft off of its intermediate bearing support. Actually the main bearing loads given in Fig. 3 are probably not fully equalized when the counterweights act downward, and the load on the intermediate main bearing is probably not fully relieved when the counterweights act upward, but the assumptions set forth offer a satisfactory method for arriving at the stress range suffered by a shaft of this kind.

37 Fig. 4 shows that when the centrifugal force acts upward, a maximum bending stress of 16,300 lb. per sq. in. results, while for downward action the bending stress is increased to about 26,600 lb. per sq. in. Combining these high stresses with the maximum twisting stress, without allowing for shock or preignition, an equivalent maximum tension stress of 27,000 lb. per sq. in. is produced in the outer shaft fiber, and when this same fiber turns through 180 deg., it is subjected to a compressive stress of about 16,700 lb. per sq. in.

38 The corresponding ratio of $f_{min.}$ to $f_{max.}$ for this partially reversing stress is about 0.62 as defined in the Appendix, and the total stress range that this surface fiber undergoes during each revolution is about 43,700 lb. per sq. in., a limit far in excess of that which the metal can continuously endure. Checking this result on the basis of the formulæ given in the Appendix, the allowable stress range for unit basic factor of safety for 65,000-lb. T.S. steel is about 39,000 lb. per sq. in., which corresponds to a tension stress limit of about $f_{max.} = 24{,}000$ lb. per sq. in. The net or basic factor of safety in the original wheel shaft is thus found to be about 0.9, i.e., less than unity, for which the corresponding excess tension stress becomes equal to about $f_{exc.} = 27{,}000 - 24{,}000 = 3000$ lb. per sq. in.

39 No record has been kept as to the running time actually required to rupture the original Krupp shafts, but judging from the average period of operation of these engines, it is probable that each of the defective shafts made between 100,000,000 and 150,000,000 revolutions prior to failure.

40 As based upon Stromeyer's fatigue formula [G] as given in the Appendix, the excess stress $f_{exc.}$ required to produce rupture

in the stated number of revolutions would lie between 2700 and 3000 lb. per sq. in. for an alternating stress cycle. Accordingly, it would appear that the original Krupp shafts must have been subjected to a maximum tension stress approaching the elastic limit of the shaft material.

41 The character of the ruptured section of these shafts is shown in Fig. 6. The high alternating stresses set up in the outermost fibers undoubtedly produced gradual fatigue, which led to a rupture of the surface metal. This irreparable damage greatly weakened the shaft, and as the rupture crept inward, a point was finally reached where the shaft could no longer bear its load.

42 The outer fibers in the lower left-hand edge of the fractured section were subjected to the greatest stress and it is not improbable that this portion of the shaft gave way some time prior to the complete shaft failure. This portion of the broken surface showed a fine crystalline fracture while the rest of the shaft showed a rather bright granular fracture, more nearly representative of the original metal.

43 The fact that all three of the original Krupp shafts became defective in practically the same place and that each shaft proved serviceable for a period approximating three years virtually precludes a defect in forging but points rather to improper shaft design as the cause of the mishap.

SHAFT REINFORCEMENTS

44 Regarding the reinforcements undertaken on the first new carbon-steel shaft which replaced the broken shaft in the engine first wrecked, the principal dimensions of the new shaft and its improved stress relations are shown in Fig. 5. The wheel weight was also reduced from 73,000 lb., to 56,000 lb., while the shaft diameter under the wheel was increased from 16 in. to 18 in.

45 The other parts of the wheel shaft were made as large as existing conditions permitted, without rebuilding the frame parts, and the effect of this reinforcement is especially noticeable at the first offset where the $12\frac{5}{8}$-in. journal diameter is made to step up to 16 in. immediately after clearing the camshaft gear as shown dotted in Fig. 1. It is apparent that this enlargement served to effect a material reduction in the peak stresses, and had this simple expedient been resorted to in the original shaft design, it would probably have obviated the considerable loss involved in these shafts failures.

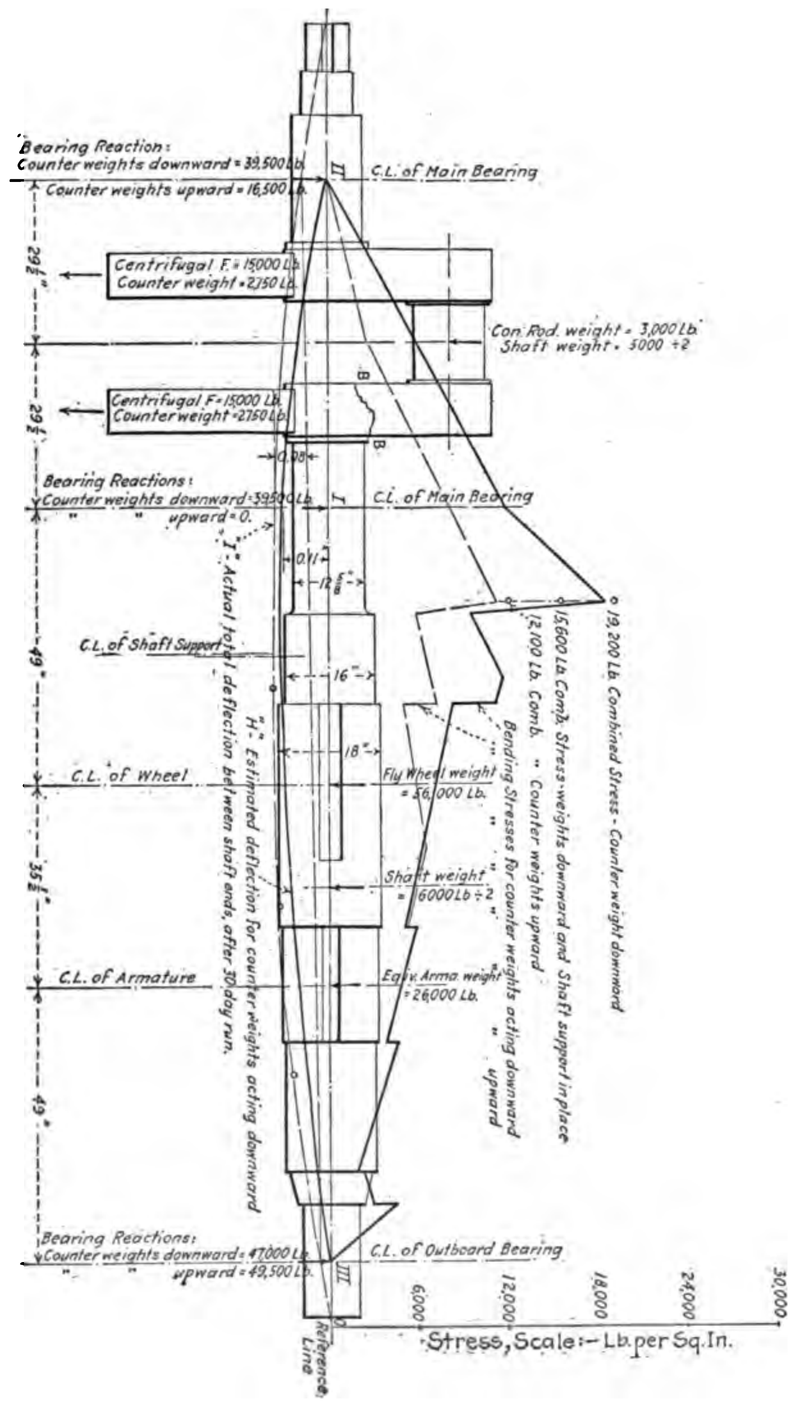

FIG. 5 DIAGRAM SHOWING STRESS REDUCTION IN FIRST REDESIGNED SHAFT

46 As redesigned, the net or basic factor of safety in the first new carbon-steel shaft shown in Fig. 5 was raised to about 1.5 as against 0.9 for the original Krupp shaft. The new carbon-steel shaft was found to give good service and even when working with this close margin of safety has shown no sign of fatigue after some years of continuous operation.

47 Immediately after putting this new shaft into service, the drop in the center bearing was closely observed. It was found that the sag in the newly aligned shells reached approximately ⅛ in. in less than 30 days of continuous operation, thus proving conclusively that the initial pressure for the aligned bearings was still excessive.

48 Because of the stiffer wheel shaft, the maximum pressure in the intermediate main bearing probably did not exceed 450 lb. per sq. in. The bearings showed no signs of heating and proved generally satisfactory, except that they could not withstand the severe load imposed.

SHAFT SPECIFICATIONS AND TESTS

49 All four of the shafts intended to replace the original Krupp shafts were bought under close specifications and the quality of the material was insured by making a series of rather elaborate chemical and physical acceptance tests.

50 The material in the first reinforced carbon-steel shaft, shown in Fig. 5, was made of thoroughly annealed basic open-hearth steel under the specifications in Table 2.

51 Shafts of the same design were also put in the other two engines, but as they carried the heavier original wheels, these were made of nickel-steel forgings under the specifications in Table 3.

52 As previously stated, one of these nickel-steel shafts failed in the crank web, the rupture being adjacent to the intermediate bearing as indicated at B–B in Fig. 5. These webs were only slightly reinforced over the original Krupp shaft dimensions, partly because of limited room, but mainly because no previous web trouble had been experienced. Furthermore, the twisting stress to which the shaft web was subjected, while relatively high, was not so severe that failure was to be expected in less than one year's operation.

53 The following tests were made upon portions of the defective nickel-steel shaft, the bars being taken in the positions indicated:

TEST BAR *A*: From outboard-bearing end of wheel shaft at half radius

TEST BAR *B*: Cut lengthwise, 3 in. off center from inner surface of opposite unbroken web, corresponding in location to web rupture

TEST BAR *C*: From defective web, close to point of rupture. The results obtained are given in Table 4.

TABLE 2 CARBON-STEEL SHAFT SPECIFICATIONS

Chemical Analysis	
Carbon	0.35 to 0.40 per cent
Manganese, about	0.60 per cent
Silicon	0.15 to 0.25 per cent
Phosphorus, not over	0.03 per cent
Sulphur, not over	0.04 per cent
Physical Characteristics	
Tensile strength, not less than	70,000 lb. per sq. in.
Elastic limit, not less than	35,000 lb. per sq. in.
Elongation, not less than	25 per cent in 2 in.
Contraction, not less than	40 per cent in area

TABLE 3 NICKEL-STEEL SHAFT SPECIFICATIONS

Chemical Analysis	
Same as for carbon open-hearth steel, except that the carbon was reduced to about 0.30 to 0.35 per cent, while 3 to 3.5 per cent nickel was added.	
Physical Characteristics	
Tensile strength, not less than	80,000 lb. per sq. in.
Elastic limit, not less than	45,000 lb. per sq. in.
Elongation, not less than	20 per cent in 2 in.
Contraction, not less than	35 per cent in area

54 Tests *B* and *C* were made in what is admittedly the weakest portion of the shaft forging, but even so the low elongation of less than 15 per cent is so far below the specified limit as to be noteworthy. Attention also is called to the fact that while the physical tests of bars *A* and *C* show good and bad metal respectively, the chemical analyses of the specimens were almost identical, although both show the carbon to be far in excess of specifications. This shaft failure is therefore in accord with numerous other experiences, in which high carbon combined with nickel has proved unsatisfactory in service.

55 The web fracture showed a rather large dull spot, pointing to a possible initial flaw such as may have been caused by insufficient heat soaking during the forging period. The high tensile strength and low contraction noted in bar *C* indicate defective heat treatment. No further tests were made on the broken nickel-steel shaft, but had microphotographic investigations been undertaken, they would no doubt have revealed an improper carbon condition.

56 Low elongation signifies brittleness and lack of capacity to absorb shock. Care should be taken to insure high-grade metal in the web portion of a crankshaft in particular. A first-class forging suitable for gas-engine work should show an elongation of at least 20 per cent and a contraction of not less than 35 per cent as made on a test bar taken at half radius. A reduction of about 10 per cent from these values would represent a fair allowance as leeway for test

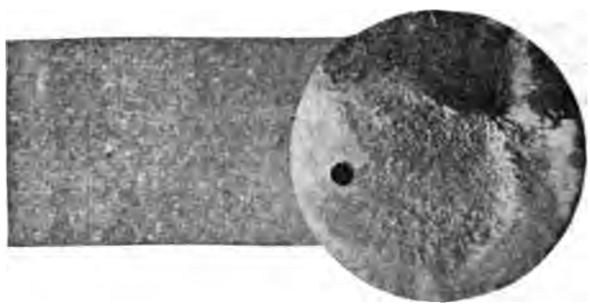

FIG. 6 FRACTURE OF KRUPP WHEEL SHAFT

bars taken out of the web or discard metal near the critical core section.

57 It is apparent that elongation becomes the vital factor of a crankshaft specification, and that this should be kept at a maximum even at the expense of tensile strength. The importance of this deduction is shown by the present experience, where a nickel-steel forging broke under conditions identical with those withstood for years by a plain carbon-steel shaft having a much lower tensile strength.

CRANK-WEB STRESS CHECK

58 The maximum stress occurring in the broken crank web appears to be considerably lower than that found for the wheel-shaft parts. A complete treatment of such stress relations is rather in-

volved, owing partly to the complex deformations that are likely to be set up in the crank throw.

59 The following check relates to the web stresses existing in the original Krupp carbon-steel shafts which withstood these stresses without apparent fatigue. The worst stress at the point of rupture *B–B*, Fig. 3, occurs when the crankpin reaches its inner dead center position. At this instant the counterweight forces act with the explosive pressure, but this is practically balanced by the inertia of the reciprocating parts. Taking the full explosive pressure P_e = 175,000 lb. as acting in the center of the crankpin, and combining the stress effect of this force with that produced by the wheel weight as indicated in Fig. 3, the resulting equivalent twisting or shear stress

TABLE 4 TESTS ON DEFECTIVE NICKEL-STEEL SHAFT

Test Bar	*A*	*B*	*C*
CHEMICAL ANALYSIS			
Carbon	0.445 per cent	....	0.475 per cent
Phosphorus	0.029	....	0.029
Manganese	0.649	....	0.636
Silicon	0.174	....	0.173
Sulphur	0.031	....	.....
Nickel	2.49	....	2.49
PHYSICAL CHARACTERISTICS			
Tensile strength	90,500 lb. per sq. ft.	97,800	97,000 lb. per sq. in.
Elastic limit	38,000	46,100	45,000
Elongation	22 per cent	14.5%	14.0 per cent
Contraction	38.5	23.1%	35.1

is found to be about 11,700 lb. per sq. in. without allowance for shock or preignition. For an assumed shock factor equal to $\frac{4}{3}$, the corresponding maximum stress would be about 15,500 lb., and even under conditions of preignition the web stress would probably not exceed 20,000 lb. per sq. in.

60 The resulting shear stress found above is only partially reversing. Taking the allowable stress limit at about $\frac{4}{5}$ of the 35,000 lb. elastic limit, and multiplying this by $\frac{4}{5}$ to allow for the reduced strength of metal in shear, then f_{lim} as determined in the Appendix for unit net factor of safety, becomes equal to about 22,000 lb. per sq. in. When working under normal conditions at about 15,500 lb. per sq. in. the net or basic factor of safety of the web therefore

appears to be 1.4, while under occasional preignition the stress would hardly reach the limit set for unit factor of safety. The fact that none of the original Krupp forgings broke in the web parts proves that they must have been operating at a greater margin of safety than the wheel-shaft parts, which according to our previous estimate were found to be working under a net or basic factor of safety of 0.9.

61 Even for equal safety factors, the web section is more likely to fail than the other shaft parts because the segregated core impurities tend to lodge in the surface metal of the web sections. The throw slab from which the web is cut can be worked in one direction only, and to insure reasonably close metal in this part of the forging a forging reduction of about 50 per cent is required. A much larger reduction would increase the relative size of the ingot core.

CRANK-WEB PROPORTIONS

62 Another reason for liability to fracture in the web section lies in its comparatively greater rigidity, which has the effect of localizing shock and preventing uniform stress distribution. The rigidity factor is to a certain extent taken care of by using empirical formulæ to fix the web proportions. In terms of the crankpin diameter D_{pin} the original Krupp shaft design, these proportions were:

Crankpin length $= 1.09\ D_{pin}$
Web dimensions:
Short length $b = \frac{3}{4}\ D_{pin}$
Long length $h = 1.16\ D_{pin}$

63 Good practice requires that the crank webs be made of liberal proportions, since failure in this vital part is likely to be not only abrupt and unexpected, but disastrous in its results. From the foregoing experience it would follow that the web dimension h for a solid forged center-crank shaft carrying a heavy wheel should be increased to at least 1.25 D_{pin}, so as to afford better protection against possible forging flaw and uncertainty of metal in this inherently weak section.

64 After fixing the web dimensions, it is advisable to check the shaft throughout for net or basic factor of safety and to provide a reasonable increase in the estimated web strength over that allowed in the other portions of the shaft. As a further precaution, the shaft should be checked against permanent injury in case the babbitt

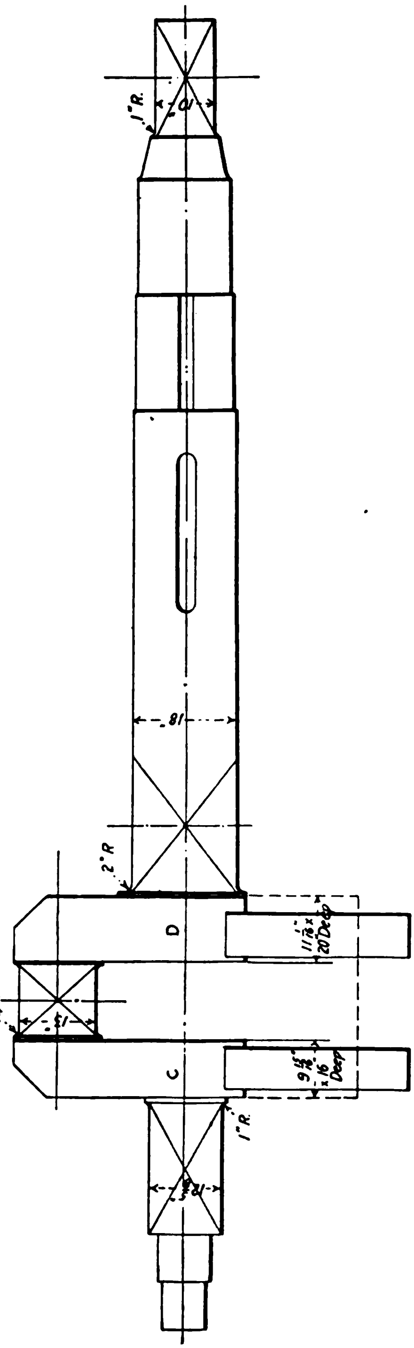

Fig. 7 Outline of Reinforcements in Final Shaft Design

Forging Specifications: Forge shaft from solid ingot without welds; forge web end of shaft from bottom end of ingot; forge with double web lugs as shown dotted and keep core of ingot in center line of wheel shaft $C-D$; metal to be thoroughly worked and free from flaws of any kind.

melts out of either the side or bottom blocks of the intermediate main bearing.

FINAL DESIGN FOR REINFORCED SHAFT

65 Reference is again made to the nickel-steel shaft that broke in the web. This shaft was replaced by one made of open-hearth

TABLE 5 TESTS ON CARBON-STEEL SHAFTS

Chemical Analysis	First Shaft	Second Shaft
Carbon	0.294 per cent	0.36 per cent
Silicon	0.208	0.25
Phosphorus	0.017	0.018
Sulphur	0.019	0.021
Manganese	0.569	0.629

Physical Characteristics	First Shaft	
	Outboard End at Half Radius	From Discard, taken 1 ft. off Center Line
Tensile strength	70,500 lb. per sq. in.	65,000 lb. per sq. in.
Elastic limit	36,500	34,500
Elongation	29.5 per cent	28 per cent
Contraction	48.3	43.4

Physical Characteristics	Second Shaft		
	Shaft End at Half Radius	From Center of Crankpin	From Discard near Crankpin
Tensile strength	68,000 lb.	72,000 lb.	67,000 lb.
Elastic limit	45,000	42,000	42,500
Elongation	28.5 per cent	27.0 per cent	23.5 per cent
Contraction	52.6	45.4	35.1

carbon steel but thoroughly reinforced as outlined in Fig. 7. As previously stated, the intermediate journal was enlarged to 18 in. in diameter, which decided increase in dimensions became necessary in order to bring the shaft stresses within the desired limits of 7000 lb. to 8000 lb. per sq. in., as based upon the load conditions shown in Fig. 5.

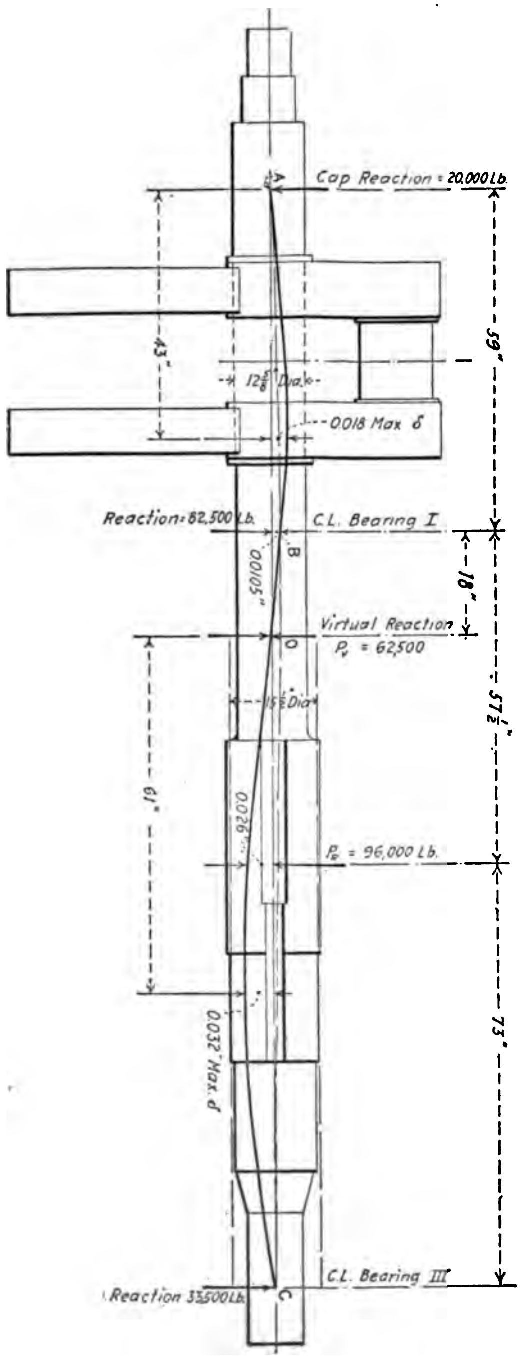

FIG. 8 DIAGRAM SHOWING METHOD USED FOR DETERMINING SINUOUS SHAFT FLEXURE

66 It is apparent that the 18-in. diameter is considerably larger than is customarily used for center-crank shaft proportions for gas engines operating at an explosive pressure of about 300 lb. per sq. in. The given diameter is fully as large as that required for a side-crank shaft, hence for large stationary engines carrying a heavy wheel it is found more satisfactory, both from a cost and constructive standpoint, to adopt the side-crank type of shaft since this construction eliminates the shaft trouble encountered in the use of a three-point bearing support.

67 The dotted web outline and forging specifications given in Fig. 7 indicate the manner in which this last shaft ingot was drawn down in order to keep the core impurities out of the web section.

68 The specifications for this reinforced carbon-steel shaft were the same as those set forth in Table 2. The first one of these shafts was lost by the maker due to an error in machining, and the material tests upon both the first and second shafts of this design are given in Table 5.

69 It will be seen that the first forging runs a little low in carbon but otherwise shows excellent test qualities. In making the second shaft, it was thought advisable to increase the carbon somewhat, but the elongation in the test piece cut from the discard did not show up so well.

70 The test values lie within the specifications given in Table 2, and since special care was taken in making these forgings they probably represent as good results as can reasonably be expected in large shafts of this kind. This reinforced carbon-steel shaft has been in service for several years and appears to have fully overcome the shaft troubles discussed in this paper.

71 The author desires to take this opportunity to express his indebtedness to Mr. Walter Bohrer for valuable assistance in working out some of the diagrams and to Mr. George H. Crocker for helpful criticisms.

APPENDIX

SAFETY FACTORS

72 Shaft metal will deteriorate from fatigue when subjected to a stress range exceeding certain well-defined limits. The character of loading is also a determining factor; shock effects from percussive loading or frequent repetitions of a gradually applied load tend to restrict the stress range to which a machine member may safely be subjected.

73 As based upon Johnson's formula,[1] an alternating stress resulting from a gradually applied reversing load, may be indefinitely repeated provided the stress is kept within the following limit:

$$f_{\text{lim.}} = \frac{\text{E. L.}}{1 + \frac{1}{2}\frac{f_{\text{min.}}}{f_{\text{max.}}}} = \frac{\frac{1}{2}\text{ U. S.}}{1 + \frac{1}{2}\frac{f_{\text{min.}}}{f_{\text{max.}}}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [A]$$

where

$f_{\text{min.}} \div f_{\text{max.}}$ = ratio of minimum to maximum intensity of an alternating stress applied without perceptible shock.

$f_{\text{lim.}}$ = limiting tension or compressive stress per square inch permissible without producing metal fatigue after an infinite number of repetitions of such load.

E. L. = elastic limit or yield point of the material which for mild-steel shaft forgings is approximately equal to ½ ultimate tensile strength, i.e., ½ U. S.

74 Equation [A] assumes equal strength for the material in resisting either tension or compressive stresses and holds true only within the elastic limit. The low ratio of E. L. = ½ U. S. insures safe construction and reasonably guards against impairing the elasticity or otherwise injuring the reliability of the material.

75 According to equation [A], when the stress varies in one direction only, i.e., repeated from $f_{\text{min.}}$ = zero to $f_{\text{max.}}$, then $f_{\text{lim.}}$ becomes equal to ½ U. S., while for reversing stress alternating between $f_{\text{max.}} = f_{\text{min.}}$, then $f_{\text{lim.}}$ is reduced to ⅓ U. S. For intermediate values of a reversing stress, in which $f_{\text{max.}}$ does not equal $f_{\text{min.}}$, the allowable stress $f_{\text{lim.}}$ will lie between ⅓ and ½ U. S., the limiting stress range without ultimate fatigue of material being confined to

$$f_{\text{max.}} + f_{\text{min.}} = \left(\frac{1}{2} + \frac{1}{6}\frac{f_{\text{min.}}}{f_{\text{max.}}}\right)\text{U. S.} \quad . \; . \; . \; . \; . \; . \quad [B]$$

76 The stress limit is furthermore dependent upon a shock factor, since a suddenly applied or percussive load may subject the material to a very considerable increase in stress. It therefore becomes necessary to red[illegible] given by equation [A] in direct proportion to the increase of th[illegible]

77 While the element of shock is present in all movin[illegible] effect is especially important when the machine memb[illegible] deformations, as is the case in a crankpin and like [illegible] bent back and forth out of their neutral position[illegible] true wheel shaft running at moderate speed [illegible]

[1] Johnson's Materials of Construction.

effect because the sag is constrained and the stress more or less gradually applied as the outer shaft fibers pass from the top to the bottom position.

78 Unity shock factor S implies a gradually applied load, while instantaneous application of load or full shock effect will double the stress produced by a gradual application of the same load. In practice the shock factor for smooth running machinery may therefore be expected to lie between 1 and 2 as a limit.

79 The shock factor that should be allowed in engine design is largely dependent upon speed. On the basis of the table of stress limits given by Seaton for marine engine design, the shock factor S works out approximately to,

$$S = \frac{1}{\sqrt[10]{N}} \quad \ldots\ldots\ldots\ldots \quad [C]$$

where N = revolutions per minute.

80 For usual engine speeds, this factor S accordingly lies between the limits 1.25 and 1.5. In engine work the shock factor is dependent to some extent upon the rigidity of frame structure and relative weight of flywheel, but ample protection as regards shock is generally afforded in work of this character by taking the factor S at about $\frac{4}{3}$.

81 Before applying the above deductions to any machine member, it is customary to further reduce the working stress by a suitable safety margin in order to guard against error in estimating stresses. This allows for internal strains and other hidden defects in material, and for unusual strains caused by ordinary accidents, wear and like contingencies that may arise in operation under reasonably competent supervision.

82 These elements of uncertainty may best be combined in a basic or net factor of safety. In engine design a margin that is usually considered consistent with safety and economy of material fixes the basic or net factor of safety at about 3 for worked metal parts, such as mild-steel forgings and the like, while for less homogeneous and uncertain metals, such as cast iron or steel castings, this factor should be increased to 4. For such parts as require considerable rigidity in addition to strength, the basic factor of safety is generally taken at twice the stipulated values.

83 When the working stress $f_{\text{max.}}$ does not exceed $f_{\text{lim.}}$, the net or basic factor of safety as based upon equation [A] becomes equal to

$$\text{F. S.}_{\text{net}} = \frac{f_{\text{lim.}}}{f_{\text{max.}}} \cdot \frac{1}{S} \quad \ldots\ldots\ldots\ldots \quad [D]$$

84 This net margin of safety may be defined as the ratio of stress, i.e., load which any machine member is capable of sustaining continuously without fatigue of material or other serious injury to the load this machine part may reasonably be expected to carry under ordinary conditions of service.

85 As based upon an E. L. = $\frac{1}{2}$ U. S., a shock factor of $\frac{4}{3}$, and a net factor of safety specified above, the gross factors of safety F.S._g, for varying characters of load, assume the values for engine parts as given in Table 6.

86 The working stress allowable in any engine part will naturally vary with the U. S. of the material used, and therefore becomes equal to

$$f_{\text{allow}} = \frac{\text{U.S.}}{\text{F.S.}_g} \quad \ldots\ldots\ldots\ldots \quad [E]$$

87 In fixing the proportions for any machine member, the requirements of rigidity may further restrict the prescribed stress limits. On the other hand, an experienced designer may, under conditions of restricted space or weight and like limitations, be justified in reducing the size of certain machine parts and work with a smaller factor of safety, especially when definite factors of safety have previously been established by experience in a similar line of machinery.

88 In a new line of machinery, the selection of a close margin of safety requires expert judgment, but even then a considerable departure from the prescribed limits for the basic factor is warranted only to the degree that the stress relations, the character of material, and conditions of service can be definitely predetermined.

89 In order to allow for such contingencies as are likely to arise in its manufacture and operation, the basic or net factor of safety for any forged machine member as fixed by Eq. [*D*] should in no case be allowed to fall below a minimum value of two. A serious mistake made in the selection of the safety factor may lead to ultimate rupture of the machine member, and it is therefore incumbent upon the designer to insure against this possibility, especially in such parts as are likely to involve heavy consequential damages or danger to life.

TABLE 6 GROSS FACTORS OF SAFETY = $F.S._g$

Load Conditions		Material Specifications	
Character	Stress Range	Forgings $F.S._{net} = 3$	Castings $F.S._{net} = 4$
Steady or dead load	0 to 0	$3 \times 1 = 3$	4
Repeated — no shock	0 to $\pm f$	$3 \times 2 = 6$	8
Repeated — $S = \frac{4}{3}$	0 to $\pm f$	$6 \times \frac{4}{3} = 8$	10
Reversing — no shock	$-f$ to $+f$	$3 \times 3 = 9$	12
Reversing — $S = \frac{4}{3}$	$-f$ to $+f$	$9 \times \frac{4}{3} = 12$	16

90 When for any reason the stress relations are allowed to rise to such an extent that $f_{max.}$ exceeds the stress $f_{lim.}$ as fixed by the above formulæ, the net or basic factor of safety will assume a value less than unity in the sense that this must lead to ultimate fatigue and breakdown of the machine part. In that event the alternating stress would exceed the limit set by equation *A*, and the resulting excess stress $f_{exc.}$ on the basis of unit shock factor *S*, becomes equal to

$$f_{exc.} = f_{max.} - f_{lim.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [F]$$

91 Some idea as to how fast shaft metal may be expected to fatigue under excessive alternating stress, is shown by experiments of Woehler, Stromeyer, and others. In analyzing the results obtained by Stromeyer[1] the probable number of revolutions required to produce rupture of a loaded iron or mild-steel wheel shaft when rotating without perceptible shock effect is approximately:

$$n = \left(\frac{300,000}{f_{exc.}}\right)^4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [G]$$

[1] The Law of Fatigue applied to Crankshaft Failures. Trans. Inst. Naval Architects, vol. lvii, 1915.

This formula serves as a fair check on past performances of iron or mild-steel shafts that have proved defective in service.

92 When a wheel shaft is overstrained, the outer or surface fibers become permanently stretched, and thus set up internal stresses which irreparably injure the metal. The stress distribution under load is no longer directly proportional to the shaft radius and the repeated application of such excess stress causes the internal strains to creep inward. This action, together with the incipient micro-flaws, gradually bring about a cumulative weakening effect upon the shaft metal that ultimately produces rupture of the fibers across the plane of maximum stress.

DISCUSSION

John B. Peddle. The writer would like to get the details of the author's computations for the maximum bending stress of 26,600, as shown in Fig. 3 of his paper.

Taking the values marked on the figure, the reaction at III is 53,000 lb., and the distance to the section under discussion is 100.5 in. The moment is therefore $53{,}000 \times 100.5 = 5{,}326{,}500$. The sum of the weights of the flywheel, armature and shaft together with the magnetic pull is given at $96{,}000 + 5000/2 = 98{,}500$, and the distance of this resultant force from the section is $100.5 - 73 = 27.5$. The product of the load and the distance is 2,708,800. The moment of the bearing reaction being of opposite sign to the load moment the latter is subtracted from the former, leaving a net moment of 2,617,700 on the section. In Fig. 1 the diameter of the shaft for the section considered is given as $12\frac{5}{8}$ in. Its section modulus, according to the writer's calculations, is 197.5, or, say, 200 in round numbers. The fiber stress would then be $2{,}617{,}700 \div 200 = 13{,}088$, about half the value found by the author.

The moment of the reaction only is 5,326,500. If this is divided by 200 we get 26,632, a value so close to the author's figure that one strongly suspects that it was obtained in this way.

The writer is aware that his method of including the shaft weight with the resultant downward force is not exact. Inasmuch, however, as the shaft weight in question is only 2500 lb. while the flywheel, armature, etc., amount to 96,000 lb., the error would be trivial. The object is not to show that a small mistake has been made but rather that it is in the magnitude of 100 per cent. A rough check on the author's figures is all that it aimed at.

If the writer understands the author's reasoning, it would lead to the conclusion that a beam with a uniformly distributed load and one loaded at the middle would have the same bending moment if the loads, length of beam, etc., were the same.

THE AUTHOR, in reply to Professor Peddle, wrote:

"I have gone over my figures again and cannot find any serious error. I would however point out to Professor Peddle that the conventional stress relations shown in Fig. 1 take no account of counterweight forces and that the resultant P_R at 96,000 lb. shown in Fig. 1 represents the total downward load acting between bearings I and III. This load has been transferred in position and amount to Fig. 3 as indicated and takes care of the wheel load, armature and magnetic pull acting between bearings I and III.

"In Fig. 3, the intermediate bearing support is assumed to be worn away and this throws a part of the wheel shaft load upon bearing II. It is this increase in the effective span that produces the high stress given.

"To find this stress I, include in Fig. 3 the downward load of the counterweights and connecting rod (the latter forces were not taken into account in Fig. 1, as pointed out in Par. 23 of the paper). If, then, moments are taken about either end bearings II or III, it will be found that a gross resultant given as P_t = 143,000 lb. equal to the entire downward load on the shaft acts at about 89 in. from the center line of bearing II. This gross resultant is the equivalent of all the downward forces acting on the shaft and as taken naturally include the wheel, armature and counterweights, etc. The bearing reactions resulting from this load are distributed as follows: 45,000 lb. on each of the main bearings and 53,000 lb. reaction in the outboard bearing.

"On this basis the shaft may now be taken as a simple beam loaded with P_t, without further considering the component forces making up the resultant P_t, in exactly the same way that the force P_R was used in Fig. 1. If then the bearing reaction in III at 53,000 lb. is multiplied by its arm of 100½ in. to get the moment acting under P_t, and this divided by the modulus of the shaft section at this point it will give the high stress stated in the paper.

"I may add that in addition to this method, each shaft section of uniform diameter was estimated for its component deflection on the stress basis outlined above and then these increment deflections were plotted in the manner indicated in Fig. 8 in order to arrive at the total deflection of the shaft. The results are given in the paper and it will be seen that they check well with the actual deflections as found from actual measurement.

"The fact that all the shafts run for approximately the same period and that this time of run checks with Stromeyer's fatigue data, would further indicate that my calculations are reasonably correct. At any rate, I have not as yet seen any convincing proof presented that any other method of analysis for shaft stress will show why these particular shafts should have failed in service as they did.

"Referring further to Prof. Peddle's discussion, I can see no reason for subtracting a counter moment from the 5,326,500 in. lb. bearing moment as advocated by him. As I see it this would only be true provided there were a support directly under the resultant load marked P_R."

C. D. ALBERT. The paper is descriptive of rather an unusual experience. It is not very often that an opportunity of the kind described comes to a designing engineer. It is rather unusual, after speculation and analysis of the probable causes of the failure of a member, that the probable stresses and deflections as computed may be checked by actual observation and measurement for an identical member under precisely the same working conditions. Considering the complexity of the member, it is quite surprising that the usual analytical methods yielded results in such close agreement with the actual observations and measurements.

Johnson's equation for repeated stress action has been used as though it gave the limiting stress for an indefinitely great number of repetitions. It would be fair to say that the equation gives for any case the limit stress for from four to ten million repetitions, — say seven million.

Moore and Seely [1] found that a decrease of about 9 per cent in the limit stress doubled the number of repetitions to cause failure. According to Basquin,[2] the limit stress varies with the number of repetitions for failure as a straight line when the results are plotted on logarithmic paper. Accepting these two statements, J. B. Kommers [3] shows that a very simple relation is established between the ratio of repetitions, r, and the ratio of limit stresses, c.

$$0.137 \log r = \log c$$

Assuming as the author does that up to failure the shaft had made about 100-million repetitions, would give us a ratio of the

[1] *Proc.* Amer. Soc. Test. Mat. 1916, vol. 16.
[2] *Proc.* Amer. Soc. Test. Mat. 1910, vol. 10.
[3] *Eng. News Record,* vol. 83, no. 20.

limit stresses of 1.44 to 1.37 depending upon whether Johnson's equation is assumed to hold for 7-or 10-million repetitions.

Taking $(\frac{1}{2}u)$ in the Johnson equation as standing for one-half the ultimate static strength of the material, and as standing also for the elastic limit stress, would give for the limit stress for 100-million repetitions:

$$p = \frac{\frac{1}{2}(65{,}600)}{1-\frac{1}{2}(-.62)\,1.44} \text{ to } \frac{0.6 \times 65{,}600}{1-\frac{1}{2}(-.62)\,1.37} =$$
$$= 17{,}400 \text{ to } 22{,}000$$

Considering the nature of the problem and the present state of our knowledge of repeated stresses, this range for the limiting stress is in fair agreement with the observed facts and the analytical results to be found in the author's paper.

C. L. Newcomb cited an experience of his with a five-throw crankshaft made of a steel casting. It had been made amply strong according to the formulæ and to the best judgment of the engineers as it was imperative that it should continue in use without accident for as long a period as possible. After the first machine using this crankshaft had been placed in use, other machines were built of the same type. At the end of a four-year period, almost to a day, the first crankshaft broke, and subsequently others broke, all having approximately the same life of four years. Eliminating the fact that a forging would have been more desirable than a steel casting, what could there have been about these shafts to have caused failure at the end of this period?

Gardner T. Voorhees asked if investigations were being carried on in this country to determine the life of engine parts, such as crankshafts, similar to those undertaken by the British Admiralty. The purpose of the British Admiralty's investigations, he said, was to make it possible to determine when to remove a part, realizing maximum useful life, but forestalling breakage. He thought it highly advisable that engineers of this country undertake a similar research.

P. J. O'Reilly said that the United States Government had undertaken no research of the character of that of the British Admiralty. He thought it a necessary undertaking.

S. A. Moss said that some experiments on the length of life of Liberty motor shafts had been made at the University of Illinois during the war and some formulæ had been arrived at in which a number of repetitions of stress had been considered. "Such a formula, including the number of repetitions of stress, will be the ultimate formula, I think. For instance, if we have a steam turbine wheel rotating at 3600 r.p.m. for a number of years, we would figure a certain allowable stress. On the contrary, with a shaft running at one per cent of this speed, the allowable stress would be considerably higher. Work is being done along these lines in this country and some day there will be evolved a final formula based upon the number of repetitions of the stresses."

E. J. KATES. In Par. 24 and 25, the author states that if the bearings were in alignment and the main bearing caps were pulled down into place, the stresses in the shaft as originally designed were not excessive. However, he then shows that the dangerous stresses which broke the shaft occurred when the intermediate main bearing, I, wore down about $\frac{1}{8}$ in, so that a considerable portion of its load was transferred to the outer bearings II and III. Is the author certain that this excessive wear of the intermediate main bearing in the remarkably short time of one month was not due to poor babbitt or imperfect lubrication?

THE AUTHOR. The aim in presenting this paper is to give the Society the benefit of some disastrous shaft experiences and to call attention to the need of improved methods for fixing the proper size of center-crank wheel shafts. This matter is of considerable importance to engine builders since shaft breakdown is likely to lead to heavy damage suits.

It is evident that the conventional stress analysis shown in Fig. 1 does not offer a dependable method for determining underlying stress conditions in a center-crank shaft carrying a heavy flywheel.

The given technical investigation was originally made simply with a view of locating the cause of this particular shaft failure and of arriving at a logical basis for correcting the defect. No attempt was made elaborately to investigate the behavior of material working under high repeated stress but it is hoped that this paper may stimulate sufficient interest to lead others to carry on further development along these lines. What the engine builder needs is a safe yet simple method for calculating stresses in large center-crank wheel shafts so that these shafts may be economically built without the possibility of ultimate breakdown.

APPENDIX

SAFETY FACTORS

72 Shaft metal will deteriorate from fatigue when subjected to a stress range exceeding certain well-defined limits. The character of loading is also a determining factor; shock effects from percussive loading or frequent repetitions of a gradually applied load tend to restrict the stress range to which a machine member may safely be subjected.

73 As based upon Johnson's formula,[1] an alternating stress resulting from a gradually applied reversing load, may be indefinitely repeated provided the stress is kept within the following limit:

$$f_{\text{lim.}} = \frac{\text{E. L.}}{1+\frac{1}{2}\frac{f_{\text{min.}}}{f_{\text{max.}}}} = \frac{\frac{1}{3}\text{ U. S.}}{1+\frac{1}{2}\frac{f_{\text{min.}}}{f_{\text{max.}}}} \quad \ldots\ldots\ldots \quad [A]$$

where

$f_{\text{min.}} \div f_{\text{max.}}$ = ratio of minimum to maximum intensity of an alternating stress applied without perceptible shock.

$f_{\text{lim.}}$ = limiting tension or compressive stress per square inch permissible without producing metal fatigue after an infinite number of repetitions of such load.

E. L. = elastic limit or yield point of the material which for mild-steel shaft forgings is approximately equal to $\frac{1}{3}$ ultimate tensile strength, i.e., $\frac{1}{3}$ U. S.

74 Equation [A] assumes equal strength for the material in resisting either tension or compressive stresses and holds true only within the elastic limit. The low ratio of E. L. = $\frac{1}{3}$ U. S. insures safe construction and reasonably guards against impairing the elasticity or otherwise injuring the reliability of the material.

75 According to equation [A], when the stress varies in one direction only, i.e., repeated from $f_{\text{min.}}$ = zero to $f_{\text{max.}}$, then $f_{\text{lim.}}$ becomes equal to $\frac{1}{3}$ U. S., while for reversing stress alternating between $f_{\text{max.}} = f_{\text{min.}}$, then $f_{\text{lim.}}$ is reduced to $\frac{2}{9}$ U. S. For intermediate values of a reversing stress, in which $f_{\text{max.}}$ does not equal $f_{\text{min.}}$, the allowable stress $f_{\text{lim.}}$ will lie between $\frac{1}{3}$ and $\frac{2}{9}$ U. S., the limiting stress range without ultimate fatigue of material being confined to

$$f_{\text{max.}} + f_{\text{min.}} = \left(\frac{1}{2} + \frac{1}{6}\frac{f_{\text{min.}}}{f_{\text{max.}}}\right) \text{U. S.} \quad \ldots\ldots \quad [B]$$

76 The stress limit is furthermore dependent upon a shock factor, since a suddenly applied or percussive load may subject the material to a very considerable increase in stress. It therefore becomes necessary to reduce $f_{\text{lim.}}$ as given by equation [A] in direct proportion to the increase of the shock factor S.

77 While the element of shock is present in all moving machine parts, its effect is especially important when the machine member is subject to varying deformations, as is the case in a crankpin and like parts, which are successively bent back and forth out of their neutral position. On the other hand, a perfectly true wheel shaft running at moderate speed suffers comparatively little shock

[1] Johnson's Materials of Construction.

effect because the sag is constrained and the stress more or less gradually applied as the outer shaft fibers pass from the top to the bottom position.

78 Unity shock factor S implies a gradually applied load, while instantaneous application of load or full shock effect will double the stress produced by a gradual application of the same load. In practice the shock factor for smooth running machinery may therefore be expected to lie between 1 and 2 as a limit.

79 The shock factor that should be allowed in engine design is largely dependent upon speed. On the basis of the table of stress limits given by Seaton for marine engine design, the shock factor S works out approximately to,

$$S = \frac{1}{\sqrt[10]{N}} \quad \ldots\ldots\ldots\ldots\ldots \quad [C]$$

where N = revolutions per minute.

80 For usual engine speeds, this factor S accordingly lies between the limits 1.25 and 1.5. In engine work the shock factor is dependent to some extent upon the rigidity of frame structure and relative weight of flywheel, but ample protection as regards shock is generally afforded in work of this character by taking the factor S at about $\frac{4}{3}$.

81 Before applying the above deductions to any machine member, it is customary to further reduce the working stress by a suitable safety margin in order to guard against error in estimating stresses. This allows for internal strains and other hidden defects in material, and for unusual strains caused by ordinary accidents, wear and like contingencies that may arise in operation under reasonably competent supervision.

82 These elements of uncertainty may best be combined in a basic or net factor of safety. In engine design a margin that is usually considered consistent with safety and economy of material fixes the basic or net factor of safety at about 3 for worked metal parts, such as mild-steel forgings and the like, while for less homogeneous and uncertain metals, such as cast iron or steel castings, this factor should be increased to 4. For such parts as require considerable rigidity in addition to strength, the basic factor of safety is generally taken at twice the stipulated values.

83 When the working stress $f_{max.}$ does not exceed $f_{lim.}$, the net or basic factor of safety as based upon equation [A] becomes equal to

$$\text{F. S.}_{net} = \frac{f_{lim.}}{f_{max.}} \cdot \frac{1}{S} \quad \ldots\ldots\ldots\ldots\ldots \quad [D]$$

84 This net margin of safety may be defined as the ratio of stress, i.e., load which any machine member is capable of sustaining continuously without fatigue of material or other serious injury to the load this machine part may reasonably be expected to carry under ordinary conditions of service.

85 As based upon an E. L. = $\frac{1}{2}$ U. S., a shock factor of $\frac{4}{3}$, and a net factor of safety specified above, the gross factors of safety F.S.$_g$, for varying characters of load, assume the values for engine parts as given in Table 6.

86 The working stress allowable in any engine part will naturally vary with the U. S. of the material used, and therefore becomes equal to

$$f_{allow} = \frac{\text{U.S.}}{\text{F.S.}_g} \quad \ldots\ldots\ldots\ldots\ldots \quad [E]$$

No. 1762

DESIGN OF FLYWHEELS FOR RECIPROCATING MACHINERY CONNECTED TO SYNCHRONOUS GENERATORS OR MOTORS

By R. E. Doherty [1] and R. F. Franklin,[1] Schenectady, N. Y.

Non-Members

This problem, although long since understood in a fundamental way, has never been completely worked out. It has been largely a matter of chance whether the flywheel installed has given trouble or satisfactory operation. As covered by the literature, the problem has not been sufficiently understood by many of the engine and compressor builders, or by generator and motor builders. But what is more important, most of those who have understood the problem have found not only a lack of reliable design constants necessary for the calculation, but also that the calculation itself was too involved and laborious to be carried out. Yet all will agree that if the thing can be calculated with practical accuracy and expedition, it would be highly desirable.

The purpose of this paper is to present a way to do this. It gives in part the results of a rather extended investigation as outlined in Par. 3 below. The results given here are: (a) *a complete solution of the problem of flywheel moment as related to the crank effort, synchronizing force and damping force;* (b) *modification of* (a) *for practical calculation;* (c) *a short method for determining the crank effort from the engine or compressor design;* (d) *practical method of determining the "initial angular deviation" from the crank-effort curve;* (e) *tests confirming the equations; and* (f) *discussion of the electrical forces involved and of the limitations which must be imposed upon power fluctuations.*

THE problem of flywheel design for reciprocating machinery connected to synchronous generators or motors is an old one, dating practically from the advent of synchronous machines in the commercial field. Much thought was given to it both by electrical and mechanical engineers, especially during the period 1900–1905. Taking the literature in that period as a whole, the problem in all fundamental respects is completely covered. The numerous contributions, as listed in the accompanying bibliography, indicate the interest which was taken in the problem. Emmet, Cooke and Seymour did much of the pioneer work, especially on

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the mechanical side, which resulted in the practical operation of a. c. generating units. The solution of the governor problem was a very important part of their contribution. Steinmetz, Berg, Astrom, Abbott, Slichter, Boucherot and others developed the theory underlying the problem, both electrically and mechanically. The most complete statement of the problem seems to be that by Hawkins and Wallis,[1] which is more or less a résumé of many other papers listed in the bibliography.[2]

2 However, although the fundamental theory had thus been worked out and most of the units have operated successfully, it is nevertheless a fact that the situation has not been and is not now satisfactory. It has been largely a matter of chance whether the flywheel installed has given trouble or satisfactory operation. Cases of trouble continue to appear and unquestionably as many more exist undetected, and, like the insidious crystallization of metals, the slow process of mischief carries on, appearing finally in the form of a mechanical failure or an electrical breakdown of insulation. In the first place, the problem, as covered by the literature, has not been sufficiently understood by many of the engine and compressor builders, or by generator and motor builders. But what is more important, most of those who have understood the problem have found not only a lack of reliable design constants, necessary or the calculation, but also that the calculation itself was too involved and laborious to be carried out. Yet all will agree that if the thing can be calculated with practical accuracy and expedition it would be highly desirable. And it is possible.

3 The purpose of the paper is to present a way to do this. It gives in part the results of a rather extended investigation covering (1) wide inspection and tests of actual operating conditions, (2) a thorough review of the theory involved, (3) a coöperative study with engine and compressor builders in the interest of acquiring a comprehensive and practical viewpoint, (4) a method of accurately and easily obtaining the necessary constants for synchronizing force,[3] (5) a search for the most readily applied, and yet accurate, method of making the necessary flywheel calculations from the engine or compressor design, and (6) a satisfactory arrangement for interchanging essential design data between the generator and engine builders.

[1] The Dynamo, vol. 2, chap. 26, 5th edition.

[2] This bibliography and Appendices B and D are omitted from the present preliminary publication of the paper.

[3] This will probably be given later in an A.I.E.E. paper.

4 The results given here are: (*a*) a complete solution of the problem of flywheel moment as related to the crank effort, synchronizing force and damping force; (*b*) modification of (*a*) for practical calculation; (*c*) a short method for determining the crank effort from the engine or compressor design; (*d*) practical method of determining the "initial angular deviation" from the crank-effort curve; (*e*) tests confirming the equations; and (*f*) discussion of the electrical forces involved and of the limitations which must be imposed upon power fluctuations.

STATEMENT OF THE PROBLEM

5 Synchronous machines on the same system must run at the same average electrical speed, that is, at the same frequency. If one of them tends to run ahead of the system, a force is exerted upon the rotor to hold it back; if it lags, the force tends to pull it up to the zero position.[1] This is the *synchronizing force*. It thus acts like a spring, the magnitude of the force being proportional and opposite to the displacement. The speed can momentarily fluctuate without causing the machine to break synchronism, provided the fluctuation does not involve an angular displacement from the zero position greater than a half pole pitch. For normal operation the fluctuation must, of course, be much less than that.

6 Fluctuating driving torque means fluctuating speed and therefore fluctuating angular position. The latter is accompanied by corresponding pulsations in electric current and proportional pulsations in the electromagnetic force; that is, the synchronizing force or torque.

7 How much current fluctuation shall be accepted as reasonable and safe? One or two per cent is, of course, absolutely harmless. On the other hand, 100 per cent would obviously be out of the question. Considerations such as the possible voltage fluctuations, increased loss and heating, fatigue of materials, particularly the insulation, subjected to the incessant reversal of stress — these considerations must determine the allowable pulsation. While experience has perhaps not yet finally settled the matter, it now seems that the pulsation should never be greater than 50 to 60 per cent of normal current, and should be 10 or 20 per cent where it is not a hardship. This means that the total actual angular fluctuation during opera-

[1] Position which the rotor would occupy at no load with constant driving torque.

tion must be limited to the order of 6 to 10 electrical degrees,[1] that is, 3 to 5 deg.[2] plus or minus deviation; and to a smaller value where it is not a hardship. Usually generators should be held to smaller fluctuations than motors, because the latter are generally of smaller relative capacity and therefore do not to the same extent introduce oscillations in the line voltage which might cause unstable operation of rotary converters or other synchronous apparatus on the system.

8 Engine governors are, of course, a very important factor in the proper operation of generators. If the natural period of the governor is near that of one revolution, or near the natural period of the unit in the case of a generator not equipped with a damper winding, the governor may feed any oscillation which may be started, thus sustaining or augmenting it. Or instability may occur if the governor is too sensitive, thus allowing continual readjustments to follow momentary speed changes. But it has been demonstrated[3] that a suitable dashpot will prevent trouble from such sources. Hence if such dashpots are provided, or provision is made for adding them after the unit is installed, the governor is eliminated as a factor in the present problem.

9 Therefore the problem becomes: From known fluctuating forces acting upon a free mass, determine the moment of inertia required to limit the angular fluctuation or deviation to a given value.

FORCES AND FUNDAMENTAL EQUATIONS

10 Four forces or torques are involved:

a Reaction of mass, $I \dfrac{d^2\theta}{dt^2}$

b Damping torque, $T_d \dfrac{d\theta}{dt}$

c Synchronizing torque, $T_s\theta$

d Variable component of crank torque, $f(t)$

These are related by the differential equation

$$I \frac{d^2\theta}{dt^2} + T_d \frac{d\theta}{dt} + T_s\theta = f(t) \quad . \; . \; . \; . \; . \; . \; . \quad [1]$$

[1] The angular deviation as determined from the crank effort must be less than this, as explained later.

[2] Mechanical degrees equal electrical degrees divided by half the number of poles. Thus on a 48-pole machine, 6 electrical degrees equal 0.25 mechanical degree.

[3] Parallel Operation of Engine-Driven Alternators, by W. L. R. Emmet, Trans. A.I.E.E., vol. 18, p. 745.

11 The crank force can be expressed as a function of time in the form of a Fourier's series;[1] that is, it may be considered as the resultant of several component alternating forces of different frequencies. However, for the purpose of illustration consider that it is a simple sine function,

$$T_o \sin wt \quad \ldots\ldots\ldots\ldots \quad [2]$$

Substituting this in [1] and solving[2] gives the maximum angular deviation from the average position:

$$\theta_m = \frac{T_o}{\sqrt{(T_d w)^2 + (T_s - Iw^2)^2}} \quad \ldots\ldots\ldots \quad [3]$$

where

T_o = maximum value of the unbalanced crank effort, lb-ft.
T_d = damping torque,[3] lb-ft. per rad/sec.
$w = 2\pi n$
n = crank revolutions per second
t = time in seconds
T_s = synchronizing torque[1] = lb-ft. per mechanical radian displacement
I = moment of inertia $= \dfrac{\text{lb-ft.}^2}{32.2}$

θ = angular displacement from the zero, synchronous position

Neglecting all electrical forces the equation would be

$$I\frac{d^2\theta}{dt^2} = T_o \sin wt \quad \ldots\ldots\ldots\ldots \quad [4]$$

and

$$\theta_o = -\frac{T_o}{Iw^2} \quad \ldots\ldots\ldots\ldots \quad [5]$$

is the maximum deviation which would be produced by the crank force acting alone. It is sometimes called the *initial deviation*,[4] since when operation starts, the synchronizing force causes the deviation to gradually build up from this initial value to a larger final value.

[1] Parallel Operation of A. C. Generators Driven by Internal-Combustion Engines, H. C. Lehn, *General Electric Review*, March 1915, p. 174.

[2] The Dynamo, Hawkins and Wallis, vol. 2, p. 1048.

[3] Discussed in Appendix C.

[4] It is the same as the angular oscillation which would occur if, when the rotating element is at a standstill, the alternating component of crank torque alone were applied. The mean position in actual oscillation during rotation is the zero synchronous position.

12 The term, $(T_s - Iw^2)$ appearing in [3] is by [5] the difference between the synchronizing torque corresponding to unit angle deviation and a hypothetical crank torque which, acting alone, would produce unit angle deviation.

13 Since the natural frequency of oscillation is

$$\eta = \frac{1}{2\pi}\sqrt{\frac{T_s}{I}} \text{ periods per sec. } \quad \ldots\ldots \quad [6]$$

[3] may be rewritten

$$\theta_m = \frac{T_o}{\sqrt{(T_d w)^2 + \left\{\frac{T_o}{\theta_o}\left[1 - \left(\frac{\eta}{n}\right)^2\right]\right\}^2}} \quad \ldots\ldots \quad [7]$$

14 Thus, the effect of synchronizing force enters the equation as the natural oscillating frequency. If

$$\eta = n$$

then

$$\theta_m = \frac{T_o}{T_d w} \quad \ldots\ldots \quad [8]$$

that is, limited only by the damping force.[1] On the other hand, if the frequencies η and n are different by 25 per cent, or more, the effect of damping $(T_d w)$ as shown by [7], may usually be neglected in comparison with the other quantity under the radical. If the ratio $\eta \div n \leqq .75$, damping can be neglected for any speed up to 270 r.p.m.; if the ratio is 1.25, damping can be neglected for any speed up to 120 r.p.m. Above these speeds damping cannot be neglected. This assumes $T_d = .01T_s$ and a permissible error of 5 per cent. The error from neglecting $(T_d w)$ is always on the safe side, the calculated value being thereby increased. Thus, neglecting $(T_d w)$,

$$\theta_m = \theta_o \frac{1}{1 - \left(\frac{\eta}{n}\right)^2} \quad \ldots\ldots \quad [9]$$

[1] These equations present an interesting analogy. Those who are familiar with electrical engineering will recognize in the general equation [1] and also in its solution, the relations of the electric circuit. Thus in [3], θ_m corresponds to the flow of current; T_o to the impressed e.m.f.; the radical quantity $\sqrt{(T_d w)^2 + (T_s - Iw^2)^2}$ to the impedance. The quadrature components of the radical, viz. $T_d w$ and $(T_s - Iw^2)$, correspond respectively to the resistance and resultant reactance, the latter being the difference between the inductive reactance and condensive reactance. When these are equal, the resultant reactance is zero and the current is limited by resistance alone; just as at resonance the deviation is limited by damping alone.

15 Consider the limit of zero flywheel moment. Equation [3] becomes

$$\theta_m = \frac{T_o}{\sqrt{(T_d w)^2 + T_s^2}} \quad . \ . \ . \ . \ . \ . \ . \ . \ [10]$$

or, neglecting $T_d w$,[1]

$$\theta_m = \frac{T_o}{T_s} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ [11]$$

16 It is thus possible in the assumed case to explore the whole range of possible flywheel moments. But it is very important to remember that those equations rest on the assumption that the unbalanced component of the crank effort, which causes the deviation, is a simple sine function of the crank frequency. On the contrary, the actual crank effort is made up of several components of different magnitudes and of different frequencies. It will be shown, however, that this method can be adjusted to the problem. Each of the several sine components can be treated separately, as above, and by superposition the resultant deviation may be determined.

PRACTICAL CALCULATION

17 The question of practical calculation is paramount. Two needs must be filled: (*a*) A general method must be available, even if it is more or less involved, by which, if some particular case warrants it, all factors may be properly taken into account. (*b*) There must also be a method and a system of routine design calculations which are at once simple, short and safe. These two cases will be taken up in order.

18 (*a*) *General Method Taking All Factors into Account.* Equation [3] is the solution of [1] assuming the variable component of the crank effort to be a simple sine function. The solution given in Appendix D is general, assuming the crank effort to be made up of a number of simple sine and cosine components of different magnitudes and different frequencies, that is, a Fourier's Series. An easy method of analyzing the crank effort to determine the coefficients of the different components or harmonics is given in Appendix B. Thus the variable crank effort is

$$T = a_1 \sin wt + b_1 \cos wt + a_2 \sin 2wt + b_2 \cos 2wt$$
$$+ a_3 \sin 3wt + b_3 \cos 3wt + \ . \ . \ . \ . \ . \ . \ . \ . \ . \ [12]$$

[1] See Appendix C.

19 The angular deviation due to all electrical and mechanical forces is[1]

$$\theta = \sum \frac{\sqrt{a_m^2 + b_m^2}}{\sqrt{M_m^2 + N_m^2}} \sin (mwt + \psi) \quad . \quad . \quad . \quad . \quad . \quad [35]$$

where

m = order of harmonic. Thus for wt, $m = 1$; for $2wt$, $m = 2$, etc.

$M_m = T_d mw$

$N_m = (T_s - Im^2w^2)$

$$\psi = \text{arc tan} \frac{N_m b_m - M_m a_m}{N_m a_m + M_m b_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [35a]$$

20 That is, each harmonic is treated separately, and the several sine waves, each representing the displacement contributed by the particular harmonic, are added in proper phase relation as given by Equation [**35a**]. The maximum displacement or deviation due to any harmonic is therefore

$$\theta_m = \frac{\sqrt{a_m^2 + b_m^2}}{\sqrt{M_m^2 + N_m^2}} = \frac{T_{mo}}{\sqrt{(T_d mw)^2 + (T_s - Im^2w^2)^2}} \quad . \quad . \quad [35b]$$

21 This will be recognized as the same form as Equation [**3**]. Hence each harmonic is treated just the same as the simple case, Equation [**3**], which was given as an illustration, the only additional factors being m, the order of the harmonic, and the phase angle ψ. A complete application of this equation (see Fig. 1) is given under Calculations and Test of an Actual Installation, Par. 35 et seq. It would probably never be worth while, except as a matter of interest to carry this calculation out in detail. Much of the labor is in determining the phase ψ and plotting the displacement curves to obtain the maximum. It would at least be a good compromise between the proposed methods (*a*) and (*b*) to calculate the maximum for the different harmonics by [**35b**], and add them arithmetically.

22 (*b*) *Shorter Method for Use in Commercial Design.* Although the above method gives a complete solution which may sometimes be desirable, it obviously could not be applied as a routine operation in commercial design. For this business we must reduce the calculation to simplicity; yet it must be safe. It works out that this can be accomplished by applying the simple relations [**2**] to [**8**], inclusive, to the general case. Some sacrifice in accuracy is thus made, but it is justified by the great saving in time and by the fact that the

[1] Derivation in Appendix D.

error is placed on the safe side. It comes about, fortunately, that for this case all design calculations can be based on the initial displacement θ_o, determined by the crank effort alone.

23 For the purpose of confirming these points examine the above relations as applied, on the one hand, to a simple sine function [2], and, on the other hand, to the general case [12]. In the latter the frequencies of the several components, that is the "forced impulse frequencies," are multiples of the crank speed.

24 In this connection recall that θ_o for a simple sine torque whose maximum is T_o is

$$\theta_o = -\frac{T_o}{Iw^2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . [5]$$

and also that if the synchronizing torque T_s is considered and the damping torque T_d is neglected in Equations [3] and [7] the maximum deviation would be,

$$\theta_m = \theta_o \frac{1}{1 - \left(\frac{\eta}{n}\right)^2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . [9]$$

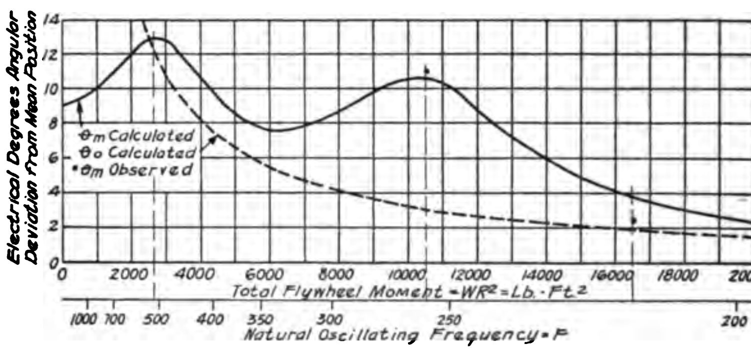


Fig. 1 Curve Showing Effect of Electrical Forces upon Angular Deviation of a 257-r.p.m. Compressor Unit for Different Values of Flywheel Moment. Observed Deviation is Shown for Two Different Values of Flywheel Moment

where $\eta \div n$ is the ratio of the natural frequency to the crank frequency. As discussed in Par. 14, if $\eta \div n$ is different from unity by 25 per cent, the damping term $(T_d w)$ in [7] can usually be neglected with small error; and so far as safety is concerned, can always be neglected in calculation. Moreover, it is obvious that even with $\eta \div n$ equal to 0.75 or to 1.25, the increase in deviation above θ_o is about 125 per cent, and with a ratio nearer unity, the increase becomes very great. Therefore ratios between 0.75 and 1.25 should be avoided; and if they are avoided, the effect of damping may be neglected.

25 Accepting this as a basis, the design rule for the case of the simple sine torque at once follows: *Make the flywheel large enough to limit the initial deviation θ_o to one-half of the actual maximum deviation θ_m which is expected to occur in operation. Then test out that flywheel by Equation [6] to make sure that the frequency ratio $\eta \div n$ does not fall between* 0.75 *and* 1.25. *If it falls in that zone, increase the flywheel until the ratio is* 0.8.[1] The further the ratio is from this danger zone, the less, relatively, the deviation will be. Thus the rule is safe for the particular case assumed.

26 But the same rule, with slight modification, can be applied to the general case, in which the unbalanced crank effort is not a simple sine function, but rather the irregular curve which is met with in practice. If θ_o is taken as the initial deviation produced by the latter crank effort alone, and if in addition the frequency ratio $\eta \div n$ be changed to $\eta \div n_m$, where η is, as before, the natural oscillating frequency and n_m is the frequency of any forced impulse,[2] the above rule becomes general.

27 Consider how the rule operates in the general case. The initial deviation θ_o, calculated from the engine design by methods given in Appendices A and B, is not now, as it was in the simple case, produced by a crank effort in which all of the unbalanced torque is concentrated in a single component, but instead by one in which the unbalanced torque is scattered or diffused, as it were, among many components of different frequencies, all of them being multiples of the crank revolutions.[3] The purpose here is to show that as a consequence the increase in deviation above θ_o, on account of the electrical forces, cannot possibly be as great as that on which the rule was based, namely about 2 : 1. It is evident from [9] that if $\eta \div n$ is 0.75, only that portion of θ_o which is produced by the torque of fundamental frequency n will be increased in the ratio of 2.25 : 1. The remainder will be increased in the ratio

$$\frac{1}{1-\left(\frac{0.75}{2}\right)^2} = 1.17$$

[1] This increase in flywheel lowers θ_o, and therefore a greater increase is permissible above this lower value. Hence a nearer approach to unity ratio. The flywheel could not be decreased; that would increase θ_o.

[2] Where n_m = any multiple of the crank frequency n. Also $0.5n$ in 4-cycle engines.

[3] In four-cycle engines one-half the crank revolutions must be also considered.

or

$$\frac{1}{1-\left(\frac{0.75}{3}\right)^2} = 1.07$$

depending upon whether the frequency of the torque which produces the remainder of θ_o is $2n$ or $3n$, etc. Thus for illustration, suppose half of θ_o is due to a torque of the fundamental frequency n and half due to the second harmonic. Then if the maxima of the two happen to occur at the same instant, which they usually do not, the deviation would be

$$\theta_m = 0.5\theta_o(2.25 + 1.17) = 1.71\theta_o$$

instead of $2.25\theta_o$.

28 But suppose in the above example that $\eta \div n = 1.25$. Then from [9], assuming the unfavorable condition that the maxima are added in phase,

$$\theta_m = 0.5\theta_o(1.78 + 1.64) = 1.71\theta_o$$

or, if

$$\eta \div 2n = 1.25,$$

$$\theta_m = 0.5\theta_o(0.19 + 1.78) = 0.985\theta_o, \text{ that, is, decreased.}$$

If

$$\eta \div 2n = 0.75$$

$$\theta_m = 0.5\theta_o(0.8 + 1.78) = 1.29\theta_o$$

29 This process can be carried out for $3n$, $4n$ and for any conceivable distribution of θ_o among the different harmonics. It is therefore clear that so long as η is not within 25 per cent of *any* multiple of the crank revolutions, the increase in deviation due to electrical forces cannot exceed 2.25, because this is the increase which would result if *all* of the unbalanced crank effort were concentrated in a single component. The rule is therefore safe, and at the same time does not involve a serious waste in flywheel. In any case where the flywheel required by method (*b*), Par. 23, may seem to be a hardship, the complete calculation should be made either by (*a*) or by the suggested modification of method (*a*), Par. 21, in which the maxima of the several components are added arithmetically.

30 The above examples bring out the interesting point that it is a fortunate circumstance to have the application of the above rule to place the oscillating frequency higher, instead of lower, than the frequency of the predominating harmonic,[1] since that condition gives relatively lower deviation.

[1] The predominating harmonic in the crank effort may not be predominating in the displacement θ_o. The higher the order of the harmonic, the less, relatively its effect in the displacement.

31 Regarding the limit which should be set to θ_m and θ_o, it is our best judgment that the total pulsation in current and power should never exceed 60 per cent and, if not a serious hardship, should be considerably less. Since the synchronizing power per electrical degree in the average machine in question is of the order of 5 per cent, the corresponding angular fluctuation would be 12 deg., or ± 6 deg. deviation. Giving some consideration to the fact that the actual crank effort is always made up of more than one harmonic, it may reasonably be assumed that θ_o will be increased by the ratio 1.75 instead of 2.25. This obviously fixes the limit of θ_o at a value corresponding to $\pm 3\frac{1}{2}$ electrical degrees.

32 It is now possible to state the rule in practical terms: The total flywheel moment of the unit should be such that

(a) *The angular deviation, as determined by the crank effort alone, does not exceed* $\pm 3\frac{1}{2}$ *electrical degrees*

(b) *The natural oscillating frequency is not within* 25 *per cent of*

1 *Any multiple of the crank revolutions in two-stroke cycle units*

2 *Any multiple of half of the crank revolutions in four-stroke cycle units.*

33 If the flywheel thus determined seems excessive, complete calculation including more accurately the effect of electrical forces may possibly show that the flywheel can be decreased. In such a calculation the criterion is that *the angular deviation including the effect of electrical forces shall not exceed* ± 6 *electrical degress. It should be less if at all feasible.* It is probable that a whole system supplied by units with this large deviation would be unsatisfactory, especially if other synchronous apparatus, such as rotary converters, were connected to it. It should therefore be understood that designing for such a large deviation is treading on rather uncertain ground, and that before it is done the effect of large power fluctuations on the particular system in question should be considered.

34 The practical formula, based on [6], for calculating the natural oscillating frequency of direct-coupled units, is

$$F = \frac{35{,}200}{\text{r.p.m.}} \sqrt{\frac{P_o f}{WR^2}} \quad \ldots\ldots\ldots \quad [13][1]$$

where

F = natural oscillating frequency, periods per minute

r.p.m. = crank revolutions per minute

[1] For derivation see *General Electric Review*, March 1915, p. 170.

P_o = factor representing synchronizing torque; and is the kw. synchronizing power corresponding to one electrical radian displacement

f = electrical frequency, cycles per sec.

WR^2 = flywheel moment in lb-ft.^{2}

CALCULATIONS AND TEST OF AN ACTUAL INSTALLATION

35 The unit consists of a two-cylinder, two-stage air compressor, cranks at 90 deg., driven by a 210-hp., 170-kva. 257-r.p.m. 60-cycle, unity power factor, 2300-volt synchronous motor. The

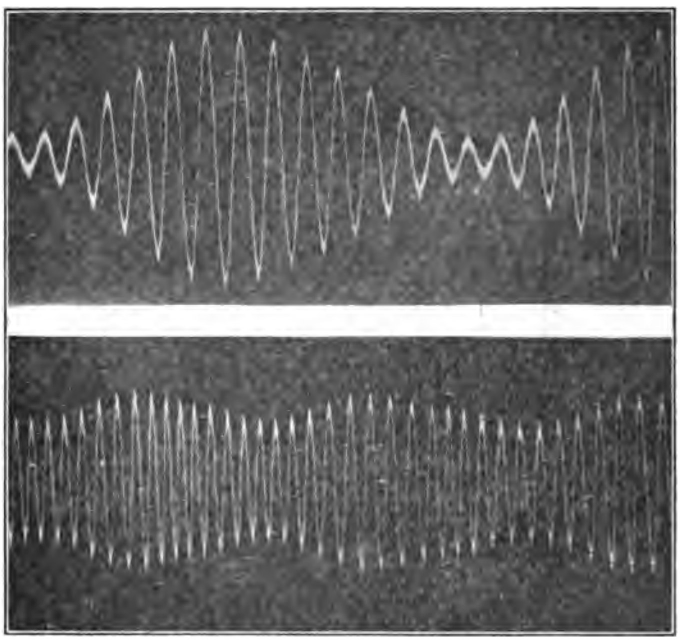


FIG. 2 OSCILLOGRAMS SHOWING CURRENT PULSATION BEFORE AND AFTER FLYWHEEL WAS ADDED. ONE REVOLUTION CORRESPONDS TO 14 ALTERNATIONS

Upper Curve (before adding flywheel): Average ammeter indication, 38.9 amp.; the peak from oscillogram = 96, corresponding to an ammeter current of 68.1 amp.

Lower Curve (after adding flywheel): Average ammeter indication, 39.7 amp.; corresponding values from oscillogram are 60 and 42.5, respectively.

motor is supplied from a large system. The test consisted principally in taking the following items simultaneously: (1) measurements of actual angular deviation; (2) oscillograph records of current pulsation; (3) indicator cards and air pressures. These were taken at full load both before and after a flywheel was added, the increase in flywheel moment being 57 per cent. Calculations given below were thus confirmed. The electrical angular deviation was decreased from about 11 deg. to 3 deg. The improvement in cur-

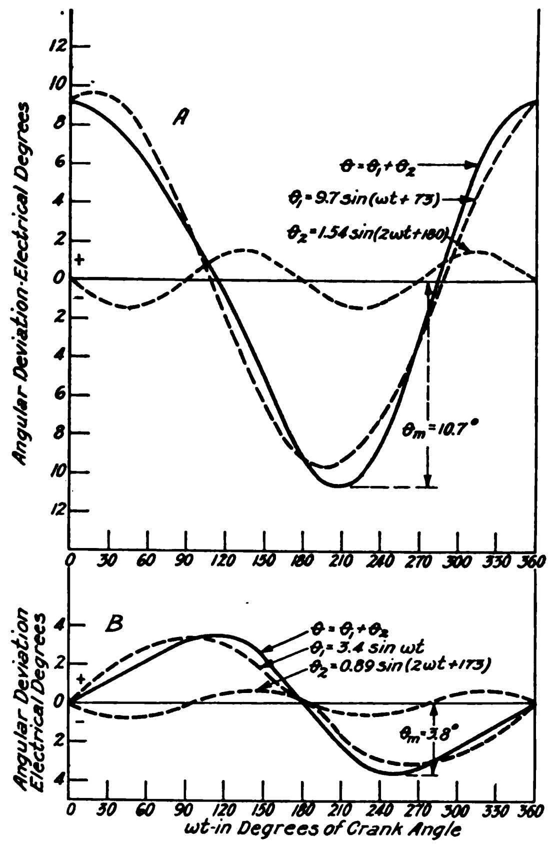


FIG. 3 CALCULATED CURVES OF ANGULAR DEVIATION, TAKING ACCOUNT OF ALL ELECTRICAL AND MECHANICAL FORCES

(A) Before flywheel was added; (B) After flywheel was added.

rent fluctuation is shown by a comparison of oscillograms in Fig. 2. The results of the complete calculation of angular deviation for all possible values of flywheel moment are shown in Fig. 1. The observed values of angular deviation are also shown. Data are as follows: (Crank-effort curve, Fig. 5 D):

Total WR^2 = 10,500 lb-ft.[2] (originally)

WR^2 = 16,500 after adding flywheel

P_o = 623 kw.; f = 60; r.p.m. = 257; poles, 28

T_d = 1800 lb-ft.[1]; w = 26.9 rad/sec.; T_s = 240,000 lb-ft.[1]

36 Calculations will now be given by the design rule, (*b*), Par. 32; and by the complete method, (*a*) Par. 18.

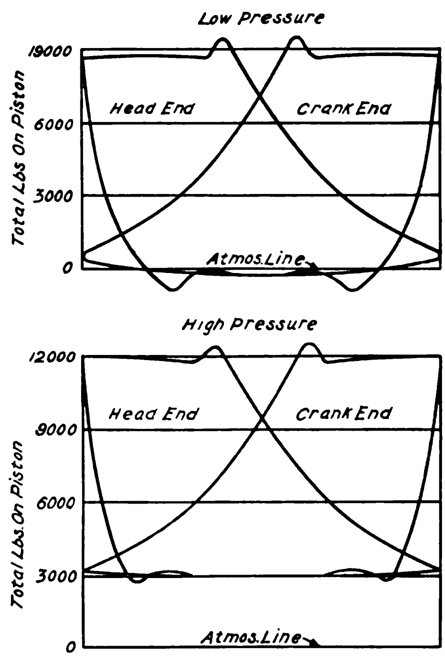


FIG. 4 INDICATOR CARDS OF THE COMPRESSOR AT FULL LOAD

37 *Calculation by the Shorter Method.* From analysis of the crank effort by the "Summation Method" given in Appendix B, θ_o = 3.0 electrical degrees. Thus far satisfactory. However, the natural frequency is

$$F = \frac{35{,}200}{257}\sqrt{\frac{623 \times 60}{10{,}500}} = 258 \text{ periods per min.}$$

that is, practically equal to the crank revolutions. It is therefore necessary to *increase* the flywheel, thus decreasing F. Increasing the WR^2 to 16,500, as was actually done, decreases F to 205, giving a frequency ratio $\eta \div n$ = 0.8, in accordance with the rule.

[1] See Appendix C.

38 *Calculation by the Complete Method.* The complete calculation requires the resolution of the crank effort into its several sine-wave components as expressed by Equation [12]. This is done by

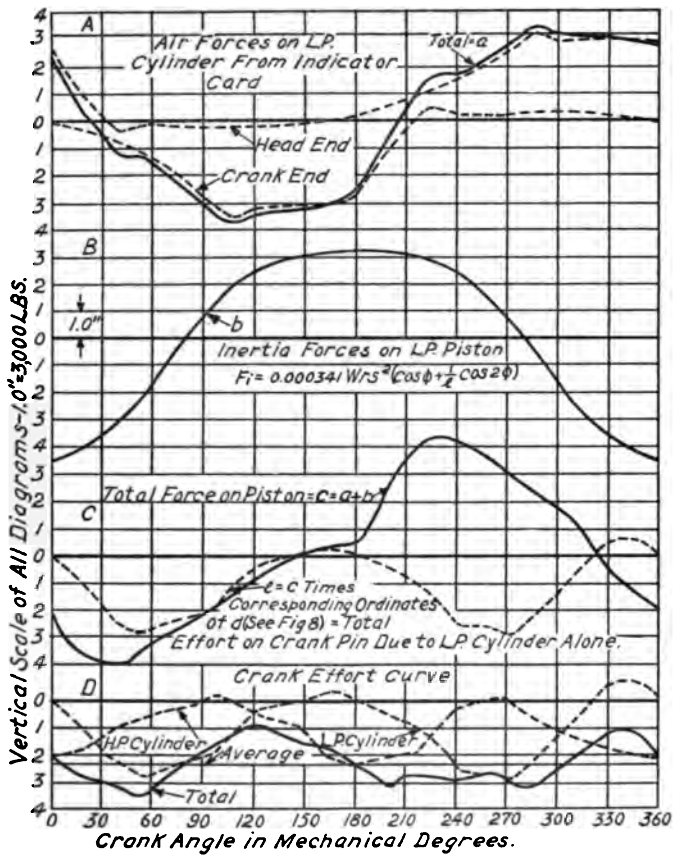


FIG. 5 DEVELOPMENT OF THE TOTAL CRANK-EFFORT FROM THE INDICATOR-CARD FORCES AND THE INERTIA FORCES OF THE RECIPROCATING MASSES. THE RADIUS OF CRANK CIRCLE IS 0.058 FT.

the method given in Appendix B. The equation of the crank-effort curve thus determined is

$$T = -542 \sin wt + 210 \cos wt + 1350 \sin 2wt - 263 \cos 2wt + \ldots$$

39 The third and higher harmonics are neglected, because the effect of a given torque in producing angular deviation is reduced by the square of the order of the harmonic. Thus the same alternating

torque will produce only $\frac{1}{9}$ of the displacement at 3 times the frequency. Consider the harmonics separately by Equation [35], taking the fundamental $m = 1$. The moment of inertia I before the flywheel was added was $10{,}500 \div 32.2 = 326$.

$$T_{mo} = \sqrt{(-542)^2 + 210^2} = 580$$

Therefore the maximum deviation produced by this component is

$$\theta_{m1} = \frac{580}{\sqrt{(1800 \times 1 \times 26.9)^2 + (240{,}000 - 326 \times 1^2 \times 26.9^2)^2}}$$

$$= 0.0121 \text{ mech. radians}$$

$$= 0.0121 \times \frac{28}{2} \times 57.3 = 9.7 \text{ elec. deg.}^1$$

The phase angle is

$$\psi_1 = \text{arc tan}\,\frac{N_1b_1 - M_1a_1}{N_1a_1 + M_1b_1} = \frac{+27.0 \times 10^6}{-8 \times 10^6} = -3.38$$

Thus,

$$\psi_1 = 73 \text{ deg.}$$

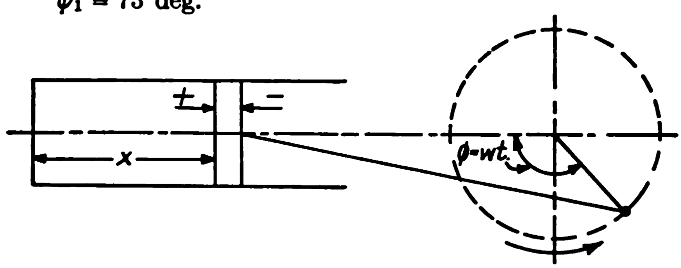


FIG. 6 DIAGRAM INDICATING THE RELATION BETWEEN THE CRANK ANGLE AND PISTON DISPLACEMENT

and

$$\theta_1 = 9.7 \sin (wt + 73) \text{ elec. deg.}$$

In like manner,

$$\theta_{m2} = 0.00192 \text{ mech. rad.} = 1.54 \text{ elec. deg.}$$
$$\psi_2 = 180 \text{ deg.}$$

hence,

$$\theta_2 = 1.54 \sin (2wt + 180)$$

40 The total angular deviation is the sum of these components Thus,

$$\theta = \theta_1 + \theta_2$$

This is plotted in Fig. 3A, giving $\theta_m = 10.7$ deg.

[1] Elec. deg. = mech. rad. $\times \frac{\text{poles}}{2} \times 57.3$.

41 In like manner the calculation is made for conditions after adding flywheel. All factors are the same except I, which is now 57 per cent greater. Thus,

$$\theta_{m1} = 3.4 \text{ elec. deg.}$$
$$\psi_1 = 0 \text{ (practically)}$$
$$\theta_{m2} = 0.89 \text{ elec. deg.}$$
$$\psi_2 = 173 \text{ deg.}$$

Hence

$$\theta_1 = 3.4 \sin wt$$
$$\theta_2 = 0.89 \sin (2wt + 173)$$

These are plotted in Fig. 3B, giving

$$\theta_m = 3.8 \text{ elec. deg.}$$

In the same manner the whole range of possible flywheel moments

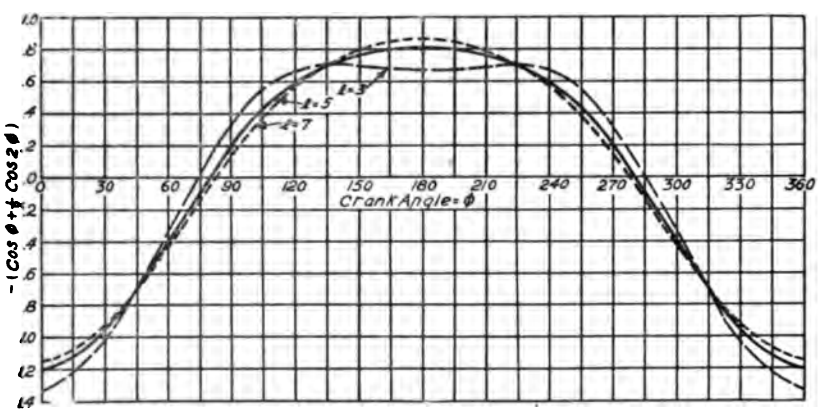


FIG. 7 CURVE FOR THE EASY DETERMINATION OF THE INERTIA FORCES

was investigated. The results are plotted in Fig. 1. The observed values are also plotted.

TESTS

The test described here is one of a number of the same character. It is particularly interesting in two respects: it gave conditions at resonance, $\eta = n$, and also afforded opportunity for observations with two different values of flywheel moment.

42 Measurements of angular deviation were taken by the aid of the stroboscope.[1] The angular fluctuation of segments painted

[1] Method was developed in 1916, and described in *General Electric Review*, October 1917.

on the rotor rim was measured by calipers. Oscillograph records of line current were taken. Fig. 2 shows the current pulsation before and after the flywheel was added. Table 1 shows the results of the tests. Indicator cards were taken to check those which had been used in the calculation of the crank effort diagram.

43 Fig. 1 shows the observed values of angular deviation for two different values of flywheel moment, as compared with calculations. Attention is called to the relatively low observed deviation, as compared with the calculated value for the larger WR^2. One possible explanation of this is that a greater per cent error is involved in the smaller observations. The probable error of observation is ±0.5 elec. deg., that is, about 5 per cent in the larger observation of 10.7 deg., 25 per cent in the lower, 2.13 deg. The corresponding

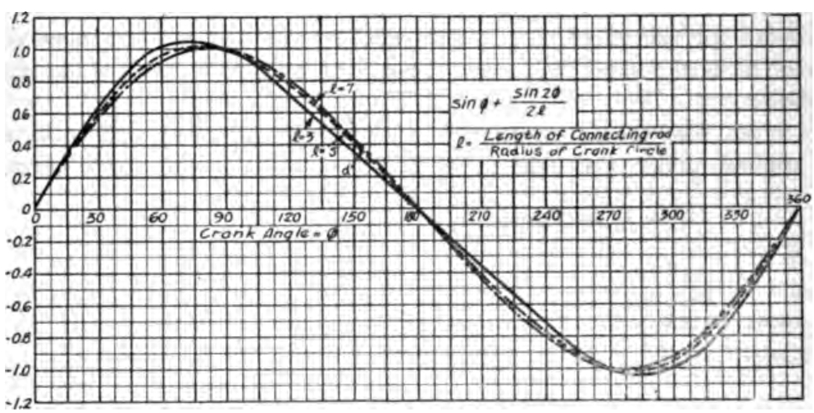


FIG. 8 CURVE FOR TRANSFORMING PISTON FORCE TO TANGENTIAL EFFORT ON THE CRANKPIN

power fluctuation per degree as given by the fluctuation of current, for the latter case, indicates that the observed angle is too low. The values kw. per elec. radian, P_o, as given by test (Table 1) for large observed angles are 591, 633, and 647 (average 624); calculated P_o = 623. However, the values determined from observations of smaller angles are 860 and 810, indicating that the angle in question was actually greater than observed. From this it would appear that it was greater by 835 ÷ 624 = 1.34; or 2.85 instead of 2.13 deg.

SUMMARY

44 The foregoing may be summarized as follows:

a Power fluctuation in synchronous machines connected to the same line is proportional to the angular fluctuation. Angular fluctuation is started by fluctuating crank forces, and once started it is modified, usually increased, by the electrical forces. The problem of designing flywheels for these units is, so far as the synchronous machine is concerned, first to decide how much power pulsation can be tolerated, and then find what flywheel moment is required to keep the angular, and therefore the power, fluctuations within the prescribed limits, when the rotor is acted upon by all of the mechanical and electrical forces.

b For practical design two needs must be filled: (*a*) a general solution, making it possible, if conditions warrant, to account in detail for the effect of all forces acting upon the rotating masses; (*b*) a short, simple and safe abridgment of (*a*) for general design work.

c The general solution (*a*) is given by Equation [35]. It is applied under Calculations and Test of an Actual Installation,

TABLE 1 TEST RESULTS AT APPROXIMATELY FULL LOAD

Angular Deviation, Elec. Deg.	Kw. Variation [1] from Average	P_e
Before adding Flywheel		
a 11.2	115.7	591
a 11.2	123.7	633
b 10.7	120.8	647
After adding Flywheel		
a 3.2	48	860
b 2.13	30	810

[1] From oscillograph record.
a Taken when two other compressor units were on the line.
b Taken when other units were shut down.

Par. 35 et seq., and illustrated in Figs. 1 and 3. Fig. 1 shows also the test values.

d The abridgment for general design work (*b*) takes the form of the following rule: The total flywheel moment of the unit should be such that

1 *The angular deviation, as determined by the crank effort alone, does not exceed* $\pm 3\frac{1}{2}$ *electrical degrees.*

2 *The natural oscillating frequency is not within 25 per cent of*

(*a*) *Any multiple of the crank revolutions in two-stroke cycle units.*

(*b*) *Any multiple of half of the crank revolutions in four-stroke cycle units.*

If the flywheel thus determined seems excessive, a complete calculation including more accurately the effect of electrical forces may possibly show that the flywheel can be decreased. In such a calculation the criterion is that *the angular deviation including the effect of electrical forces shall not exceed* ± 6 *electrical degrees. It should be less if at all feasible.*

It is probable that a whole system supplied by units with this large deviation would be unsatisfactory, especially if other synchronous apparatus, such as rotary converters, were connected to it. It should therefore be understood that designing for such a large deviation is treading on rather uncertain ground, and that before it is done the effect of large power fluctuations on the particular system in question should be considered.

The practical formula for calculating the natural oscillating frequency of direct-coupled units is given as Equation [**13**].

e The above rule involves the calculation of the angular deviation θ_o which the crank forces alone would produce. A method is given in Appendix B for calculating this from the crank-effort curve.

f A simplified method is given in Appendix A for calculating the crank-effort curve.

g Considering the effect of the flywheel upon governor operation, it is desirable to equip the governor with a suitable dashpot, or else make provision for adding one later if it proves to be necessary. This is usually done, but not always. The purpose is to prevent the governor from sustaining or augmenting an oscillation: which might otherwise happen if the period of the governor is near that of one revolution, or, in undamped machines (steam-engine drive), near the natural period of the unit; or, if the governor is too sensitive, allowing continual readjustment for momentary speed fluctuations.

45 The authors gratefully acknowledge the valuable assistance of Messrs. A. R. Stevenson, Jr., W. T. Berkshire, H. L. Reich, H. C. Lehn, F. D. Longacre, E. J. Kates, C. R. Neeson, R. G. Webber and R. Wintzer.

APPENDIX A

CALCULATION OF CRANK-EFFORT CURVE

46 The calculation of the crank-effort curve is given in any good text on the subject. It is included here partly in the interest of illustrating the entire calculation, and partly to offer one or two time-saving suggestions.

47 The calculation involves finding the total force on the piston or the crosshead as a function of piston displacement, the total force including both the gas pressure on the piston and the reaction of the reciprocating masses. The crank effort is the tangential component of this total force as transmitted to the crankpin by the connecting rod.

48 For illustration, take the two-stage compressor unit on which the foregoing tests were made. Considering the low-pressure cylinder first, the indicator cards shown in Fig. 4 are developed on the piston base in Fig. 5. Zero crank angle is taken as the beginning of the intake stroke for the head end. Those forces acting on the piston in a direction to increase x in Fig. 6 are taken as positive and those to decrease x, negative. The resultant force acting on the piston due to the indicator card forces alone is given by curve a, Fig. 5A. The reaction of the reciprocating masses, that is the inertia forces, is given by the equation [1]

$$F_i = -0.000341\, WrS^2(\cos\phi + \frac{1}{l}\cos 2\phi) \quad . \; . \; . \; . \; . \; . \; [14]$$

where

F_i = total inertia force in pounds on the piston
W = weight of reciprocating parts, pounds
r = radius of crankpin circle in feet
S = r.p.m. of crank
ϕ = crank angle = wt
l = ratio of the length of connecting rod to radius of crankpin circle

49 In Fig. 7, the two cosine functions of this equation are plotted for different values of l. Thus, to find the inertia force for any given value of crank angle it is only necessary to multiply the corresponding ordinate of the curve for the given value of l by the factor $0.000341 WrS^2$. The inertia forces thus obtained are given by curve b, Fig. 5B. Curve b added to curve a gives curve c, which is the total force acting on the piston. This force can readily be transferred to tangential force on the crankpin by the transformation factor

$$s = \sin\phi + \frac{1}{2l}\sin 2\phi \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [15]$$

This function is plotted in Fig. 8 for three different values of l.

50 Thus, to obtain the crank-effort curve for the low-pressure cylinder, multiply the ordinates of curve c, Fig. 5, by the corresponding ordinates of curve d, Fig. 8. This gives curve e, Fig. 5C. In a similar manner the crank-effort curve for the high-pressure cylinder can be obtained. The addition of these two curves

[1] For derivation see Heck's Steam Engine and Turbine, chap. 7, p. 308.

displaced relatively to each other by the angle between their cranks, in this case 90 deg., gives the total tangential effort shown by the full curve in Fig. 5D. It will be seen that this curve falls on the negative side of the zero line. This is because the derivation assumes that all forces that tend to maintain rotation are positive, and those that tend to retard rotation, negative. For an engine, therefore, the crank-effort curve would fall above the zero line; for a compressor, below.

APPENDIX B

DETERMINATION OF ANGULAR DEVIATIONS FROM CRANK EFFORT CURVE

51 This has been the most difficult and discouraging phase of the investigation. It requires two successive integrations of the crank effort curve as shown by Equation [4], and this is at best a laborious process where the crank effort is not a simple sine function; and it usually is not. The first integration gives velocity pulsation, the second gives angular deviation. There is no other way about it, so far as we can tell, except in certain special cases, and in the methods or "tools" which are here proposed, advantage has been taken of the possibility, afforded by these special and rather rare cases, to shorten the calculation.

52 The search for a method of calculating that would not be prohibitively complicated involved a study and trial of four different suggestions. All but one of these were merely different ways of integrating an irregular curve; the one was an unsuccessful attempt to apply generally the process, which as mentioned above, is applicable to certain special cases only.

53 The results are:

a The scheme of summation of ordinates, as illustrated in Fig. 13, has been chosen for general design calculations

b A much shorter method, illustrated in Fig. 11, for cases where one cycle of the crank effort curve contains only two "loops" or "lobes"

c A schedule shown in Fig. 9 for resolving the crank effort into its separate harmonic components. This is necessary for a complete solution by Equation [35]. The initial deviation θ_0 can also be determined directly from this analysis as indicated in Fig. 9.

54 The four proposed schemes of integrating will be briefly discussed in order.

55 *Planimeter Method.* Any one who has made two successive planimeter integrations of a crank effort curve taking properly small sections at a time, will agree that, as a work-a-day method, it is out of the question.

56 *Fourier Series Method.*[1] This involves expressing the crank effort curve in a mathematical equation, a Fourier's Series, representing acceleration, and then integrating the equation twice. This is a most interesting and instructive method, and is not the most complicated. It is instructive in showing the relative magnitude of the torques of different frequencies. With the aid of a schedule, such as shown in Fig. 9, the determination of the coefficients of the

[1] Described by H. C. Lehn, *General Electric Review*, March, 1915, p. 174.

ANALYSIS OF CRANK EFFORT CURVE
FOR 2 STROKE CYCLE

$y = a_0 + a_1 \sin\omega t + b_1 \cos\omega t + a_2 \sin 2\omega t + b_2 \cos 2\omega t + a_3 \sin 3\omega t + b_3 \cos 3\omega t + a_4 \sin 4\omega t + b_4 \cos 4\omega t$

y = ordinates of crank effort curve in inches.

ωt	y	a_1 ±	$\sin\omega t$	$y \sin\omega t$ +	$y \sin\omega t$ −	b_1 ±	$\cos\omega t$	$y \cos\omega t$ +	$y \cos\omega t$ −	a_2 ±	$\sin 2\omega t$	$y \sin 2\omega t$ +	$y \sin 2\omega t$ −	b_2 ±	$\cos 2\omega t$	$y \cos 2\omega t$ +	$y \cos 2\omega t$ −
0	−.23	+	0			+	1000		.23	+	0			+	1000		.23
15	.37	+	.258	.09		+	.965	.36		+	.500	.19		+	.866	.32	
30	.65	+	.500	.33		+	.866	.56		+	.866	.56		+	.500	.33	
45	1.11	+	.707	.78		+	.707	.78		+	1000	1.11		+	0		
60	.79	+	.866	.69		+	.500	.40		+	.866	.69		−	.500		.40
75	.17	+	.965	.16		+	.258	.04		+	.500	.09		−	.866		.15
90	−.20	+	1000		.20	+	0			+	0			−	1000	.20	
105	−1.05	+	.965		1.01	−	.258	.27		−	.500	.53		−	.866	.91	
120	−1.37	+	.866		1.19	−	.500	.69		−	.866	1.19		−	.500	.69	
135	−.99	+	.707		.70	−	.707	.70		−	1000	.99		−	0		
150	−.74	+	.500		.37	−	.866	.64		−	.866	.64		+	.500		.37
165	−.48	+	.258		.12	−	.965	.46		−	.500	.24		+	.866		.42
180	−.05	+	0			−	1000	.05		−	0			+	1000		.05
195	.59	−	.258		.15	−	.965		.57	+	.500	.30		+	.866	.51	
210	.45	−	.500		.23	−	.866		.39	+	.866	.39		+	.500	.23	
225	.41	−	.707		.29	−	.707		.29	+	1000	.41		+	0		
240	.61	−	.866		.53	−	.500		.31	+	.866	.53		−	.500		.31
255	.52	−	.965		.50	−	.258		.13	+	.500	.26		−	.866		.45
270	.56	−	1000		.56	−	0			+	0			−	1000		.56
285	.68	−	.965		.66	+	.258	.18		−	.500		.34	−	.866		.59
300	.29	−	.866		.25	+	.500	.15		−	.866		.25	−	.500		.15
315	−.31	−	.707	.22		+	.707		.22	−	1000	.31		−	0		
330	−.99	−	.500	.50		+	.866		.86	−	.866	.86		+	.500		.55
345	−.99	−	.258	.26		+	.965		.95	−	.500	.50		+	.866		.86
Σy =	00	Sum		3.03	6.76	Sum		5.28	3.72	Sum		9.79	.59	Sum		3.19	5.09
		Net Sum		−3.73		Net Sum		+1.56		Net Sum		+9.20		Net Sum		−1.90	
$a_0 = \frac{\Sigma y}{24}$ =	00	$a_1 = \frac{\text{net sum}}{12}$ =		−.31		$b_1 = \frac{\text{net sum}}{12}$ =		+.13		$a_2 = \frac{\text{net sum}}{12}$ =		+.77		$b_2 = \frac{\text{net sum}}{12}$ =		−.158	

ωt	a_3 ±	$\sin 3\omega t$	$y \sin 3\omega t$ +	$y \sin 3\omega t$ −	b_3 ±	$\cos 3\omega t$	$y \cos 3\omega t$ +	$y \cos 3\omega t$ −	a_4 ±	$\sin 4\omega t$	$y \sin 4\omega t$ +	$y \sin 4\omega t$ −	b_4 ±	$\cos 4\omega t$	$y \cos 4\omega t$ +	$y \cos 4\omega t$ −
0	+	0			+	1000		.23	+	0			+	1.000		.23
15	+	.707	.26		+	.707	.26		+	.866	.32		+	.500	.19	
30	+	1.000	.65		+	0			+	.866	.56		−	.500		.33
45	+	.707	.78		−	.707		.78	+	0			−	1.000		1.11
60	+	0			−	1.000		.79	−	.866		.69	−	.500		.40
75	−	.707		.12	−	.707		.12	−	.866		.15	+	.500	.08	
90	−	1.000	.20		−	0			−	0			+	1000		.20
105	−	.707	.74		+	.707		.74	+	.866		.91	+	.500		.53
120	−	0			+	1.000		1.37	+	.866		1.19	−	.500	.69	
135	+	.707		.70	+	.707		.70	+	0			−	1.000	.99	
150	+	1.000		.74	+	0			−	.866	.64		−	.500	.37	
165	+	.707		.34	−	.707	.34		−	.866	.42		+	.500		.24
180	+	0			−	1.000	.05		−	0			+	1.000		.05
195	−	.707		.42	−	.707		.42	+	.866	.51		+	.500	.30	
210	−	1.000		.45	−	0			+	.866	.39		−	.500		.23
225	−	.707		.29	+	.707	.29		+	0			−	1000		.41
240	−	0			+	1000	.61		−	.866		.53	−	.500		.31
255	+	.707	.37		+	.707	.37		−	.866		.45	+	.500	.26	
270	+	1.000	.56		+	0			−	0			+	1.000	.56	
285	+	.707	.48		−	.707		.48	+	.866	.59		+	.500	.34	
300	+	0			−	1.000		.29	+	.866	.25		−	.500		.15
315	−	.707	.22		−	.707	.22		+	0			−	1.000	.31	
330	−	1.000	.99		−	0			−	.866	.86		−	.500	.50	
345	−	.707	.70		+	.707		.70	−	.866	.86		+	.500		.50
	Sum		5.95	3.06	Sum		2.14	6.62	Sum		5.40	3.92	Sum		4.59	4.69
	Net Sum		+2.89		Net Sum		−4.48		Net Sum		+1.48		Net Sum		−.10	
	$a_3 = \frac{\text{net sum}}{12}$ =		+.24		$b_3 = \frac{\text{net sum}}{12}$ =		−.37		$a_4 = \frac{\text{net sum}}{12}$ =		+.12		$b_4 = \frac{\text{net sum}}{12}$ =		−.01	

Acceleration $= \alpha = \frac{11000Kfr}{SWR^2}[a_1 \sin\omega t + b_1 \cos\omega t + a_2 \sin 2\omega t + b_2 \cos 2\omega t + a_3 \sin 3\omega t + b_3 \cos 3\omega t + a_4 \sin 4\omega t + b_4 \cos 4\omega t]$ Electrical degrees per sec².

Velocity $= V = \frac{1058000Kfr}{S^2WR^2}[a_1 \cos\omega t + b_1 \sin\omega t - \frac{a_2}{2} \cos 2\omega t + \frac{b_2}{2} \sin 2\omega t - \frac{a_3}{3} \cos 3\omega t + \frac{b_3}{3} \sin 3\omega t - \frac{a_4}{4} \cos 4\omega t + \frac{b_4}{4} \sin 4\omega t]$ Electrical degrees per sec.

Displacement $= \theta_0 = \frac{10{,}100{,}000Kfr}{S^3WR^2}[a_1 \sin\omega t - b_1 \cos\omega t - \frac{a_2}{4} \sin 2\omega t - \frac{b_2}{4} \cos 2\omega t - \frac{a_3}{9} \sin 3\omega t - \frac{b_3}{9} \cos 3\omega t - \frac{a_4}{16} \sin 4\omega t - \frac{b_4}{16} \cos 4\omega t]$ Electrical degrees.

S = Rev. per min. K = lb. on crank pin per inch scale of crank effort curve. f = Elec. freq. of generator. $\overline{WR^2}$ = Total lb. ft.² r = Crank pin radius in feet.

FIG. 9

several harmonics, is not difficult. In fact we used it for 3 years in calculating flywheels, and it must still be used where the complete calculation as described in this paper is made. The analysis of a curve requires 2 or 3 hours. Experience is, however, that extremely few engine and compressor builders will use it on account of its apparent complexity. Hence a simpler method was sought.

57 But record should be made here of the thorough and extensive mathematical investigation which was made of a possible application of this method. The idea was to express the indicator card itself as a Fourier's Series; to transform this mathematically to tangential effort on the crank pin; to add the equation of tangential inertia forces, which is itself in the form of a Fourier's Series, thus arriving at the total crank effort equation for one cylinder in terms of the crank angle wt. If there are N cylinders, that is, N cranks, then N equations are added at the proper phase angle $(wt + a)$. Thus for 3 cranks at 120 deg., the angles would be, respectively, (wt), $(wt + 120)$, $(wt + 240)$. Moreover, since usually the same configuration of indicator card may be used in design calculations of gas engines, regardless of the size of the cylinder, it seems that for a whole line of engines it is only necessary to analyze one card. The mathematical work was carried out, and as a practical test, the resulting equations were applied to a whole line of engines of one manufacturer.

58 From this method of analysis it was possible to compile a chart giving the equations of angular displacement itself for all combinations of cranks, 2-cycle and 4-cycle. These equations are in terms of the coefficients of the indicator card equation and of reciprocating weights. Although such a method would be considered by many as too complicated and therefore does not properly belong here in detail, it is nevertheless interesting and very instructive in showing how different component forces cancel out. For instance, in a 6-crank, 2-cycle engine, all components of alternating torque cancel out, except that having a frequency 6 times the crank revolution and multiples thereof. It is probable, therefore, that the work may be published later.

59 *Sine Wave Method.* This assumes that each "loop" of the crank effort curve above or below the average line can be replaced by a half cycle loop of a sine curve of the same area. Each successive loop is thus treated mathematically and a summation made of all loops. This is almost as accurate and requires about the same time as the Fourier's Series method, if there are no more than four loops per crank effort cycle. Above four, it is more complicated. For two loops, however, the equation becomes extremely simple and it is therefore given in Fig. 11 for the calculation of these cases. Its derivation follows:

60 The crank effort curve of two loops per crank effort cycle shown by the full curve in Fig. 10A, is replaced by the crank effort curve of sine loops of equal area shown by the dotted curve. The loops A and B represent unbalanced energy, $\triangle E$, which causes acceleration and deceleration of the rotor from its mean velocity. The angular velocity above or below synchronous speed will therefore be proportional to these areas A and B. Thus, during the crank angle ϕ_a the velocity will increase (for an engine) from a minimum at m, Fig. 10B to a maximum at n by an amount proportional to the area of A[1]. During the

[1] Velocity change is proportional to square root of energy change. Since small fraction δ is involved in change from E to $E(1 + \delta)$ the velocity change is 1 to $\sqrt{1+\delta} = 1 + \frac{\delta}{b}$. Thus, velocity change = ½ energy change.

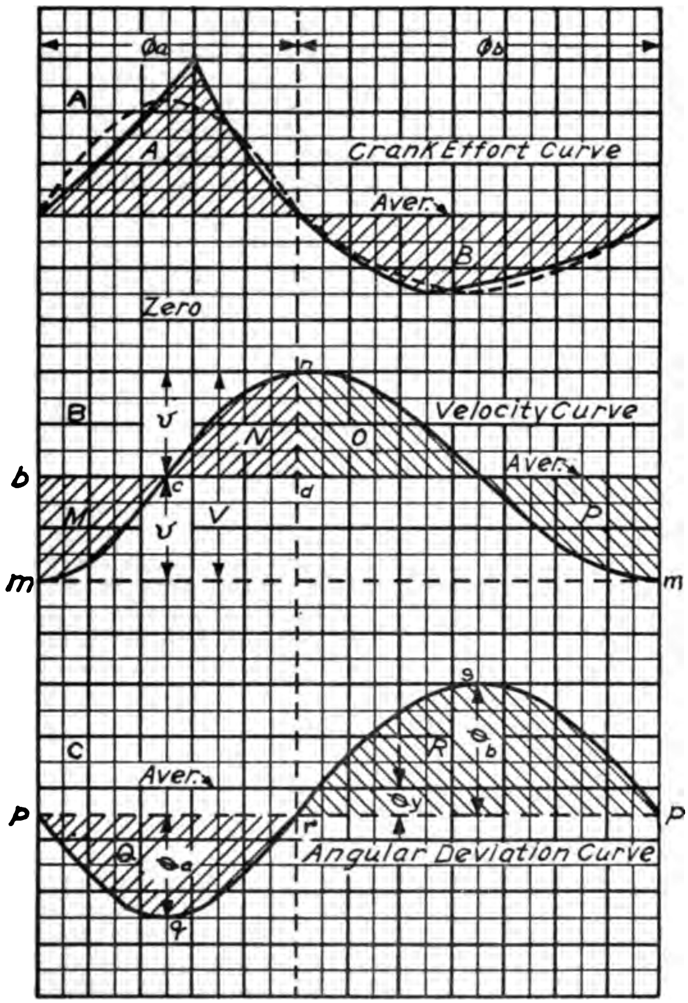


FIG. 10

crank angle ϕ_b it will decrease again to its former minimum speed at m, since the area of B is equal to that of A. Between these limits the velocity curve will, by the assumption, be half cycle portions of sine waves. The average velocity, therefore, will be half way between maximum and minimum velocities. The areas M, N, O and P being the product of velocity and time, represent displacement. Thus, the deficient velocity during bc (for an engine) will cause the rotor to fall behind by the amount equal to θ_a (Fig. 10C) while the excess velocity during the time cd will cause the rotor to catch up again by the equal amount N, so that at d the rotor will be displaced from its average load position by the same amount as at b. During the crank angle ϕ_b the velocity is first above the average, which causes the rotor to continue to gain displacement by the additional amount represented by O equal to θ_b, and then below the average causing the rotor to lose the equal amount P. At the end of ϕ_b the rotor will therefore have the same angular position relative to its average load position as at the beginning of ϕ_a. This gives the displacement curve $pqrsp$ which is also composed of sine loops. However, as yet the average position of the rotor is not known. This can be found relative to the rotor position at the beginning of ϕ_a by taking the algebraic sum of Q and R and dividing by the base pp. This gives the value of θ_y. Now during the crank angle ϕ_a the maximum deviation is,

$$\theta_o = \theta_a + \theta_y \quad \ldots\ldots\ldots\ldots \quad [16]$$

and during the crank angle ϕ_b

$$\theta_o = \theta_b - \theta_y.$$

61 First determine θ_a. As stated above, the deviation θ_a is represented by the area M. Since the curve mc is a quarter of a sine wave with maximum value v, its area and consequently the deviation θ_a will be

$$\theta_a = \frac{2}{\pi} v t_{bc} \text{ mechanical radians} \quad \ldots\ldots \quad [17]$$

where t_{bc} is the length of the base bc in seconds.

The amount of energy in loop A is

$$\triangle E = \frac{T\phi_a}{57.3}$$

where,

ϕ_a is in mechanical degrees of crank angle

T = average excess torque due to the unbalanced excess energy in A.

Therefore, $$T = \frac{57.3 \triangle E}{\phi_a}$$

The average acceleration during ϕ_a is,

$$\alpha_1 = \frac{T}{I} = \frac{57.3 \triangle E}{I\,\phi_a} \quad \ldots\ldots\ldots\ldots \quad [18]$$

where I = moment of inertia of rotating parts. Thus the increase in angular velocity during ϕ_a is

$$V = \alpha_1 t_{bd} \quad \ldots\ldots\ldots\ldots \quad [19]$$

But,

$$t_{bd} = \frac{\phi_a}{360} \times \frac{60}{S} \quad \ldots\ldots\ldots\ldots \quad [20]$$

Where S = r.p.m. of crank. Hence from [18], [19], and [20],

$$V = \frac{57.3\triangle E}{6IS}$$

and,

$$v = \frac{V}{2} = \frac{57.3\triangle E}{12IS} \quad \ldots\ldots\ldots\ldots \quad [21]$$

Also

$$t_{bc} = \frac{\frac{\phi_a}{2}}{360} \times \frac{60}{S} = \frac{\phi_a}{12S} \text{ seconds.}$$

Substituting these values in [17]

$$\theta_a = \frac{2}{\pi} \times \frac{57.3\Delta E}{12IS} \times \frac{\phi_a}{12S} \text{ mechanical radians}$$

or,

$$\theta_a = \frac{360}{2\pi} \times \frac{q}{2} \times \frac{2}{\pi} \times \frac{57.3\Delta E}{12IS} \times \frac{\phi_a}{12S} \text{ elec. deg.}$$

where

$$q = \text{no. of poles.}$$

But

$$q = \frac{120f}{S} \text{ and } I = \frac{WR^2}{g} = \frac{WR^2}{32.2}$$

where f = electrical frequency, cycles per second

WR^2 = total flywheel moment lb. ft.²

Therefore,

$$\theta_a = \frac{28{,}000 f \Delta E \phi_a}{WR^2 S^3} \text{ elec. deg.} \quad [22]$$

In a like manner,

$$\theta_b = \frac{28{,}000 f \Delta E \phi_b}{WR^2 S^3} \text{ elec. deg.} \quad [23]$$

62 Next, determine θ_y. Since the deviation curve is made up of portions of sine waves the area of the first loop Q is

$$\frac{2}{\pi} \theta_a \phi_a$$

and of the second loop R is

$$\frac{2}{\pi} \theta_b \phi_b.$$

Therefore, θ_y is

$$\theta_y = \frac{\text{net sum}}{\text{base}} = \frac{2}{\pi} \left(\frac{\theta_b \phi_b - \theta_a \phi_a}{\phi_a + \phi_b} \right) \quad [24]$$

Substituting [22], [23] and [24]

$$\theta_y = \frac{17{,}800 \Delta E f}{WR^2 S^3} (\phi_b - \phi_a) \quad [25]$$

Therefore, from [16], [22] and [25]

$$\theta_e = \frac{17{,}800 \Delta E f \phi_a}{WR^2 S^3} (\alpha + 0.6) \text{ elec. deg.} \quad [26]$$

where

$$\alpha = \phi_b / \phi_a$$

Although another maximum

$$\theta_0 = \theta_b - \theta_y$$

occurs in the second loop, it can be shown that if ϕ_a and ϕ_b are chosen so that

$$\alpha = \phi_b / \phi_a > 1$$

then

$$(\theta_a + \theta_y) > (\theta_b - \theta_y).$$

Therefore, it is unnecessary to calculate $(\theta_b - \theta_y)$. From [22], [23] and [24]

$$\theta_b = \alpha \theta_a \text{ and } \theta_y = \frac{2}{\pi} \theta_a (\alpha - 1).$$

Therefore,

$$\theta_a + \frac{2}{\pi} \theta_a (\alpha - 1) > \alpha \theta_a - \frac{2}{\pi} \theta_a (\alpha - 1).$$

Dividing then by θ_a

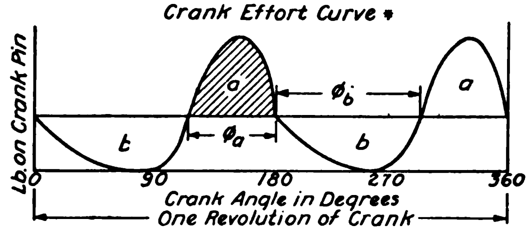
$$(0.636\alpha + 0.364) > (0.364\alpha + 0.636).$$

63 It is obvious that the lefthand member is always greater than the righthand member when $\alpha > 1$. Thus Equation [26] gives the maximum angular

Angular Deviation.
By Analysis of Crank Effort Curve

For use where each cycle of crank effort curve contains only two loops. (For more than this use Fig. 13.)

Crank Effort Curve *



(Choose ϕ_a less than ϕ_b)

$$\text{Angular Deviation} = \theta_0 = \pm \frac{17{,}800\, f\, \phi_a\, \Delta E}{S^3 WR^2} (\alpha + 0.6)$$

$$\text{Angular Deviation} = \theta_0 = \pm \frac{17{,}800\ X\ \ X\ \ X}{X}\ X \quad = \quad \text{Elec. deg.}$$

Note: δ does not appear in this equation.

f = electrical frequency in cycles per second.

S = revolutions per minute of crank.

WR^2 = total fly-wheel effect in lb.-ft.^{2}

$\alpha = \frac{\phi_b}{\phi_a}$ Note: α will always be greater than unity.

ϕ_a = (See illustration) in mechanical degrees.

ϕ_b = (See illustration) in mechanical degrees.

ΔE = ft-lb. of energy in "a" = ft-lb. of energy in "b".

$\Delta E = AKK'$

Where; A = area of "a" = area of "b" in sq. in.

K = vertical scale of crank effort curve,

i.e., 1" = K lb. tangential effort on crank pin.

K' = horizontal scale of crank effort curve,

i.e., 1" = K' ft. on crank pin circle.

* This illustration has only two cycles of crank effort per revolution. The formula, however, applies equally well for any number of cycles of effort per revolution of crank.

Rating-

FIG. 11

deviation when α is greater than unity. This equation with symbols and reference curve is given in compact form in Fig. 11.

64 *Summation Method.* This method is applicable to all crank effort curves, although, as previously stated, a two loop per cycle crank effort curve can be more quickly analyzed by the method just described.

65 All methods of calculating the angular deviation involve a double integration or summation of the crank effort curve. The first integration gives the velocity curve and the second integration the displacement curve from which the maximum deviation can be measured. The summation method performs

this double integration by twice summing up a number of ordinates of the crank effort curve to obtain a factor D which when used in the proper formula gives the maximum deviation desired. Its derivation is as follows:

66 Take one crank effort cycle, i.e., that portion of the crank effort curve which successively repeats itself, and divide its base into 24 equal parts, labeling the ordinates, separating these parts, a, b, c,........x as shown in Fig. 12A. The length of each of these ordinates from the average effort line to the crank

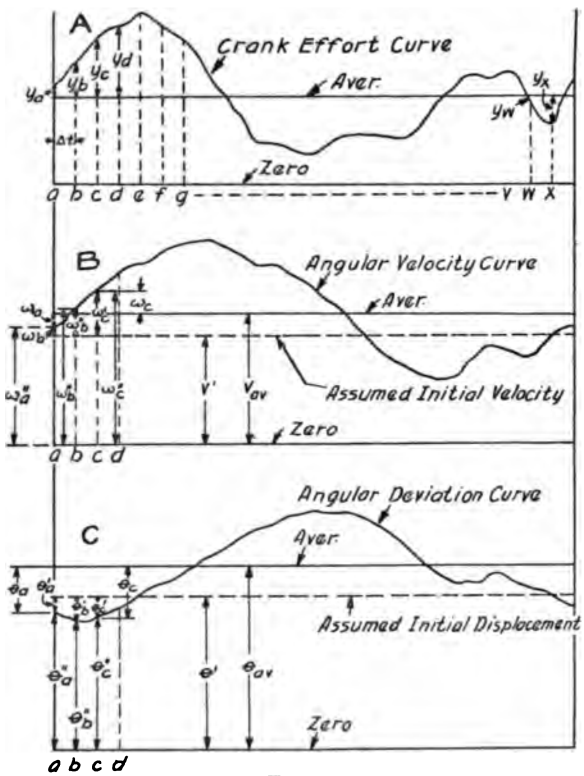


FIG. 12

effort curve, multiplied by the vertical scale and the radius of the crank pin circle, gives the unbalanced torque which at that particular instant produces acceleration or decleration of the rotor. Thus the unbalanced torque at a, b, c, etc., is

$$T_a = Kry_a$$
$$T_b = Kry_b$$
$$T_c = Kry_c, \text{ etc.,}$$

where K = vertical scale of crank effort curve in lb. per inch; r = radius of crank pin circle in feet; y_a, y_b, y_c, etc., = ordinates in inches from average effort line to crank effort curve.

67 Torque divided by moment of inertia gives angular acceleration. Thus the angular acceleration at a, b, c, etc., is

$$\alpha_a = \frac{T_a}{I} = Cy_a$$

$$\alpha_b = \frac{T_b}{I} = Cy_b$$

$$\alpha_c = \frac{T_c}{I} = Cy_c, \text{ etc.}$$

where

$$C = \frac{Kr}{I}.$$

I = moment of inertia of moving parts.

68 Angular acceleration multiplied by the time through which it acts gives the change in angular velocity during that particular time. The change in angular velocity at the points a, b, c, *etc.*, is

$$\begin{aligned} \omega'_a &= \alpha_a \Delta t = C \Delta t y_a \text{ (See Fig. 12B)} \\ \omega'_b &= \alpha_b \Delta t = C \Delta t y_b, \\ \omega'_c &= \alpha_c \Delta t = C \Delta t y_c, \text{ etc.} \end{aligned}$$

where Δt = small interval of time in seconds between adjacent ordinates y_a, y_b, y_c, etc. Thus the angular velocity, relative to the initial assumed velocity, is determined for the whole crank effort cycle. The problem is now to find the average angular velocity, or synchronous speed with reference to the actual angular velocity curve. This can be done by summing up the absolute angular velocity at the points a, b, c, etc., and dividing by 24. However, the absolute. angular velocity of the motor at any time during the crank effort cycle is not known. Assume for a reference that just prior to point a, that is, half way between x and a, the rotor has an absolute angular velocity V' radians per second. The absolute angular velocity at a (half way between a and b, to be exact) will then be the algebraic sum of the assumed velocity V' and the velocity ω'_a due to the unbalanced torque, represented by y_a. Likewise at b the absolute angular velocity will be equal to the algebraic sum of the absolute angular velocity at a and the velocity ω'_b, and so on, until at the point just beyond x the absolute angular velocity will again be the assumed velocity, V'. Thus the absolute velocity at the points a, b, c, etc., is

$$\begin{aligned} \omega''_a &= V' + \omega'_a = V' + C \Delta t y_a = V' + C \Delta t x_a \\ \omega''_b &= V' + \omega'_b = V' + C \Delta t (x_a + y_b) = V' + C \Delta t x_b \\ \omega''_c &= V' + \omega'_c = V' + C \Delta t (x_b + y_c) = V' + C \Delta t x_c, \\ &\text{etc.} \\ \omega''_x &= V' + \omega'_x = V' + C \Delta t (x_w + y_x) = V' + C \Delta t x_x = V' \end{aligned}$$

and the average angular velocity will be,

$$\begin{aligned} V_{av} &= \frac{\text{net sum}}{24} = \frac{\omega''_a + \omega''_b + \ldots + \omega''_x}{24} = V' + C \Delta t \frac{(x_a + x_b + \ldots + x_x)}{24} \\ &= V' + C \Delta t x' \end{aligned}$$

69 The amount of velocity above or below the average velocity will be the algebraic difference between the absolute angular velocity at the points a, b, c, etc., and the average velocity (see Fig. 12B). Thus the excess or deficient velocity at the points a, b, c, etc., is

$$\omega_a = \omega''_a - V_{av} = (V' + C \Delta t x_a) - (V' + C \Delta t x') = C \Delta t (x_a - x') = C \Delta t v_a$$

$$\omega_b = \omega''_b - V_{av} = (V' + C \Delta t x_b) - (V' + C \Delta t x') = C \Delta t (x_b - x') = C \Delta t v_b, \text{ etc.}$$

70 The amount of excess or deficient angular velocity multiplied by the time through which it acts gives the change in angular deviation which occurs during that time. Thus the change in angular deviation at a, b. c, etc., is

$$\theta'_a = \omega_a \triangle t = C \triangle t^2 V_a. \quad \text{(See Fig. 12C).}$$

$$\theta'_b = \omega_b \triangle t = C \triangle t^2 V_b, \text{ etc.}$$

This determines the angular deviation curve relative to an initial assumed displacement of the rotor θ' mechanical radians from the no load position. The problem is now, similar to the velocity problem above, to find the average displacement relative to the actual deviation curve. As before, this can be done by summing up the ordinates of the deviation curve and dividing by 24. Carrying out this process, the total deviation at a relative to the no load position of the rotor, will be equal to the algebraic sum of the assumed displacement θ' and the deviation θ'_a due to the excess velocity represented by ω'_a. Likewise at b the total displacement will be the algebraic sum of the total displacement at a and the deviation θ'_b. Thus the total displacement from the no load position at a, b, c, etc., is

$$\theta''_a = \theta' + \theta'_a = \theta' + C \triangle t^2 V_a = \theta' + C \triangle t^2 R_a$$

$$\theta''_b = \theta' + \theta'_b = \theta' + C \triangle t^2 (R_a + V_b) = \theta' + C \triangle t^2 + R_b$$

etc.

And the average displacement from the no load position is,

$$\theta'_{av} = \frac{\text{net sum}}{24} = \frac{\theta_a'' + \theta_b'' + \ldots + \theta_x''}{24} = \theta' + C \triangle t^2 \left(\frac{R_a + R_b + \ldots + R_x}{24}\right)$$

$$= \theta' + C \triangle t^2 R'$$

71 The amount of deviation above or below the average will be the algebraic difference between the total deviation at a, b, c, etc., and the average (see Fig. 12C). Thus the deviation from the average position at a, b, c, etc., is

$$\theta_a = \theta_a'' - \theta_{av} = (\theta' + C \triangle t^2 R_a) - (\theta' + C \triangle t^2 R') = C \triangle t^2 (R_a - R')$$

$$\theta_b = \theta_b'' - \theta_{av} = (\theta' + C \triangle t^2 R_b) - (\theta' + C \triangle t^2 R') = C \triangle t^2 (R_b - R')$$

etc.

However, only the maximum deviation is important, i.e., the maximum of the deviations θ_a, θ_b, θ_c, etc.

Let this maximum be

$$\theta_e = C \triangle t^2 (R - R)' = C \triangle t^2 D \quad \text{mechanical radians}$$

where D is the maximum value of the expressions $(R_a - R')$, $(R_b - R')$, $(R_e - R')$ etc.

Thus the maximum deviation is

$$\theta_e = \frac{Kr}{I} \triangle t^2 D \text{ mechanical radius}$$

where I = moment of inertia $= \frac{WR^2}{32.2}$

and, $\triangle t$ = time in seconds between each of the 24 ordinates, or 1/24 of the time of a crank effort cycle. If S is the crank revolutions per minute, the time of one revolution will be

$$\frac{1}{S} \text{ minutes} = \frac{60}{S} \text{ seconds,}$$

and if γ is the number of cycles of crank effort per minute divided by S, then the time of one crank effort cycle will be $\frac{60}{\gamma S}$ seconds.

Angular Deviation
By Analysis of Crank Effort Curve

For use where one cycle of crank effort curve contains more than two loops (for two loops use Fig. 11)*

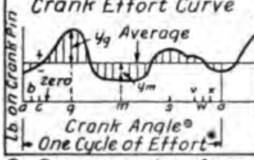
Divide one cyc. of crank effort* into 24 parts label a, b, c ··· x (See illustration)

Abscissae of crank⊙ effort curve		y Distance in inches from "Average" to Crank Effort Curve (See illustration) +	−		X = Summation of y +	−	V = X − X′ +	−		R = Summation of V +	−
a	$y_a=$	.23		$X_a=y_a=$	.23		.71		$R_a=V_a=$	.71	
b	$y_b=$		.37	$X_b=X_a+y_b=$		.14	.34		$R_b=R_a+V_b=$	1.05	
c	$y_c=$		.65	$X_c=X_b+y_c=$		.79		.31	$R_c=R_b+V_c=$	.74	
d	$y_d=$		1.11	$X_d=X_c+y_d=$		1.90		1.42	$R_d=R_c+V_d=$		.68
e	$y_e=$		.79	$X_e=X_d+y_e=$		2.69		2.21	$R_e=R_d+V_e=$		2.89
f	$y_f=$		.17	$X_f=X_e+y_f=$		2.86		2.38	$R_f=R_e+V_f=$		5.27
g	$y_g=$	.20		$X_g=X_f+y_g=$		2.66		2.18	$R_g=R_f+V_g=$		7.45
h	$y_h=$	1.05		$X_h=X_g+y_h=$		1.61		1.13	$R_h=R_g+V_h=$		8.58
i	$y_i=$	1.37		$X_i=X_h+y_i$		.24	.24		$R_i=R_h+V_i=$		8.34
j	$y_j=$	.99		$X_j=X_i+y_j$	.75		1.23		$R_j=R_i+V_j=$		7.11
k	$y_k=$	.74		$X_k=X_j+y_k=$	1.49		1.97		$R_k=R_j+V_k=$		5.14
l	$y_l=$	.48		$X_l=X_k+y_l=$	1.97		2.45		$R_l=R_k+V_l=$		2.69
m	$y_m=$	.05		$X_m=X_l+y_m=$	2.02		2.50		$R_m=R_l+V_m=$		.19
n	$y_n=$		.59	$X_n=X_m+y_n=$	1.43		1.91		$R_n=R_m+V_n=$	1.72	
o	$y_o=$		.45	$X_o=X_n+y_o=$	.98		1.46		$R_o=R_n+V_o=$	3.18	
p	$y_p=$		.41	$X_p=X_o+y_p=$	.57		1.05		$R_p=R_o+V_p=$	4.23	
q	$y_q=$		.61	$X_q=X_p+y_q=$		.04	.44		$R_q=R_p+V_q=$	4.67	
r	$y_r=$		.52	$X_r=X_q+y_r=$		.56		.08	$R_r=R_q+V_r=$	4.59	
s	$y_s=$		.56	$X_s=X_r+y_s=$		1.12		.64	$R_s=R_r+V_s=$	3.95	
t	$y_t=$		.68	$X_t=X_s+y_t=$		1.80		1.32	$R_t=R_s+V_t=$	2.63	
u	$y_u=$		.29	$X_u=X_t+y_u=$		2.09		1.61	$R_u=R_t+V_u=$	1.02	
v	$y_v=$	.31		$X_v=X_u+y_v=$		1.78		1.30	$R_v=R_u+V_v=$		.28
w	$y_w=$	.99		$X_w=X_v+y_w=$		.79		.31	$R_w=R_v+V_w=$		.59
x	$y_x=$	.99		$X_x=X_w+y_x=$	.20		.68		$R_x=R_w+V_x=$	.09	
					9.64	21.07				28.58	49.21

Net sum = −11.43   Net sum = −20.63

$X' = \frac{\text{Net sum}}{24} = -.48$ $R' = \frac{\text{Net sum}}{24} = -.86$

Crank Effort Curve



Angular Deviation

$$\theta_e = \pm \frac{692400\,Kfr}{\gamma^2 S^3\, WR^2} D$$

$$= \pm \frac{692{,}400 \times 3000 \times 60 \times .5}{1^2 \times 257^3 \times 10500} \times 7.72 = 3.0 \text{ Elec. deg.}$$

D = Max. Value of (R − R′) = 7.72

S = Rev. per min. of crank. WR^2 = total fly-wheel effect in lb-ft.^{2}

r = Radius of crank circle in ft. f = elec. freq. in cycles per sec.

K = Vertical scale of crank effort curve (1″ = K lb. tangential effort on crank pin). $\gamma = \frac{\text{Cycles of crank effort per min.}}{\text{Rev. per min. of crank}}$

* One cycle of crank effort is that portion of the crank effort curve which successively repeats itself.

⊙ The crank angle in degrees corresponding to a, b, c, d etc. is 0, 15, 30, 45 etc. respectively when $\gamma = 1.0$

For other values of γ the crank angle corresponding to a, b, c, d etc. is $0, \frac{15}{\gamma}, \frac{30}{\gamma}, \frac{45}{\gamma}$ etc. respectively

Rating -

FIG. 13

Therefore, $\Delta t = \frac{1}{24} \times \frac{60}{\gamma S}$ seconds.

The deviation in electrical degrees will then be

$$\theta_e = \frac{360}{2\pi} \times \frac{120 f}{2\,S} \times \frac{32.2\,Kr}{WR^2} \times \left(\frac{1}{24} \times \frac{60}{\gamma S}\right)^2 D$$

$$= \frac{692{,}400\,Kfr}{\gamma^2\, WR^2\, S^3} D$$

A tool to aid in the calculation of the factor D is given in Fig. 13.

APPENDIX C

ELECTROMAGNETIC FORCES

72 In Equation [1] two electromagnetic forces are considered: (*a*) Synchronizing force and (*b*) damping force.

SYNCHRONIZING FORCE

73 Synchronous machines are so constituted that when running in parallel they are constrained by an elastic electromagnetic coupling to run at the same average speed. Fluctuation may occur so long as the elastic coupling does not fail, and this does not happen until the rotor has been thrown out of its no-load stable position, by one-quarter pole pitch. Physically the elastic coupling is the magnetic field, which literally stretches like a spring when the rotor is moved either way from the zero position. Thus, the magnitude of this force is proportional to the angular displacement from zero, and its direction is opposite to the displacement. Thus, expressed as torque,

$$t_s = -T_s\theta$$

where T_s = synchronizing torque corresponding to one mechanical radian displacement, and θ is the angular displacement. Synchronizing torque is rarely given by the generator or motor builder as torque, but usually as the power in kilowatts corresponding to one *electrical* radian displacement. This is called P_o. The corresponding torque is

$$7040 \frac{P_o}{\text{r.p.m.}} \text{ lb-ft.}$$

and the synchronizing torque per mechanical radian [1] is

$$T_s = 7040 \frac{P_o}{\text{r.p.m.}} \times \frac{q}{2} \text{ lb-ft.}^2$$

where q is the number of poles. The factor P_o is given for normal load condition. Its value varies as the product of the line voltage and field excitation.

74 Of course the synchronizing torque does not increase without limit in proportion to the displacement. In fact, the torque is nearly a sine function of displacement, reaching maximum near ½ pole pitch, i.e., 90 electrical degrees. But the first part of the curve is straight, and we are interested only in that section. Thus, while the relation

$$t_s = -T_s\theta$$

does not hold rigidly, it nevertheless holds in the range to which our calculations apply.

[1] Transformation factor derived in Heck's Steam Engine and Turbine, chap. 7, p. 315.

[2] In a two-pole generator, electrical and mechanical angles are identical. But since each pair of poles constitutes two π electrical radians, one mechanical radian equals $q/2$ electrical radians, where q is the number of poles.

75 On a definite, or salient, pole machine the calculation of P_o is very involved, requiring an analysis of the distribution of magnetic flux between poles. For cylindrical rotor construction, however, P_o is simply the kilowatts corresponding to short-circuit current and normal voltage. The difference between the two is of the order of 2:1, the salient pole having the higher value.

DAMPING FORCE

76 When angular displacement occurs, there is a tangential shift in the magnetic field. This causes currents, and therefore forces in the damper winding in the pole face that are proportional to the speed of the tangential shift, that is, proportional to $\frac{d\theta}{dt}$. This torque is

$$t_d = T_d \frac{d\theta}{dt}$$

where T_d is the damping torque corresponding to a tangential shift at the rate of one mechanical radian per second. This factor is calculated from the design of the generator.

77 T_d is of the order of 1 per cent of T_s. If the damper or amortisseur winding is so designed that about this value of T_d is obtained, then all free swings, that is, those occurring at the natural oscillating frequency, will be damped out, leaving only those corresponding to the forced impulse frequencies. The unit described in this paper illustrates this. Fig. 2 shows always one pulsation per revolution.

APPENDIX D

GENERAL EQUATION OF THE MOTION OF SYNCHRONOUS MOTORS OR GENERATORS CONNECTED TO RECIPROCATING MACHINERY

78 Let I = The total moment inertia of the revolving parts and flywheel.

T_s = The synchronizing torque of the motor or generator per mechanical radian displacement from no load synchronous position.

b = Damping torque due to amort. winding and eddy currents in the pole faces per mechanical radian per second slip. $b = T_d$ in Equations [1], [3], [7].

ω = Angular velocity of synchronous speed in mechanical radians per second.

Torque due to inertia $= -I \frac{d^2\theta}{dt^2}$

Synchronizing torque $= T_s (\omega t - \theta)$

Damping torque $= b \frac{d}{dt} (\omega t - \theta) = b\omega - b \frac{d\theta}{dt}$

79 Let the torque of the reciprocating machinery be any function of time, $f(t)$. The differential equation is obtained from the condition that the sum of all these torques must be equal to zero.

$$-I\frac{d^2\theta}{dt^2}+T_s(\omega t-\theta)+b\frac{d}{dt}(\omega t-\theta)+f(t)=0 \quad [27]$$

Rearranging

$$\frac{d^2\theta}{dt^2}+\frac{b}{I}\frac{d\theta}{dt}+\frac{T_s}{I}\theta=\frac{T_s}{I}\omega t+\frac{b\omega}{I}+\frac{1}{I}f(t) \quad [28]$$

The complementary function in symbolic form is

$$D^2+\frac{b}{I}D+\frac{T_s}{I}=0 \quad [29]$$

The roots of this equation are:

$$D=\frac{-b}{2I}\pm j\sqrt{\frac{T_s}{I}-\left(\frac{b}{2I}\right)^2}=-\alpha\pm j\beta$$

The solution of the equation is therefore

$$\theta=A\epsilon^{-\alpha t}\sin(\beta t+B)+\frac{\frac{T_s}{I}\omega t+\frac{b\omega}{I}+\frac{1}{I}f(t)}{D^2+\frac{b}{I}D+\frac{T_s}{I}} \quad [30]$$

80 The last term of this equation is the particular integral. This can be separated into three parts corresponding to the three terms in the numerator. Working out the first two of these the equation becomes.

$$\theta=A\epsilon^{-\alpha t}\sin(\beta t+B)+\omega t+\frac{\frac{1}{I}f(t)}{D^2+\frac{b}{I}+\frac{T_s}{I}} \quad [31]$$

81 In order to solve the last term of the above equation, it will be assumed that $f(t)$ can be represented by the following Fourier's Series:

$$f(t)=1/2\,b_o+b_1\cos\omega t+b_2\cos 2\,\omega t\ldots\ldots$$
$$+a_1\sin\omega t+a_2\sin 2\,\omega t\ldots\ldots$$

This series will be represented as:

$$f(t)=1/2\,b_o+\sum_{m=1}^{m=\infty}b_m\cos m\omega t+a_m\sin m\omega t$$

It will be noted that $1/2\,b_o$ is the average torque.

82 The equation may then be written:

$$\theta=A\epsilon^{-\alpha t}\sin(\beta t+B)+\omega t+\frac{1}{I}\left\{\frac{1/2\,b_o+\sum_{m=1}^{m=\infty}b_m\cos m\omega t+a_m\sin m\omega t}{D^2+\frac{b}{I}D+\frac{T_s}{I}}\right\} \quad [32]$$

83 The last term of this can be separated into three parts corresponding the three terms in the numerator.

84 These three parts become respectively

$$\frac{+\frac{1}{I}1/2\,b_o}{D^2+\frac{b}{I}D+\frac{T_s}{I}}=1/2\frac{b_o}{T_s}$$

$$\frac{\frac{1}{I} b_m \cos m\omega t}{D^2 + \frac{b}{I} D + \frac{T_s}{I}} = \frac{\frac{b\, b_m\, m\omega}{I^2} \sin m\omega t + \left(\frac{T_s}{I} - m^2 \omega^2\right)\frac{b_m}{I} \cos m\omega t}{\left(\frac{bm\omega}{I}\right)^2 + \left(\frac{T_s}{I} - m^2 \omega^2\right)}$$

$$\frac{\frac{1}{I} a_m \sin m\omega t}{D^2 + \frac{b}{I} D + \frac{T_s}{I}} = \frac{\left[\frac{b a_m m\omega}{I^2} \cos m\omega t - \left(\frac{T_s}{I} - m^2 \omega^2\right) \frac{a_m}{I} \sin m\omega t\right]}{\left(\frac{bm\omega}{I}\right)^2 + \left(\frac{T_s}{I} - m^2 \omega^2\right)^2}$$

85 Substituting the above in Equation [32] and combining and rearranging terms:

$$\theta = A\epsilon^{-\alpha t} \sin(\beta t + B) + \omega t + 1/2\, \frac{b_o}{T_s} + \sum_{m=1}^{m=\infty} \frac{\sqrt{M^2 + N^2} \sin(m\omega t + \psi)}{\left(\frac{bm\omega}{I}\right)^2 + \left(\frac{T_s}{I} - m^2\omega^2\right)^2} \quad \ldots \quad [33]$$

where $\tan \psi = \frac{N}{M}$

and $M = \frac{b b_m m\omega}{I^2} + \left(\frac{T_s}{I} - m^2 \omega^2\right) \frac{a_m}{I}$

$$N = \left(\frac{T_s}{I} - m^2 \omega^2\right) \frac{b_m}{I} - \frac{b a_m m\omega}{I^2}$$

86 This can be still further simplified because

$$\sqrt{M^2 + N^2} = \sqrt{\frac{a^2_m + b^2_m}{I^2} \left\{ \left(\frac{bm\omega}{I}\right)^2 + \left(\frac{T_s}{I} - m^2 \omega^2\right)^2 \right\}}$$

Substituting and dividing:

$$\theta = A\epsilon^{-\alpha t} \sin(\beta t + B) + \omega t + \frac{1}{2} \frac{b_o}{T_s} + \sum_{m=1}^{m=\infty} \frac{\frac{1}{I} \sqrt{a_m^2 + b_m^2} \sin(m\omega t + \psi)}{\sqrt{\frac{b^2 m^2 \omega^2}{I^2} + \left(\frac{T_s}{I} - m^2 \omega^2\right)^2}} \quad \ldots \quad [34]$$

If $M_m = b\, m\, \omega$

$$N_m = T_s - m^2 \omega^2 I$$

then

$$\theta = A\epsilon^{-\alpha t} \sin(\beta t + B) + \omega t + \frac{1}{2} \frac{b_o}{T_s} + \sum_{m=1}^{m=\infty} \frac{\sqrt{a_m^2 + b_m^2}}{\sqrt{M_m^2 + N_m^2}} \sin\left[m\omega t + \tan^{-1} \left(\frac{N_m b_m - M_m a_m}{N_m a_m + M_m b_m}\right)\right] \quad \ldots \quad [35]$$

The terms in this equation are respectively, the transient displacement, accumulated displacement, due to angle of rotation, displacement from no load position due to average load torque, and the variable displacement due to the unbalanced torque. Only the latter is considered in this paper.

A and B are constants of integration.

$$\alpha = \frac{b}{2I} \qquad \beta = \sqrt{\frac{T_s}{I} - \left(\frac{b}{2I}\right)^2}$$

ω = constant angular velocity of synchronous speed in mechanical radians per second

$\frac{1}{2} b_o$ = average torque of reciprocating machine

a_m = coefficient of sin $m\omega t$ in analysis of torque effort

b_m = coefficient of cos $m\omega t$ in analysis of torque effort

M_m = $bm\omega$ $\quad N_m = T_s - m^2\omega^2 I$

I = moment of inertia $= \frac{WR^2}{g}$

T_s = the synchronizing torque per mechanical radian displacement from synchronous position

b = damping torque per mechanical radian per second slip = T_d in Equations [1], [3] and [7].

The above derivation is due to Mr. A. R. Stevenson, Jr.

DISCUSSION

H. C. LEHN. The writer has used the formulæ developed in this paper in calculating flywheels for a considerable length of time and invariably the results have been successful. These calculations have been applied to a comprehensive range of types of engines, from large double-acting gas engines to small oil engines. The increase in permissible displacement under certain conditions where the natural period is far removed from force period as described in the paper has also been applied with equal success and units with flywheels so calculated have been in satisfactory service for a number of years.

In another case an existing plant was not paralleling satisfactorily and application of the formulæ showed that conditions could be improved by increasing the excitation which in turn would lower the natural period. This was tried and the operation improved in accordance with the theory.

Where calculations are required for a large number of different sizes of engines of varying types the possible increased complication involved in an analysis by the Fourier method appears to be justified and in such an analysis the method by grouping is desirable as requiring the minimum length of time and affording a ready check on the results. In particular, by using a group schedule the equation for a crank effort diagram giving sines to the eleventh and cosines to the twelfth partial can be obtained in about three-quarters of an hour. The partials above the fourth are rarely necessary for the calculation, but having them the results can be checked at a few points by simple addition of the coefficients. The second integration then so simplifies the equation, that the maximum point can be determined with a very slight amount of numerical work.

The analysis of a very few indicator cards by the Fourier method will suffice as the basis of the calculations of a large number of types and sizes, as by adding the inertia equation and combining in the proper manner, the equation for any number of cylinders, single- or double-acting, can be readily obtained from the single indicator diagram equation. The variations due to varying mean effective pressures in the different cylinders of multi-cylinder engines can also be arrived at by inspection of original equation.

E. J. Kates. The writer wishes to congratulate the authors of this paper upon their masterful presentation of a subject which has long been an enigma to many. Those who have been engaged in engine manufacture know that the determination of flywheel inertia for driving alternating-current generators in synchronism has been largely a matter of empiric design based on previously successful units. Engine designers were divided into two classes, those who were familiar with the general theory but considered it hopeless to work it out in a practical problem, and those who were not even acquainted with the theory. Among the latter may be mentioned the adherents of the "coefficient of speed fluctuation" idea, who guided themselves by the variation in *velocity*, and not the *displacement* from mean position, thus following the example of Guldner and other similar works. As a matter of fact the graphical integration of the tangential effort diagram to secure even a velocity curve was often too laborious a task and they resorted to crude empiric formulæ.

The authors of this paper have clearly shown that it is the displacement of the rotor from mean position which determines the amount of current fluctuation, which is the governing factor in synchronous operation. They have clearly expounded the general theory so that all may grasp it and more important still, they have developed practical methods for applying the theory to routine design problems.

Appendix B is a valuable exposition of various methods for determining the angular deviation from the crank effort curve and contains simplified schedules and formulæ of great usefulness to those who have to work out problems of this sort.

The method proposed by the authors for the routine solution of flywheel moment consists in

1 Drawing the tangential effort diagram for each different cylinder size, the work being simplified by using the authors' graphic

charts for inertia force and for transferring piston force to the crank-pin,

2 Calculating the maximum displacement due to engine forces only by the so-called "summation method" which is a great simplification over the method of double graphical integration of the tangential effort curve hitherto used, and

3 If the maximum displacement as determined by the crank effort alone does not exceed 3½ electrical degrees and if the natural oscillating frequency is not within 25 per cent of the frequency of the engine impulses, the flywheel moment is satisfactory.

The authors have fixed upon this limit of 3½ electrical degrees on the assumption that the effect of the electrical forces is to multiply by 1¾ the displacement due to engine forces only; in other words, if the displacement due to engine forces only is 3½ degrees, it will increase to 6 degrees under the combined influence of the electrical as well as the engine forces, and it is this 6-degree total displacement which is the limit for keeping the current fluctuation within satisfactory bounds.

The writer wishes to point out, however, that this ratio of 1¾ is based on the assumption that the natural oscillating frequency is almost within 25 per cent of the frequency of the engine impulses and if the natural frequency is, say, only 50 per cent of that of the engine impulses this ratio becomes much less, in the neighborhood of 1¼. In such a case, the displacement, based on the crank effort alone can be almost 5 degrees without exceeding the limiting total displacement due to all forces of 6 degrees.

This is illustrated in Fig. 1 of the paper at the right-hand extremity of the curves where for a flywheel moment of 20,000 the ratio of the natural oscillating frequency to the compressor impulses is approximately 195/257 or 0.75. It will be noted that for these conditions the ordinate of the upper curve corresponding to total displacement is hardly 1.5 times the ordinate of the lower dotted curve corresponding to the displacement due to crank effort only. If these curves were extended to the right so that the ratio between the natural oscillating frequency and the frequency of the compressor impulses approached 0.5, it would be found that the curve of total displacement would approach still closer to the curve of displacement due to crank force only.

In certain heavy, slow-speed oil engines these conditions exist by nature and a flywheel heavy enough to reduce the cyclic displacement due to engine forces to about 4½ electrical degrees also reduces the natural oscillating frequency to such an extent that the ratio of $\eta \div n$ approaches 0.5. In such cases it would therefore seem safe to permit a displacement based on engine forces only, of more than $3\frac{1}{2}$ degrees and the proper figure for any particular type of engine can quite easily be determined by the methods of calculation presented by the authors of this paper.

An example of such an engine is a three-cylinder, single-acting, four-cycle engine rated 450 hp. at 164 r.p.m. for which the total WR^2 of the flywheel and generator is 840,500 lb. ft.[2] The generator is a 60-cycle machine. For this unit θ_0 (that is displacement due to engine forces only) is calculated to be 4.2 electrical degrees. However, the natural oscillating frequency is 45 periods per minute and as the predominating critical frequency is that of complete cycles of crank effort or 246 periods per minute the ratio of $\eta \div n$ is only 0.18. Three engines of this type have been running in synchronism with perfect satisfaction for several years in spite of the fact that θ_0 is 4.2 degrees instead of $3\frac{1}{2}$. This is no doubt to be accounted for by the fact that because of the small ratio of $\eta \div n$, the ratio between the total displacement and that due to engine forces only is less than 1.75 so that in this case the total displacement is less than 6 electrical degrees.

As a check upon the accuracy of the summation method presented by the authors, the writer has calculated the displacement due to engine forces only, for the compressor unit mentioned in the paper for the case of total flywheel WR^2 equal to 16,500 and has worked this out both by the summation method and by the general Formula [35 b] based upon the Fourier series method but setting $T_d = 0$ and $T_s = 0$ or in other words neglecting all electrical forces.

In this way θ_0 by summation method equals 1.91 whereas θ_0 by the exact Fourier series method equals 1.80. The error in the summation method is only 6 per cent and this is more than justified by the time saved through the simpler calculation.

In conclusion the writer would like to ask a question from a mechanical engineering standpoint which may perhaps seem foolish to electrical experts. Referring to Par. 31, the authors state "Regarding the limit which should be set to θ_m and θ_0, it is our best judgment that the total pulsation in current and power should never exceed 60 per cent and, if not a serious hardship, should be considerably less. Since the synchronizing power per electrical degree in the average machine in question is of the order of 5 per cent, the corresponding angular fluctuation would be 12 deg., or ± 6 deg. deviation."

Can the synchronizing power per electrical degree be reduced by a change in generator design to much less than 5 per cent so that the total allowable current pulsation of 60 per cent would permit an angular fluctuation of more than 12 electrical degrees?

GARDNER T. VOORHEES asked how the subject under discussion affected the design of flywheels for compressors. Did it presuppose a change in existing flywheels and compressors driven by synchronous motors?

CHAS. R. NEESON cited the case of compressor driven by belt with a total deviation of 26 electrical degrees resulting in fluctuations which disturbed a space of large area in the city of New York. The motor was of 150 hp. Another case was that of an ice machine driven by a 350-hp. synchronous motor with a deviation of 8.5 electrical degrees. There was no complaint of this from the power company, and the machine operated satisfactorily for several years. Then suddenly, he said, a transformer burned out and later a set of coils in the motor. It was thought that these accidents were caused by the surging current due to the light flywheel, as the coils could be felt to vibrate while the motor was in operation.

A. A. ADLER spoke of the design of keys for flywheels and of the importance of computing the period of the shaft.

F. L. FAIRBANKS stated that this paper should be of great value in approximating the flywheel necessary with a given motor and compressor.

It has to do, however, only with the application of a remedy for unsatisfactory electrical conditions and it is easily possible that the remedy may be more serious than the disease. It is a foregone conclusion that any material increase in flywheel effect must be given due consideration as to its effect upon other correlated details of the mechanical design.

It is a very simple matter to add weight to the rim of a flywheel, but unless it is done with due care the mechanical conditions may be upset without equivalent results from the electrical equipment.

Any unnecessary weight in flywheel is bound to increase the effort required of the motor in starting, accelerating and "pulling in." It is also a liability instead of an asset in sudden impact loads causing a pull out of the motor in advance of the circuit breaker, to which the ammonia compressor is always liable in operation.

The initial prevention should be in the design of the compressor for given conditions, as it is quite possible to so flatten the crank effort diagram as to eliminate the necessity of an unduly heavy flywheel and in some cases eliminate the flywheel entirely.

The three-cylinder, single-acting, compressor can easily be the worst offender in this respect and the two cylinder, double-acting compressor the least troublesome, especially if the possible advantage of inertia effects of the reciprocating parts are utilized.

It is not only possible but logical under these conditions to get ample inertia in the motor rotor where it should properly be, and do away with all considerations made necessary by the flywheel as an added member.

He stated that he knows of large synchronous motors driving direct-connected compressors with no flywheel and no appreciable electrical disturbance, and further, that he has corrected very severe electrical disturbances and line surge by applying the remedy to the electrical conditions only.

THE AUTHORS. For the most part, the discussion has been gratifying: even that involving criticism has been encouraging, indicating as it has, an awakened interest in the flywheel problem.

The discussion by H. C. Lehn, E. J. Kates and C. R. Neeson shows clearly that the application of the necessary mathematics connected with flywheel calculations is a feasible and practical matter, and not a fruitless and difficult process, which is based on electrical considerations only and which can be applied only by electrical theorists. Time has already confirmed the usefulness of the method to both mechanical and electrical engineers. What is now needed is that all designers of machinery which is coupled to synchronous motors and generators should become acquainted with the problem, at least with the method of design calculation.

We agree with H. C. Lehn that in analyzing crank effort curves by Fourier series method, time will be saved by using a "group schedule," that is, grouping in sums and differences, those ordinates which in the analysis are to be multiplied by the same value of sine or cosine. This is particularly true if the analysis is made for a large number of harmonies.

E. J. Kates' interpretation of the paper from the mechanical designer's viewpoint is very instructive. Regarding the possibility of reducing the magnitude of the current pulsation by a change in the design of the generator, this is possible to a limited extent. If carried too far, the synchronizing force becomes too weak to hold the machine in synchronism, and certain other desirable features are impaired. Therefore, as usual, the design is the best balance between these conflicting factors. That is, the answer to the question is that the synchronizing force is made as low as safety permits, and the flywheel must be adjusted to that value.

F. L. Fairbanks is alarmed about putting any flywheel at all on the shaft with a synchronous motor; while he acknowledges that some flywheel may be desirable from an operating standpoint, he nevertheless points out the danger from "reaction trouble," and warns against the added flywheel, least some formidable characteristic of the synchronous motor should break the flywheel arms; he feels, moreover, that from the standpoint of ease in starting, the flywheel should be of minimum weight.

Anyone will agree that if starting torque were the only consideration, the less flywheel, the better. There are, unfortunately, other more serious factors to deal with. Hence the flywheel is designed with respect to these factors and the motor is designed to start whatever flywheel is thus required; and if this is not possible, some other type of motor is used.

The most direct refutation of Mr. Fairbanks' contention that it is dangerous to put a flywheel on the same shaft with a synchronous motor is the fact that hundreds of such installations are at this time in successful operation.

The fact that in two similar installations — and to the author's knowledge, there have been only two — failure of mechanical parts, including flywheel arms, has occurred, could not be accepted as reason why a long established and, with these exceptions, successful practice should be discontinued.

A. A. Adler mentions the importance of computing the natural frequency of the system made up of the shaft and the two rotating masses. This is usually very high compared with any appreciable harmonic in the crank effort. However there have been rare instances of trouble from resonance between the shaft frequency and a significant harmonic in the crank effort; hence the point should be checked up in any proposed unit to avoid multiplication of stress from this cause. The natural frequency of two rotating masses A

and B both mounted on a steel shaft of diameter d inches, and separated by a distance l inches is

$$F = 410 \sqrt{\sigma \frac{I_A + I_B}{I_A I_B}} \text{ oscillations per minute}$$

where I_A and I_B are respectively the moments of inertia of A and B.

$$\sigma = 1710 \frac{d^4}{l}$$

No. 1763

SIDE-CUTTING OF THREAD-MILLING HOBS

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Associate-Member of the Society

It has long been known that, due to the helix of a thread, the side-cutting action of a hob distorts the form of the thread on the work. In other words, the form of the tooth on the hob is not reproduced on the threaded part. The present paper is the result of a mathematical investigation of this subject and points out the corrections in the form of thread-milling hobs which can be readily made and also produce threads sufficiently correct as to form for all practical purposes.

The profile of the thread cut with a hob is a combination of two distinct curves. First, a small fillet is formed at the root of the thread which is the path of the outside corner of the hob. No correction in the form of the hob is possible to correct this point. Second, the larger part of the flank of the thread consists of a slightly curved profile which is formed by the overlapping paths of the infinite number of cutting points which form the cutting edge of the hob. Mathematically, a curved correction can be applied to the form of the hob which will correct this profile entirely. Practically, a straight-line correction can be applied which is almost exact, as the amount of the actual curvature on the flanks of the thread is seldom greater than one-tenth of a thousandth part of one inch. The greater the angle of helix of the thread, the greater the amount of correction necessary.

One very interesting fact is that the diameter of the hob has no effect on the form of the main part of the profile. The actual amount of side-cutting is more and the height of the fillet at the root of the thread is greater, as the diameter of the hob is increased, but the rest of the profile is unchanged.

This paper deals with both externally and internally threaded parts. The general conditions of side-cutting are identical in both cases. On a screw, however, the flanks of the hobbed thread will be convex, while in a nut they will be concave. Furthermore the height of the fillet at the root of the thread and the actual amount of side-cutting are relatively greater in a nut than on a screw.

When a thread is chased in a lathe and the cutting tool has proper clearance and is set so that the plane of the cutting edges contains the axis of the thread, the exact form of the tool will be duplicated on the work. Assuming that the thread is completed, if the tool in its cutting position is brought into contact with the flanks of the thread, it will have a line bearing only. If sufficient

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Presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

clearance can be provided on the tool, this holds true regardless of the pitch of the thread, the angle of the flanks, or its diameter.

2 When a thread is hobbed, however, the axis of the hob being parallel to the axis of the thread, the path of any one cutting point is a circle, and this circle will interfere with the helix of the thread to an amount depending upon the pitch of the thread, the angle of the flanks, and the diameters of both hob and thread.

3 It is assumed in this discussion that the cutting teeth of the hob are backed off sufficiently to prevent any dragging of the relieved portion of the tool on the work. The interference between the cutting edge of the hob and the helix of the thread therefore results in the removal of additional metal, thus distorting the form of the thread. The amount of this distortion varies as the values of any of the following factors change: the pitch of the thread, the form of the thread, the diameter of the work, and the diameter of the hob.

4 It will be shown that correction for some of this distortion inevitable with this method of manufacture is impossible. The amount of this distortion, however, can be reduced in many cases by the proper relation between the diameters of work and hob. It will also be shown that most of the distortion can be corrected by a suitable alteration in the form of the cutting tool.

I. THE HOBBING OF SCREWS

5 In Fig. 1, which shows a diagram of a hob and screws, let

R = radius of any cutting point on the hob
r = radius of deepest point on work touched by R
N = number of threads per inch
A = angle of rotation of hob
B = angle of point of contact of R at angle A
C = $\frac{1}{2}$ included angle of thread
r' = radius of point of contact of R on work.

6 Formulæ will first be derived to show the interference between the path of any cutting point on the hob and the flanks of the thread and for purposes of plotting the value of y will be taken as the radial distance of the cutting point R from the deepest point on the work touched by R. In other words,

$$y = r' - r \qquad [1]$$

The value of x will be taken as the longitudinal (or axial) distance of the cutting point R from the flank of the theoretical thread. A plus value of x will indicate a clearance, while a minus value of x will indicate an interference or side-cutting.

7 In order to determine the value of y the triangle shown in heavy lines in Fig. 1 must be solved. The known factors will be taken as r, R, and B. We first have

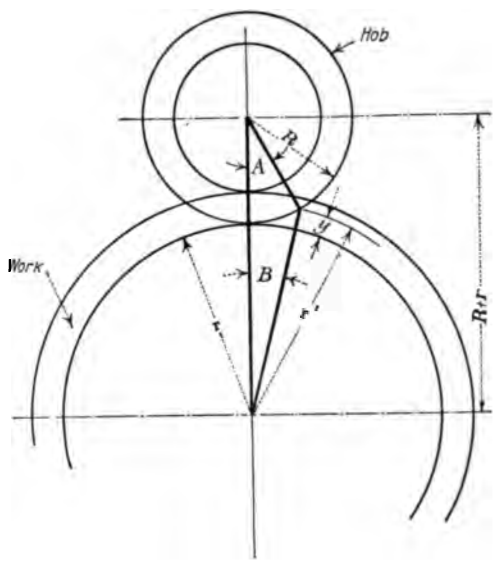


FIG. 1 DIAGRAM OF HOB AND EXTERNAL SCREW

$$\frac{(R+r)\sin B}{R} = \sin 180° - (A+B)$$

$$\sin 180° - (A+B) = \sin (A+B)$$

$$\frac{(R+r)\sin B}{R} = \sin (A+B) \quad \ldots\ldots\ldots [2]$$

From this equation we determine the value of A. Solving the triangle for r', we have

$$r' = \frac{R \sin A}{\sin B} \quad \ldots\ldots\ldots\ldots [3]$$

and when the value of r' is determined, the value of y is established from Equation [1].

8 As the hob revolves away from the common center line of the hob and work, the cutting points on one side of the tooth of the hob,

due to the helix of the thread, will have a clearance with the flank of the thread, while the cutting points on the other side of the tooth of the hob will develop an interference. The particular side involved depends upon the direction of the helix of the thread, whether left-handed or right-handed. But the side of the hob which clears the helix as the cutting point revolves away from the common center line will interfere as the cutting points approach to the common center line, and the nature and extent of this interference will be symmetrical and equal on both flanks of the thread as long as the form of the thread is symmetrical, such as in V-threads, U. S. form threads, Whitworth threads, Acme threads, etc. For buttress-formed threads, the interference or side-cutting on each flank must be determined separately. As this last form of thread is seldom used only symmetrical threads will be considered.

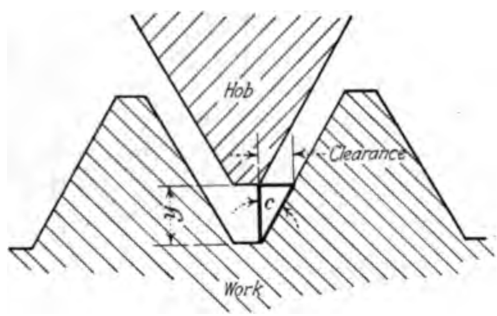


Fig. 2 Diagram Showing Clearance between Hob and Work

9 The amount of interference depends upon the value of B and the pitch of the thread, or number of threads per inch, N. Thus

$$\text{Interference Due to Helix} = \frac{B}{360N} \quad . \; . \; . \; . \; . \; . \quad [4]$$

If the included angle of the flanks of the thread is greater than zero, — which is the case for all but square threads, — as the cutting point of the side of the cutting tooth departs from the common center line of the hob and work, a clearance develops between the cutting point on the hob and the flanks of the thread. The amount of this clearance depends upon the value of y and the included angle of the thread, and referring to Fig. 2, it will be seen that its value, ignoring for the present the helix of the thread, may be expressed as:

$$\text{Clearance} = y \tan C \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [5]$$

The value of x is therefore found by subtracting the amount of interference given by Equation [4] from the amount of clearance given by Equation [5], or

$$x = y \tan C - \frac{B}{360N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

10 As an example of the use of the above formulæ, we will assume that we wish to hob a thread which has a radius r of 0.5625 in. at the minor diameter with a hob whose radius R is 2.250 in. at its outside diameter with 4 threads per inch. A thread with a relatively large helix angle is taken as the first example in order to show the nature of the resulting side-cutting of a hob. The thread has an included angle of 60 deg., making $C = 30$ deg., and $N = 4$. The

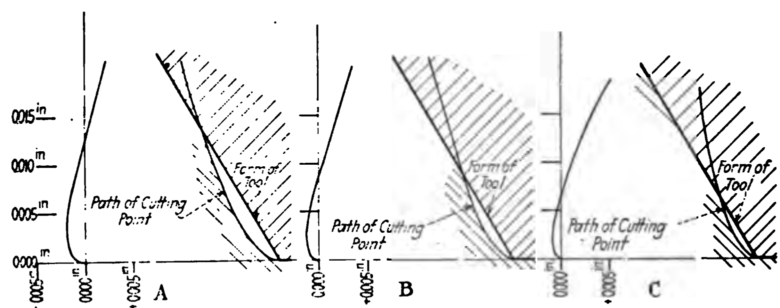


FIG. 3 CURVES SHOWING PATHS OF CUTTING POINT

value of B is taken consecutively from 0 deg. to 12 deg., which gives the following values for x and y:

For B = 0°	2°	3°	4°	5°	6°	7°	8°	10°	12°
x = 0	−0.00114	−0.00152	−0.00176	−0.00193	−0.00194	−0.00183	−0.00157	−0.00064	−0 00084
y = 0	0.00043	0.00097	0.00176	0.00267	0.00386	0.00524	0.00691	0.01092	0.01589

The above values are plotted in Fig. 3-A at the left, and the actual path of the cutting point is shown at the right. These curves show the general form of the side-cutting of any point on the cutting face of a thread hob.

11 This cutting face of the hob is made up of an infinite number of points. As the positions of these points vary, the ratio between R and r varies, as also does the helix angle of the thread, and therefore each cutting point travels in a different form of path. Thus, in order to determine the resulting form of a thread cut with a hob it will be necessary to plot the paths of a few other points. A point 0.20 in. higher on the flank of the thread will therefore next be taken.

This gives $R = 2.050$, and $r = 0.7625$, as before, $C = 30$ deg. and $N = 4$, and the values for x and y with these factors are as follows:

For $B = 0°$	1°	2°	3°	4°	5°	6°	7°	8°	9°
$x = 0$	−0.00061	−0.00102	−0.00126	−0.00129	−0.00117	−0.00084	−0.00032	−0.00041	−0.00136
$y = 0$	0.00015	0.00063	0.00143	0.00258	0.00398	0.00577	0.00786	0.01034	0.01318

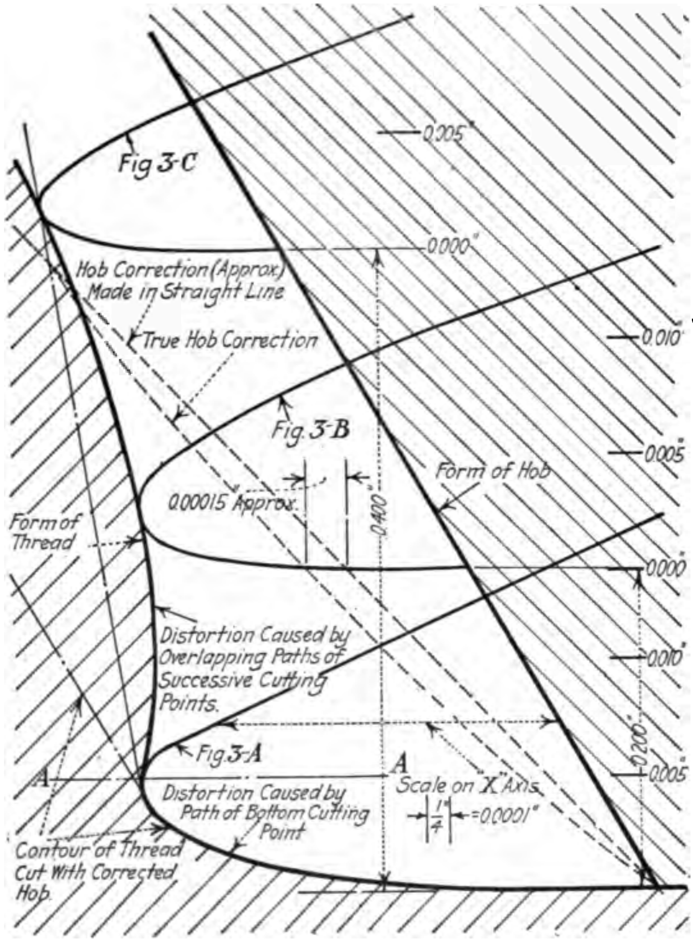


Fig. 4 Curves Showing Shape of Cutting Path

These values are plotted at the left in Fig. 3-B. The actual path of the cutting point is shown at the right.

12 A third point 0.40 in. above the bottom of the thread will next be taken. This point is beyond the top of the thread, but it is

taken to accentuate the distortion developed by hobbing. In this case $R = 1.85$, $r = 0.9625$, $C = 30$ deg., and $N = 4$. The values for x and y with these factors are as follows:

For $B =$ 0°	1°	2°	3°	4°	5°	6°	7°
$x =$ 0	−0.00057	−0.00088	−0.00093	−0.00073	−0.00026	−0.00049	−0.00151
$y =$ 0	−0.00022	0.00087	0.00199	0.00354	0.00566	0.00807	0.01104

The above values are plotted at the left in Fig. 3-C. The actual path of the cutting point is shown at the right.

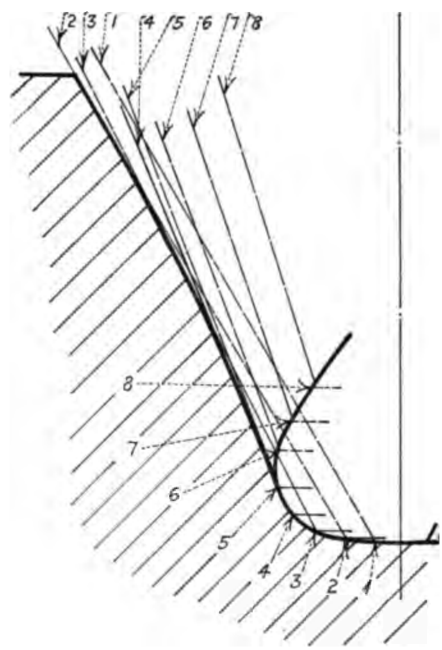


FIG. 5 DIAGRAM SHOWING POSITIONS OF CUTTING EDGE OF HOB IN RELATION TO THREAD

13 In order to show more clearly the nature of the side-cutting of hobs, the curve in Fig. 4 is plotted in an exaggerated form with the intervals along the y-axis equal to 0.001 in., and those along the x-axis equal to 0.0001 in. The curves shown in Figs. 3-A, 3-B and 3-C are plotted to this scale and brought together proportionately; that is, the origins of these curves are spaced equally from the bottom, but these spaces are not to scale. This distorts still further the exact shape of the curve, but its general properties are correct.

14 This curve shows the general nature of the distortion in the form of a thread which is caused by the side-cutting of the hob. It will be noted that it is a double curve, the lower part (below the line A–A) being developed by the bottom corner of the hob tooth while the upper part (above the line A–A) is developed by the overlapping paths of successive cutting points on the cutting face of the hob. It is evident that the distortion shown at the bottom of the curve is inevitable and no correction in the form of the hob is possible that will eliminate it. It can be reduced in many cases, however, by making the hob smaller in diameter, as will be discussed later. On the other hand, the distortion shown in the upper part of the curve can be eliminated by changing the form of the cutting edge on the hob.

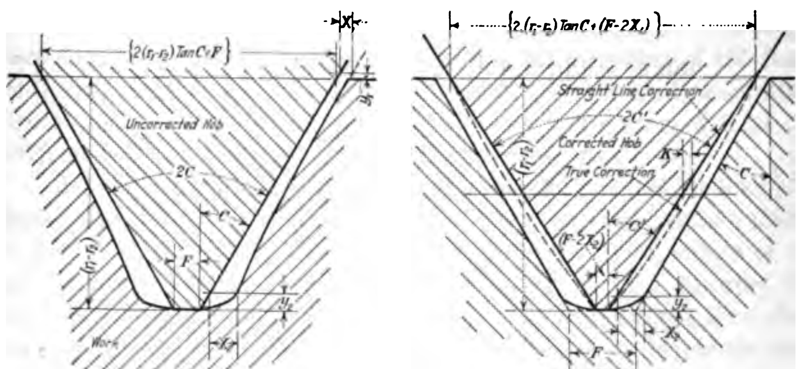
15 Fig. 5 shows graphically the successive positions of the cutting edge of the hob in relation to the thread, illustrating just how the distorted form develops. The line 1–1 shows the flank of the hob tooth when the cutting edge is on the common center line of the hob and the work; 2–2 shows this edge as it is about to leave the upper part of the flank of the thread; 3–3 and 4–4 are finishing intermediate points of the flank; 5–5 shows the cutting edge at the point where the side-cutting is greatest; and 6–6, 7–7, and 8–8 show it in successive positions after it has finished cutting.

16 Referring again to Fig. 4, two dotted lines will be seen, one marked "Hob Correction (approx.) Made in a Straight Line," while the other is marked "True Hob Correction." These represent the contour of the correction. In order to maintain the proper width of thread space, the thickness of the hob teeth must be reduced.

17 The correction of the hob is determined in the following manner: The greatest amount of side-cutting is done by the bottom corner of the hob. In the foregoing example the tabulation shows 0.00194 in. when $B = 6$ deg. This is not necessarily the exact maximum. If a closer value is required, the tabulations must be made with increments of B of smaller amounts. Tabulations as shown, however, will be correct to the fourth decimal place, which is sufficiently accurate for most purposes.

18 If the tooth of the hob be narrowed at each point of the cutting edge by the amount which it side-cuts the thread form, the contour of the thread above the line A–A in Fig. 4 will be correct. To do this exactly would require a curved form on the hob. For example, the maximum side-cutting at the top of the thread form shown in Fig. 4 amounts to 0.00093 in. when $B = 3$ deg., while at

the middle point it amounts to 0.00129 in. when $B = 4$ deg. If the bottom of the hob tooth is reduced 0.00194 in. on a side and the upper point is reduced 0.00093 in. on a side and the two points are connected by a straight line, the middle point will be reduced but 0.00129 in. In this case a correction in a straight line will introduce an error of about 0.00015 in. In most cases it will be found that the correction in a straight line will be sufficiently accurate as the resulting error will be negligible.



A B

Fig. 6 Form of External Thread Cut with Uncorrected and Corrected Hob

19 Fig. 6-A represents the form of a thread cut with a hob having the form of the cutting edges identical with the true form of the thread. In this figure,

C = half included angle of thread
F = width of flat of thread at the root or minor diam.
r_1 = largest value of r employed (radius of major diam.)
r_2 = smallest value of r employed (radius of minor diam.)
x_1 = maximum minus value of x for r_1
x_2 = maximum minus value of x for r_2
$(r_1 - r_2)$ = depth of thread
$2\,(r_1 - r_2)\tan C + F$ = width of space at major diameter (outside).

20 Fig. 6-B illustrates a corrected hob and the form of thread cut with it. In this figure,

$F - 2x_2$ = width of flat at bottom of hob form
$2\,(r_1 - r_2)\tan C + (F - 2x_1)$ = thickness of hob form at top
C' = half the included angle of corrected hob.

21 If the cutting edge of the hob is kept as a straight line, the tangent of half the included angle of the hob form will be equal to half the difference between the widths of hob form at the top and bottom divided by the height of the form. Using the values shown in Fig. 6-B, we have the following:

$$\tan C' = \frac{2\,(r_1 - r_2)\tan C + (F - 2x_1) - (F - 2x_2)}{2\,(r_1 - r_2)}$$

$$\tan C' = \tan C + \frac{F - 2x_1 - F + 2x_2}{2\,(r_1 - r_2)}$$

$$\tan C' = \tan C + \frac{x_2 - x_1}{r_1 - r_2} \quad \ldots\ldots\ldots\ldots\ldots [7]$$

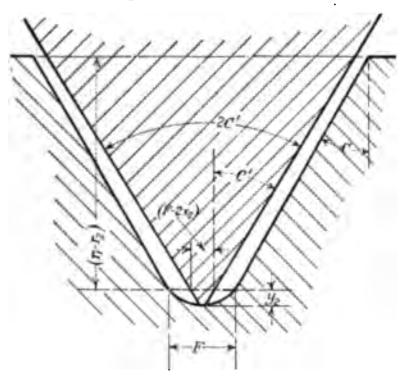


FIG. 7 FORM OF EXTERNAL THREAD CUT WITH POINT OF HOB EXTENDED

It will be seen from Equation [7] and also from the figures that a corrected hob will have a greater included angle than the thread itself; or, in other words, the included angle of a hobbed thread on a screw or male-threaded part will be less than the included angle of the hob.

22 The dotted line in Fig. 6-B represents the true correction of the hob. Also in this figure,

r_3 = value of r at middle of thread flank (or at pitch diameter)

x_3 = maximum minus value of x for r_3

K = difference at r_3 between straight-line correction on hob and true correction

$$\frac{x_2 - x_1}{2} + x_1 = \frac{x_2 - x_1 + 2x_1}{2} = \frac{x_2 + x_1}{2}$$ = correction at r_3 when hob form remains a straight line.

$$K = \frac{x_1 + x_2 - x_3}{2} \quad \ldots\ldots\ldots\ldots [8]$$

23 If the rounding or fillet at the bottom of the thread as shown in Fig. 6-B is objectionable, the point of the hob may be extended by an amount about equal to y_2, provided that such an undercut is permissible. By so doing an almost theoretically perfect thread form will be secured. This condition is illustrated in Fig. 7. Theoretically, the point should extend slightly more than y_2, as at this point r becomes smaller and R becomes larger than the values of r_1 and R_1. Referring to Equation [3], this increase in the value of R will increase the value of r'. Referring to Equation [1], this increase in the value of r^1 and the decrease in the value of r will increase the value of y. But for all practical purposes an extension on the point of the hob of the nearest even dimension to y_2 will usually be sufficiently accurate. On standard threads cut with hobs the resulting error will be in fifth or sixth decimal place.

24 For the purpose of simplifying calculations Tables 1 and 2 have been developed. These tables are based on work whose radius is 1.000. The value of $B/360$ is given opposite varying values of B and values of y/r are given for different values of r/R. In order to use the tables the value of r/R must first be determined, then the value of y/r is multiplied by the radius of the work to obtain the value of y. The values of y/r may be obtained by interpolation when the value of r/R does not agree with any of those given. The values given under $B/360$ are divided by N and substituted in the equation $x = y \tan C - (B/360N)$ to obtain the value of x.

25 In order to illustrate the use of these tables and to determine the effect of varying the diameter of the hob, the following three examples will be taken: An Acme thread, 5 threads per inch, one inch in outside diameter will be cut with hobs (1), one inch in diameter, (2) two inches in diameter, and (3) four inches in diameter. An Acme thread is selected because the smaller included angle of thread results in more side-cutting, thus making more pronounced the effect of varying the diameter of the hob. The depth of an Acme thread of this pitch is 0.110 in. The included angle is 29 deg. The width of the flat at the root of the thread is 0.0689 in.

26 In the first example we have the following:

$C = 14$ deg. 30 min.	$N = 5$
$r_1 = 0.500$ in.	$R_1 = 0.390$ in.
$r_3 = 0.445$ in.	$R_3 = 0.445$ in.
$r_2 = 0.390$ in.	$R_2 = 0.500$ in.

This gives the following values for $\dfrac{r}{R}$:

TABLE I VALUES OF RATIO y/r FOR $r/R = 0.05$ TO 0.45 (EXTERNAL THREADS)

B	$B/360°$	Ratio r/R of Radius of Deepest Point on Work to Radius of Cutting Point on Hob															
		0.05	0.06	0.07	0.08	0.09	0.10	0.12	0.14	0.16	0.18	0.20	0.25	0.30	0.35	0.40	0.45
0° 30′	0.001389	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00004	0.00005	0.00005	0.00005	0.00005	0.00005
1°	0.002778	0.00015	0.00015	0.00015	0.00015	0.00016	0.00016	0.00016	0.00017	0.00017	0.00018	0.00018	0.00019	0.00019	0.00020	0.00021	0.00022
1° 30′	0.004167	0.00034	0.00034	0.00035	0.00035	0.00036	0.00037	0.00038	0.00039	0.00040	0.00041	0.00042	0.00042	0.00043	0.00043	0.00044	0.00044
2°	0.005556	0.00064	0.00065	0.00066	0.00067	0.00068	0.00069	0.00070	0.00071	0.00073	0.00074	0.00075	0.00077	0.00079	0.00082	0.00085	0.00088
2° 30′	0.006944	0.00100	0.00101	0.00102	0.00103	0.00104	0.00105	0.00107	0.00109	0.00111	0.00113	0.00115	0.00118	0.00121	0.00124	0.00128	0.00132
3°	0.008333	0.00148	0.00149	0.00150	0.00151	0.00152	0.00153	0.00156	0.00159	0.00161	0.00164	0.00167	0.00173	0.00179	0.00185	0.00191	0.00197
3° 30′	0.009722	0.00204	0.00205	0.00206	0.00207	0.00208	0.00209	0.00215	0.00220	0.00226	0.00231	0.00237	0.00242	0.00248	0.00254	0.00260	0.00266
4°	0.011111	0.00269	0.00270	0.00271	0.00272	0.00273	0.00274	0.00279	0.00285	0.00291	0.00298	0.00304	0.00313	0.00323	0.00333	0.00343	0.00353
4° 30′	0.012500	0.00328	0.00329	0.00330	0.00331	0.00332	0.00333	0.00340	0.00347	0.00355	0.00362	0.00369	0.00383	0.00397	0.00411	0.00425	0.00440
5°	0.013889	0.00406	0.00407	0.00408	0.00409	0.00410	0.00411	0.00420	0.00429	0.00438	0.00447	0.00456	0.00475	0.00494	0.00513	0.00532	0.00551
6°	0.016667	0.00598	0.00599	0.00601	0.00602	0.00604	0.00605	0.00616	0.00626	0.00637	0.00647	0.00658	0.00686	0.00714	0.00742	0.00770	0.00798
7°	0.019444	0.00822	0.00824	0.00826	0.00827	0.00829	0.00831	0.00843	0.00856	0.00868	0.00881	0.00893	0.00932	0.00971	0.01011	0.01051	0.01091
8°	0.022222	0.01068	0.01072	0.01077	0.01081	0.01086	0.01090	0.01107	0.01125	0.01142	0.01160	0.01177	0.01228	0.01279	0.01330	0.01382	0.01434
9°	0.025000	0.01312	0.01350	0.01358	0.01367	0.01375	0.01383	0.01406	0.01429	0.01453	0.01476	0.01499	0.01564	0.01629	0.01695	0.01761	0.01827
10°	0.027778	0.01646	0.01659	0.01672	0.01685	0.01698	0.01711	0.01741	0.01771	0.01800	0.01830	0.01860	0.01942	0.02024	0.02106	0.02188	0.02270
11°	0.030556	0.01984	0.02002	0.02020	0.02039	0.02057	0.02075	0.02112	0.02149	0.02187	0.02224	0.02261	0.02361	0.02461	0.02562	0.02663	0.02764
12°	0.033333	0.02357	0.02380	0.02405	0.02428	0.02452	0.02476	0.02521	0.02567	0.02612	0.02658	0.02703	0.02824	0.02945	0.03067	0.03189	0.03311
13°	0.036111	0.02766	0.02796	0.02826	0.02855	0.02885	0.02915	0.02969	0.03024	0.03078	0.03133	0.03187	0.03331	0.03476	0.03621	0.03766	0.03911
14°	0.038889	0.03212	0.03248	0.03284	0.03321	0.03357	0.03393	0.03457	0.03521	0.03586	0.03650	0.03714	0.03884	0.04054	0.04224	0.04394	0.04565
15°	0.041667	0.03696	0.03739	0.03782	0.03825	0.03868	0.03911	0.03986	0.04061	0.04135	0.04210	0.04285	0.04483	0.04681	0.04879	0.05077	0.05275

$$\frac{r_1}{R_1} = \frac{0.500}{0.390} = 1.28 \qquad (a)$$

$$\frac{r_3}{R_3} = \frac{0.445}{0.445} = 1.00 \qquad (b)$$

$$\frac{r_2}{R_2} = \frac{0.390}{0.500} = 0.78 \qquad (c)$$

TABLE 2 VALUES OF y/r FOR r/R = 0.50 TO 2.00 (EXTERNAL THREADS)

B	B/360°	Ratio r/R							
		0.50	0.60	0.70	0.80	0.90	1.00	1.50	2.00
0° 30′	0.001389	0.00006	0.00006	0.00006	0.00007	0.00007	0.00008	0.00010	0.00012
40′	0.001852	0.00010	0.00011	0.00011	0.00012	0.00013	0.00014	0.00017	0.00021
50′	0.002315	0.00016	0.00017	0.00018	0.00019	0.00020	0.00022	0.00027	0.00032
1°	0.002778	0.00023	0.00024	0.00025	0.00026	0.00028	0.00030	0.00038	0.00046
1° 10′	0.003241	0.00027	0.00030	0.00033	0.00036	0.00039	0.00042	0.00053	0.00064
1° 20′	0.003704	0.00034	0.00038	0.00042	0.00046	0.00050	0.00055	0.00069	0.00083
1° 30′	0.004167	0.00045	0.00049	0.00054	0.00059	0.00064	0.00069	0.00086	0.00103
1° 40′	0.004630	0.00057	0.00062	0.00068	0.00074	0.00080	0.00086	0.00106	0.00126
1° 50′	0.005093	0.00073	0.00079	0.00085	0.00091	0.00097	0.00103	0.00127	0.00151
2°	0.005556	0.00091	0.00096	0.00102	0.00108	0.00114	0.00120	0.00151	0.00183
2° 30′	0.006944	0.00136	0.00146	0.00156	0.00167	0.00178	0.00189	0.00238	0.00287
3°	0.008333	0.00204	0.00217	0.00230	0.00244	0.00258	0.00272	0.00344	0.00416
3° 30′	0.009722	0 00272	0.00292	0 00312	0.00332	0.00352	0.00373	0.00469	0.00566
4°	0.011111	0.00363	0.00388	0.00413	0.00438	0.00463	0.00488	0.00614	0.00740
4° 30′	0.012500	0.00455	0.00488	0.00521	0.00554	0.00587	0.00620	0.00781	0.00943
5°	0.013889	0.00570	0.00609	0 00649	0.00689	0.00729	0.00769	0.00969	0.01169
6°	0.016667	0.00826	0 00884	0 00942	0 01000	0.01059	0.01118	0.01408	0.01701
7°	0.019444	0.01131	0.01212	0 01293	0 01374	0.01455	0.01537	0.01938	0.02343
8°	0.022222	0.01486	0.01594	0.01702	0.01810	0.01919	0.02028	0.02563	0.03107
9°	0.025000	0.01893	0.02033	0.02173	0.02313	0 02453	0.02593	0.03293	0.04002
10°	0.027778	0.02353	0.02529	0.02705	0.02881	0.03057	0.03233	0.04132	0.05041
11°	0.030556	0.02865	0.03082	0.03299	0 03516	0.03733	0.03951	0.05084	0.06247
12°	0.033333	0.03433	0.03696	0.03959	0.04222	0.04485	0.04748	0 06153	0.07650
13°	0.036111	0.04056	0.04370	0.04684	0.04998	0 05312	0.05627	0.07343	0.09258
14°	0.038889	0.04736	0.05106	0.05476	0.05847	0.06218	0.06589	0.08658	0.11147
15°	0.041667	0.05173	0.05905	0.06337	0.06770	0.07203	0 07636	0.10102	0.13380

27 The values of x and y for the ratio (a) will first be computed. Referring to Table 2, and interpolating between r/R = 1.00 and r/R = 1.50 for r/R = 1.28 gives the values for y/r. Dividing the values of $B/360$ in this same table by the value of N (= 5) gives the values of $B/360\,N$. These values are as follows:

For $B =$	5°	6°	7°	8°
$\frac{y}{r} =$	0.00881	0.01280	0.01762	0.02328
$\frac{B}{360N} =$	0.002778	0.003333	0.003889	0.004444

Multiplying the values of y/r by r (=0.500) gives the values of y. Substituting these values of y and the values of $B/360N$ in the equation $x = y \tan C - B/360N$, gives the values of x. These values thus obtained are as follows:

For $B =$	5°	6°	7°
$x =$	-0.001637	-0.001678	-0.001611
$y =$	0.004405	0.006400	0.008810

As soon as the maximum minus value of x is determined, it is unnecessary to proceed further.

28 In like manner the values of x and y for the ratio (b) are determined. These are as follows:

For $B =$	6°	7°	8°
$x =$	-0.002046	-0.002123	-0.002110
$y =$	0.004975	0.006830	0.009025

29 The values of x and y for ratio (c) are as follows:

For $B =$	7°	8°	9°	10°
$x =$	-0.002519	-0.002641	-0.002694	-0.002671
$y =$	0.005296	0.006973	0.008912	0.011099

30 Using Equation [7] the correction in the angle of the hob form is computed. The tabulations above give the following factors:

$$r_1 = 0.500 \qquad x_2 = 0.002695$$
$$r_2 = 0.390 \qquad x_1 = 0.001678$$
$$r_1 - r_2 = 0.110 \qquad x_2 - x_1 = 0.001017$$
$$\tan C = \tan (14° \, 30') = 0.25862$$

Substituting these values in Equation [7] we have

$$\tan C' = 0.25862 + \frac{0.001017}{0.110} = 0.26786$$
$$C' = 14° \, 59' \, 43''$$
$$2\,C' = 29° \, 59' \, 26''$$

In this case the difference between the angle of the hob and the angle of the work is 59 min. 26 sec.

31 The above correction is made in a straight line. Equation [8] will show the difference between this correction and the true correction at the middle of the flank; thus:

$$x_1 + x_2 = 0.004373$$
$$x_3 = 0.002123$$
$$K = \frac{x_1 + x_2}{2} - x_3 = \frac{0.004373}{2} - 0.002123 = 0.000064 \text{ in.}$$

This is negligible and can safely be ignored. In fact, it is much less than the probable error in the hob.

32 Referring to Fig. 6-B, the dimensions of the corrected hob and the thread produced will be as follows:

Work	Hob
$F = 0.0689$ in.	$F - 2x_2 = 0.0635$ in.
$2C = 29°$	$2C' = 29° 59'$
$y_2 = 0.0089$	

This means that the corrected hob will cut an almost theoretically perfect thread form above the fillet which extends 0.0089 in. above the root of the thread. In an Acme thread a clearance of 0.010 in. is provided at this point, so that no further correction need be made. If desired, however, the point of the hob could be extended about 0.010 in. as shown in Fig. 7.

33 In like manner the following values are determined in the second example:

$$2C' = 30° 0' 16'' \qquad F - 2x_2 = 0.0619 \text{ in.}$$
$$K = 0.000054 \text{ in.} \qquad y_2 = 0.012344 \text{ in.}$$

34 The values in the third example are as follows:

$$2C' = 30° 0' 54'' \qquad F - 2x_2 = 0.0608 \text{ in.}$$
$$K = 0.000026 \text{ in.} \qquad y_2 = 0.014422 \text{ in.}$$

35 The following tabulation is made to show the various effects of varying the diameter of the hob. (Fig. 6-B.)

Hob. diam.	= 1.000 in.	2.0000 in.	4.000 in.
x_2	= 0.002695	0.003475	0.004048
y_2	= 0.008912	0.012344	0.014422
$2C'$	= 29° 59′ 26″	30° 0′ 16″	30° 0′ 54″
$F - 2x_2$	= 0.0635	0.0619	0.0608
K	= 0.000064	0.000054	0.000026

36 From the above tabulation it will be seen that the amount of side-cutting at the bottom of the thread (x_2) increases as the diameter of the hob is increased. The height of the fillet at the bottom of the thread (y_2) also increases as the diameter of the hob increases, in fact, it increases about three times as much in this case as x_2. The included angle of the corrected hob ($2C'$) increases very

slightly as the diameter of the hob increases. The width of the point of the corrected hob $(F - 2x_2)$ varies less than 0.003 in. as the diameter of the hob is increased from 1 in. to 4 in. The dimension K in Fig. 6-B is reduced as the diameter of the hob increases.

37 The correction for angle used in these solutions is a chordal correction. This shows a slight change in angle as the diameter of the hob varies. A geometrical solution of this side-cutting will show that, except for the fillet at the root of the thread, the profile is un-

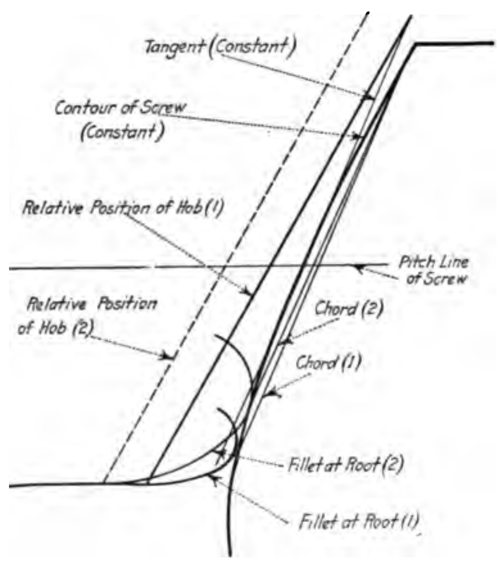


FIG. 8 DIAGRAM SHOWING CORRECTION FOR ANGLE

changed regardless of the diameter of the hob. Therefore, if the correction is taken as the tangent of the profile at the pitch line of the thread, the angle C' will remain unchanged. This is shown in Fig. 8. The formulæ for determining this angle are as follows:

Let $C = \frac{1}{2}$ included angle of thread
$C' = \frac{1}{2}$ included angle of corrected hob
$H =$ helix angle of thread at pitch line

Then

$$\tan H = \frac{1}{\pi N \times \text{pitch diam. of thread}}$$

and

$$\tan^2 C' = \tan^2 C + \tan^2 H$$

38 It is evident from the above that if the form of the cutting edge of a hob of a certain diameter be corrected to cut a thread of a certain diameter and pitch, a variation in the diameter of the hob of, say, 25 per cent will have but little effect on the form of the thread produced on the work.

39 In order to determine the effect of varying the diameter of the work, the following tabulation was made for a hob 2.000 in. in diameter with 5 Acme threads per inch:

Work diam.	= 1.000 in.	2.000 in.	4.000 in.
x_2	= 0.003475	0.001161	0.000350
y_2	= 0.012344	0.004103	0.001512
$2C'$	= 30° 0′ 16″	29° 14′ 8″	29° 3′ 6″
$F - 2x_2$	= 0.0619	0.0666	0.0682
K	= 0.000054	0.000012	0.000003

40 The above tabulation shows that the amount of side-cutting at the bottom of the thread (x_2) decreases as the diameter of the work increases. The height of the fillet at the bottom of the thread (y_2) decreases as the diameter of the work increases. In this case it decreases about four times as much as x_2. The included angle of the corrected hob decreases quite rapidly as the diameter of the work increases. The width of the point of the corrected hob increases about 0.006 in. as the diameter of the work is increased from 1 in. to 4 in. The dimension K in Fig. 6-B decreases as the diameter of the work increases. In this case the amount of error introduced by a straight-line correction in the hob is in the fifth and sixth decimal place and is negligible.

41 It is evident, therefore, that a hob which is corrected for a certain diameter of work cannot be used on work which varies very much in diameter if accurate results are desired. The smaller the diameter of the work, the more this condition is accentuated. This is due, in large measure, to the rapid increase of the helix angle on smaller diameters. On work of large diameters, where the helix angle is very small, little or no correction is required on the hob.

II. THE HOBBING OF INTERNALLY THREADED PARTS

42 The hobbing of internally threaded parts, when the axis of the hob is parallel to the axis of the thread which is being cut, develops conditions which are similar in many respects to those which occur in the hobbing of screws which have previously been discussed. The same symbols will be used in this discussion as were used in regard to the hobbing of screws and formulæ will be derived to show

the interference between the path of any cutting point on the hob and the flanks of the thread.

43 Fig. 9 shows a diagram of a hob and the thread in a nut which it is cutting. It will be noted that the value of y is:

$$y = r - r' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

The equation for x is identical with Equation [6] for hobbing screws, namely,

$$x = y \tan C - \frac{B}{360N}$$

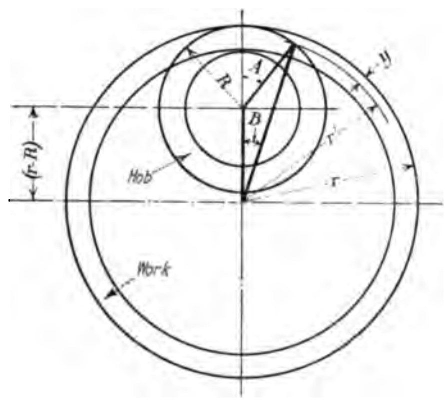


FIG. 9 DIAGRAM OF HOB AND INTERNAL SCREW

In order to determine the value of y, the triangle shown in heavy lines in Fig. 9 must be solved. The known values will be taken as r, R, and B. From the relation

$$(r-R) \sin B = R \sin (A-B)$$

$$\frac{(r-R) \sin B}{R} = \sin (A-B) \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

And given the value of B and $(A-B)$, the value of A is readily determined. Solving the triangle for r', we have

$$r' = \frac{R \sin (180° - A)}{\sin B} = R \frac{\sin A}{\sin B} \quad . \quad . \quad . \quad . \quad . \quad [11]$$

The value of y is then determined by Equation [9].

44 In order to determine the nature of the side-cutting the following example will be taken: A hob 0.800 in. in outside diameter will be used to hob an internal Acme thread, 2.000 in. in outside diameter, 4 threads per inch.

45 First determine the maximum side-cutting at the largest diameter of the work. The known values are then $r = 1.000$, $R = 0.400$, $N = 4$, and $C = 14\frac{1}{2}$ deg., from which the following coefficients are obtained:

For B =	5°	6°	7°
x =	− 0.001995	− 0.002039	− 0.001928
y =	0.005710	0.008230	0.011240

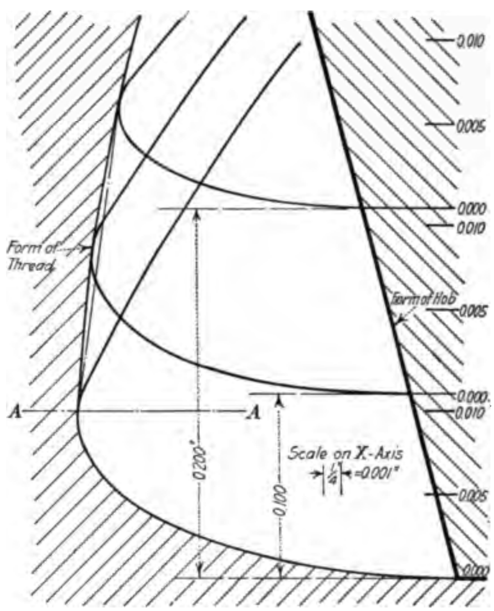


FIG. 10 CURVES SHOWING FORM OF INTERNAL THREAD

46 The side-cutting at a point 0.100 in. farther down the flank of the thread will next be determined. In this case $r = 0.900$ in., $R = 0.300$ in. and these give the following coefficients:

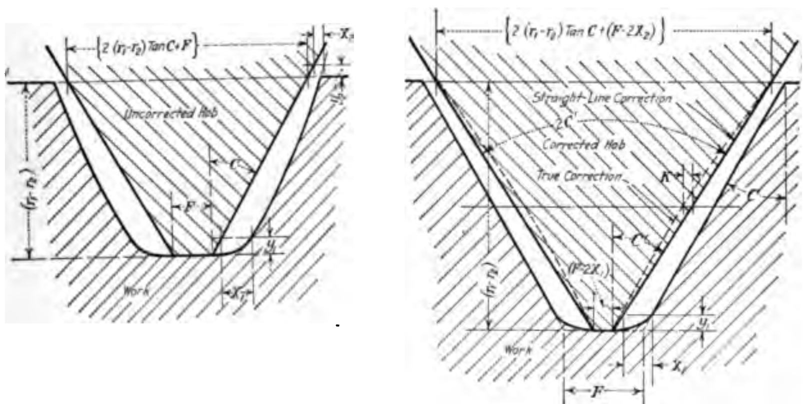
For B =	4°	5°	6°
x =	− 0.001644	− 0.001698	− 0.001607
y =	0.004383	0.006858	0.009900

47 When the side-cutting is at a point 0.200 in. below the largest diameter of the work, $r = 0.800$ in. and $R = 0.200$ in., giving the following coefficients:

For B =	3° 30′	4°	4° 30′
x =	− 0.001297	− 0.001303	− 0.001236
y =	0.004384	0.005704	0.007304

48 These values are plotted in Fig. 10. In general, the curve is very similar to the one shown in Fig. 4, except that the flank of the thread in the work is concave instead of convex as in the case of a hobbed screw. It will also be found that on hobbed nut threads the curvature of the flank of the thread is much less than on hobbed screw threads, while the amount of side-cutting at the point, both on the x-axis and y-axis, is much greater; in other words, the fillet at the bottom of the hobbed thread will be larger.

49 Fig. 11-A represents the form of the thread which would be cut with a hob whose form was that of a standard thread. The values employed are the same as those used in discussing the hobbing



A B

FIG. 11 FORM OF INTERNAL THREAD CUT WITH UNCORRECTED AND CORRECTED HOB

of screws. Fig. 11-B represents a corrected hob and the form of thread cut with it. The formulæ for the corrected angle of the hob will be very similar to that for external hobbing, and is given by the equation:

$$\tan C' = \tan C + \frac{x_1 - x_2}{r_1 - r_2} \quad \text{[12]}$$

50 As with the external hobs, the true correction of the hob would be in a curved line. This is indicated by the dotted line in Fig. 10-B. In this case, however, the flank of the hob should be concave instead of convex as in the case of external hobs. If the fillet at the bottom of the thread should be objectionable, the point of the hob could be extended by an amount equal approximately to y_2 if under-cutting was permissible.

51 In the example given previously, we have the following values:

$C^1 = 14° 41' 52''$ $\qquad x_1 = 0.002039$ in.
$K = 0.000027$ in. $\qquad y_1 = 0.008230$ in.

In order to determine the effect of varying the diameter of the hob, two further examples will be taken: First, with the outside diameter of the hob equal to 1.200 in. and second, with it equal to 1.600 in.

52 The following tabulation is made to show the effects of varying the diameter of the hob. (Diameter of work = 2.000 in.; 4 Acme threads per inch, see Fig. 11-A):

Diam. of hob = 0.800 in.	1.200 in.	1.600 in.
x_1 = 0.002039	0.004615	0.012734
y_1 = 0.00823	0.017063	0.052800
$2C'$ = 29° 23′ 44″	29° 25′ 6″	29° 28′ 8″

It will be noted that the same general principles hold true, as regards the effect of increasing the diameter of the hob, for both external and internal hobbing. The amount of side-cutting (x_1) and the height of the fillet (y_1) increase considerably as the diameter of the hob increases, while the included angle of the corrected hob increases but very slightly. If a tangential correction at the pitch line is used, the formula for determining C' given for screws can also be used for nuts.

53 In order to determine the effect of varying the diameter of the work, the following tabulation is made:

Diam. of work = 2.000 in.	3.000 in.	4.000 in.
x_1 = 0.012734	0.002342	0.001028
y_1 = 0.052800	0.009739	0.004080
$2C'$ = 29° 28′ 8″	29° 10′ 24″	29° 5′ 52″

This tabulation shows the same general conditions as exist in external hobbing. As the diameter of the work increases, the side-cutting (x_1), the height of the fillet (y_1) and the angle of the corrected hob decrease quite rapidly. Thus a hob which is corrected for a certain diameter of work would introduce a noticeable error when used for cutting a different diameter. The amount of this change also decreases quite rapidly as the diameter of the work increases. Beyond a certain diameter of work, therefore, no correction would be practically required.

54 It will be noted, if comparison is made with external hobbing, that the amount of side cutting in internal hobbing is very much greater although the correction in angle is somewhat similar.

TABLE 3 VALUES OF y/r FOR r/R = 1.1 TO 4.0 (INTERNAL THREADS)

B	$B/360°$	Ratio r/R													
		1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0	2.5	3.0	3.5	4.0
1°	0.002778	0.00001	0.00001	0.00001	0.00002	0.00002	0.00003	0.00004	0.00005	0.00006	0.00007	0.00019	0.00031	0.00039	0.00046
2°	0.005556	0.00005	0.00010	0.00015	0.00020	0.00025	0.00031	0.00037	0.00043	0.00049	0.00055	0.00089	0.00122	0.00147	0.00178
3°	0.008333	0.00010	0.00023	0.00036	0.00049	0.00063	0.00077	0.00091	0.00105	0.00119	0.00133	0.00204	0.00274	0.00336	0.00398
4°	0.011111	0.00021	0.00045	0.00069	0.00093	0.00117	0.00141	0.00166	0.00191	0.00216	0.00241	0.00364	0.00487	0.00600	0.00713
5°	0.013889	0.00035	0.00073	0.00111	0.00149	0.00187	0.00225	0.00263	0.00301	0.00340	0.00379	0.00571	0.00762	0.00945	0.01128
6°	0.016667	0.00052	0.00107	0.00162	0.00217	0.00272	0.00327	0.00382	0.00437	0.00492	0.00547	0.00823	0.01100	0.01369	0.01637
7°	0.019444	0.00072	0.00146	0.00220	0.00295	0.00370	0.00445	0.00520	0.00595	0.00670	0.00745	0.01124	0.01502	0.01879	0.02255
8°	0.022222	0.00095	0.00192	0.00289	0.00386	0.00483	0.00581	0.00679	0.00777	0.00875	0.00973	0.01470	0.01967	0.02477	0.02987
9°	0.025000	0.00121	0.00244	0.00367	0.00490	0.00613	0.00736	0.00859	0.00983	0.01107	0.01231	0.01864	0.02498	0.03167	0.03877
10°	0.027778	0.00150	0.00302	0.00454	0.00606	0.00758	0.00910	0.01062	0.01214	0.01366	0.01519	0.02306	0.03094	0.03952	0.04811
11°	0.030556	0.00182	0.00365	0.00549	0.00733	0.00917	0.01101	0.01285	0.01479	0.01663	0.01837	0.02797	0.03757	0.04835	0.05913
12°	0.033333	0.00217	0.00435	0.00653	0.00871	0.01090	0.01309	0.01528	0.01747	0.01966	0.02185	0.03331	0.04486	0.05817	0.07148
13°	0.036111	0.00255	0.00511	0.00767	0.01023	0.01279	0.01535	0.01792	0.02049	0.02306	0.02563	0.03923	0.05283	0.06902	0.08521
14°	0.038889	0.00296	0.00593	0.00890	0.01187	0.01484	0.01781	0.02078	0.02375	0.02673	0.02971	0.04560	0.06149	0.08092	0.10036
15°	0.041667	0.00340	0.00681	0.01022	0.01363	0.01704	0.02045	0.02386	0.02727	0.03068	0.03409	0.05246	0.07084	0.09392	0.11699
16°	0.044444	0.00387	0.00774	0.01161	0.01548	0.01935	0.02323	0.02711	0.03099	0.03487	0.03875	0.05983	0.08091	0.10807	
17°	0.047222	0.00437	0.00874	0.01311	0.01748	0.02185	0.02622	0.03059	0.03496	0.03933	0.04371	0.06777	0.09183	0.12341	
18°	0.050000	0.00489	0.00978	0.01467	0.01956	0.02446	0.02936	0.03426	0.03916	0.04406	0.04896	0.07630	0.10365	0.14000	
19°	0.052778	0.00544	0.01089	0.01634	0.02179	0.02724	0.03269	0.03814	0.04359	0.04904	0.05449	0.08548	0.11647	0.15789	
20°	0.055556	0.00602	0.01205	0.01808	0.02411	0.03014	0.03617	0.04220	0.04823	0.05427	0.06031	0.09535	0.13039	0.17713	
21°	0.058333	0.00663	0.01327	0.01991	0.02655	0.03319	0.03983	0.04647	0.05311	0.05976	0.06641	0.10593	0.14546		
22°	0.061111	0.00726	0.01454	0.02182	0.02910	0.03638	0.04366	0.05094	0.05822	0.06551	0.07280	0.11725	0.16171		
23°	0.063889	0.00792	0.01587	0.02382	0.03177	0.03972	0.04767	0.05562	0.06357	0.07152	0.07947	0.12932	0.17917		
24°	0.066667	0.00861	0.01725	0.02589	0.03454	0.04319	0.05184	0.06049	0.06914	0.07779	0.08644	0.14213	0.19783		
25°	0.069444	0.00933	0.01870	0.02807	0.03744	0.04681	0.05618	0.06555	0.07493	0.08431	0.09369	0.15567	0.21766		

55 In order to simplify the calculations, Tables 3 and 4 have been prepared. These are similar to Tables 1 and 2, and are to be used in exactly the same manner.

56 It has been a common belief that for internal hobbing the

TABLE 4 VALUES OF y/r FOR $r/R = 1.1$ TO 1.7 (INTERNAL THREADS)

B	$B/360°$	Ratio r/R						
		1.1	1.2	1.3	1.4	1.5	1.6	1.7
26°	0.072222	0.01008	0.02020	0.03032	0.04044	0.05056	0.06069	0.07082
27°	0.075000	0.01085	0.02175	0.03265	0.04355	0.05445	0.06536	0.07627
28°	0.077778	0.01165	0.02336	0.03507	0.04678	0.05849	0.07020	0.08191
29°	0.080556	0.01247	0.02501	0.03755	0.05009	0.06263	0.07518	0.08773
30°	0.083333	0.01331	0.02671	0.04011	0.05351	0.06692	0.08033	0.09374
31°	0.086111	0.01418	0.02847	0.04276	0.05705	0.07134	0.08584	0.09994
32°	0.088889	0.01507	0.03028	0.04549	0.06070	0.07591	0.09112	0.10633
33°	0.091667	0.01599	0.03213	0.04828	0.06443	0.08058	0.09673	0.11288
34°	0.094444	0.01694	0.03405	0.05116	0.06827	0.08538	0.10249	0.11960
35°	0.097222	0.01791	0.03601	0.05411	0.07221	0.09031	0.10841	0.12651
36°	0.100000	0.01891	0.03802	0.05713	0.07624	0.09535	0.11446	0.13358
37°	0.102778	0.01993	0.04008	0.06023	0.08038	0.10053	0.12068	0.14084
38°	0.105556	0.02097	0.04219	0.06341	0.08463	0.10585	0.12707	0.14829
39°	0.108333	0.02205	0.04435	0.06666	0.08897	0.11128	0.13359	0.15590
40°	0.111111	0.02315	0.04657	0.06999	0.09341	0.11683	0.14025	0.16367
41°	0.113889	0.02427	0.04882	0.07337	0.09792	0.12248	0.14704	0.17160
42°	0.116667	0.02540	0.05111	0.07682	0.10254	0.12826	0.15398	0.17970
43°	0.119444	0.02655	0.05345	0.08035	0.10725	0.13415	0.16105	0.18796
44°	0.122222	0.02772	0.05583	0.08394	0.11205	0.14016	0.16827	0.19638
45°	0.125000	0.02891	0.05825	0.08759	0.11694	0.14629	0.17564	0.20499

hob should be made as small as possible. As a matter of fact, any 60-deg. thread which does not have a sharper helix than any of the U. S. standard threads can be hobbed satisfactorily, with as large a hob as will clear the tap-drill diameter, provided the hob is suitably corrected. The only effect, beyond a slight modification in the hob form, which is caused by the use of a smaller hob is the reduction of the height of fillet (y_1) at the bottom of the hob. To overcome this, when its height may be objectionable, the point of the hob may be extended, thus producing a slight undercut, which is nearly always permissible.

57 Due to the above-mentioned misapprehension in regard to internal hobbing, this method of manufacture has been seldom used except on threads of about two inches and over in diameter. It will be seen from the foregoing, however, that it is practical for threads three-quarters of an inch in diameter and upward, and under some circumstances it can be successfully used on even smaller diameters.

DISCUSSION

R. E. FLANDERS. One side of this problem is very interesting from the standpoint of the work being done with the Hartness screw-thread comparator. Small errors show up large if a thread is magnified; but even when these errors are too insignificant to be of practical importance, they cause a feeling of dissatisfaction in the mind of the person who observes them. This is one of the dangers of using the thread comparator. Good enough does not look so good when magnified in this way, and this is reason enough for the author of the paper to examine these small errors in hobs which are supposed to be tools of precision. A high degree of accuracy has a certain advantage in selling threaded products, and in selling taps and dies. It is comforting to think of these five- and ten-minute errors as being corrected, and the manufacturer finds himself in a much better position in so far as the acceptability of his product is concerned.

HOWARD P. FAIRFIELD. The paper interests me from the viewpoint of the present tendency of not permitting an error to persist because of itself it is relatively unimportant. The present tendency of engineers and scientists is wholly toward knowing what the errors of construction are and learning how they may be corrected. This can be seen, for example, in the recent great advances in methods used in gaging and measurements, and the author has done engineers a real service by his researches in hobbing, showing that errors exist, what they are, and how they may be corrected. Even admitting that uncorrected hobbed threads can be effectively used in construction, the paper is a benefit to the art of hobbing threads, in that it increases our knowledge of the subject and leaves nothing to uncertainty.

JAMES HARTNESS. There is a very good reason for considering the elements which the author has set forth. These small elements, which apparently need no correction, are the very things which must be corrected if more dependable pieces of mechanism are to be obtained, and a dependable screw is certainly desirable. The author's contribution is invaluable on this score.

HENRY J. EBERHARDT desired to add his word of appreciation of the paper inasmuch as it emphasized the necessity for standardizing some of the tools for making threads by this process. For instance, he said, there was the need for standardizing the diameter of the cutter used for standard threads, such as used in the S.A.E. thread, as the diameter variation would affect this error. It was also desirable to standardize on the diameter of disc cutters and end mill cutters for producing worms, and in grinding worms it was desirable to standardize the diameter of the grinding wheel.

THE AUTHOR. In the unabridged paper from which the present was taken, some examples were given of standard U.S. threads. In revising the paper to put it in briefer form, Acme threads were used because they accentuated the side cutting conditions. On the average U.S. standard pitches, which, of course, are coarser than those used in the automotive industry, the corrections in angle would run up to 20 min., and from there down to about 5 min. For the S.A.E. pitches they are usually much less than the probable error in the hob itself.

A similar correction is necessary when a thread is rolled as when it is hobbed, although the side cutting is much less in the rolled thread. The formulæ for thread rolling dies are as follows:

For single threaded rolls

$$y = r' - r$$

$$x = y \tan C - \frac{B - A}{360\ N}$$

For double threaded rolls

$$y = r' - r$$

$$x = y \tan C - \frac{B - 2A}{360\ N}$$

For triple threaded rolls

$$y = r' - r$$

$$x = y \tan C - \frac{B - 3A}{360\ N}$$

No. 1764

CYLINDRICAL GRINDING IN 1920

By W. H. Chapman, Worcester, Mass.
Associate-Member of the Society

This paper comprises a study of the laws involved in cylindrical grinding and an analysis of grinding action (1) for draw-in cuts and (2) for traversed cuts.

Grinding efficiency is usually considered as (production) ÷ (wheel wear). Accordingly formulas are derived for wheel wear in terms of grain size of wheel, work speed, wheel speed, feed, etc. By calculating values for wheel wear for different conditions and comparing them with production figures calculated under the same conditions, a proper selection of wheels may be made.

The fact that wheel wear is a comparatively unimportant item in the total cost of grinding, however, frequently leads to wrong conclusions if this is made the controlling factor. The all-important factor is the rate at which the wheel may be made to cut and still not get out of truth. Assuming wheel cost to be, for example, 7 cents per cu. in. and other production costs 5 cents per minute, it is shown that production efficiency is materially increased by increasing production at the expense of wheel wear within limits which are defined. The paper concludes with a discussion of production costs on this basis and with a series of practical conclusions, one of the most important of which is that increase of traverse speed increases production without increase of wheel-wearing action.

INDUSTRIAL engineers have long realized the value of the application of science to the operation of machine tools. One has but to consider the work of Taylor and others to appreciate this. As the grinding machine has come to be an important production machine only within a relatively short period, there has not been a large amount of accurate information available with regard to its operating features, more especially from the standpoint of obtaining the most economical grinding conditions. It is the purpose of this paper to report some recent developments along these lines.

THEORETICAL CONSIDERATIONS

2 In December, 1914, Prof. Geo. I. Alden presented a paper to the Society setting forth his theory of Grain Depth of Cut. A formula was developed involving the velocities of the wheel and

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work surfaces, the grain size of the wheel (number of cutting particles per unit length of circumference) and sine functions of the angle formed by tangents to the work and wheel circles at their point of intersection. This work was a pioneer of its kind in this country. It clearly showed that the kinematic relations between the cutting grains and the work surface were of the greatest importance during cylindrical grinding operations, and that wheel wear and production rates would vary according to the mathematical laws involved.

3 In his book Grinding Machinery, published in England, Mr. J. J. Guest develops expressions to indicate the mathematical relationships between the cutting grains and the work surface. He calculates what he terms "Normal Velocity of Material," "Controlling Factor," and "Maximum Output," and he finds that a change of work speed has an effect upon wheel wear far greater than a change in depth of cut. This agrees with Professor Alden's conclusions. Mr. Guest deduces a formula for Controlling Factor, in which the work speed enters as a squared function and depth of cut or feed enters as a first power.

4 The operator must know how to select wheels of the proper abrasive, grain and grade for the piece to be ground. This he will determine by experimenting until he learns what to choose in any particular case.

5 The variables which must be controlled by the operator, once he has chosen his wheel and set up for the job, are:

a Wheel speed — usually constant r.p.m. with surface speed decreasing as wheel wears
b Work speed — variable r.p.m. provided over a large range
c Traverse speed — variable from about 3 to 36 ft. per min. in the latest machines
d Depth of cut — controlled by graduated feed mechanism, graduated in units of 0.00025 in. on work diameter
e Rate of feed — uniform in-feed for non-traversed work, in-feed at end of traverse zone for traversed work
f Ratio of traverse per work revolution to width of wheel face.

6 We will assume the wheel speed to be constant, since a well-designed machine will be sufficiently powered to maintain its speeds, even under reasonably heavy cuts. Theoretically, the traverse per revolution should be sufficient to allow the wheel face to just cover the lead and not present the same work surface to the cutting face more than once per traverse. This condition can be brought about

by computing the lead of the work for the various combinations of work and traverse speeds, or from inspection of the work itself while grinding is in process. In the following mathematical considerations the exact coverage of the lead by the wheel face is assumed.

7 The rate of feed should theoretically be one increment per revolution for non-traversed cuts and one increment per traverse for traversed cuts. A traverse is considered to be a single stroke between the traverse limits, and may therefore be either left to right or right to left. Where one traverse limit is at the driven end and the wheel cannot be allowed to overtravel the end of the work, a non-traversed cut should be taken down to very nearly the finished size, and then the traverse limit established so that the wheel will not strike the shoulder on the limit side of this cut, but will overtravel the opposite side of the cut for part of the wheel width. This practice saves the edge of the wheel on the side toward the work dog.

8 Having established our method of grinding, we are now prepared to study the scientific laws involved and their operation under these conditions.

MATHEMATICAL ANALYSES

9 In order to analyze the grinding action we must visualize just what occurs as the grains pass through the arc of contact with the revolving work piece. This may be done by imagining a cross-section through the wheel and work, and by studying action in the section plane, the entire wheel action being the integration of all section planes normal to the axes of rotation of the wheel and the work. We may also greatly magnify the size of the grains and the dimensions of the rotating members, for the same mathematical relations obtain regardless of actual dimensional values. Therefore, in the accompanying diagrams the dimensions are not in proportion to those actually existing, but have been distorted for the purpose of exaggerating the conditions within the arc of contact. The depth of cut, size of chip, and size of grains are all greatly exaggerated. This, however, should not result in any confusion as to the proper illustration of the theoretical grinding action.

ANALYSIS OF GRINDING ACTION FOR DRAW-IN (NON-TRAVERSED) CUTS

10 We will first consider the case of draw-in cuts where the zone to be ground is of a length equal to the width of face of the wheel. Theoretically, the feed increment should occur once per

work revolution to maintain uniform action. The work revolves in a direction opposite to that of the wheel.

11 Considering the two intersecting circles, the wheel and work peripheries, the wheel travels at a much higher rate than the work. A point on the wheel periphery describes a path within the point of intersection. It is the form of this path traced upon the work itself in which we are interested, for the shape of the chip removed by a cutting point on the wheel is an area bounded by the work periphery and by two successive grain paths. (The third dimension, width of chip, is unimportant in so far as cutting action is concerned.) An analysis of the motion of the wheel relative to the work (consider the work not moving) shows that the wheel is rolling around the work surface but revolving at a higher rate than that required to accomplish pure rolling. The grain path, therefore, is in the nature of an epicycloid, but extended because of the slip between work and wheel surfaces.

12 The criterion of wheel wear is the maximum depth of the chip removed, this depth being measured at right angles to the path of the cutting grain. We will therefore develop an expression for this depth, the variables of which are controlling factors of grinding action, as follows:

N = r.p.m. of wheel
V = surface speed of wheel
u = work speed in ft. per min.
R = wheel radius in inches
r = work radius in inches
f = depth of cut, inches on work radius
μ = interval between grains in fractions of an inch.

Other symbols used are:

S = feed circle = $r - f$, inches
G = maximum depth of chip, inches.

13 The value G corresponds to Professor Alden's Grain Depth of Cut. The grain interval μ is the quotient of the number of cutting particles on the wheel periphery and the circumference of the wheel in inches (considering the wheel width equivalent to a single grain). This value may be determined for various wheels by actual count and measurement. It varies with both grain size and grade of wheel.

14 Refer to Fig. 1 in which greatly enlarged section of wheel and work is shown. The grain paths across the work are indicated and a shaded area indicates the shape of a single chip. The grain

paths are shown as straight lines tangent to the feed circle. Actually these paths are not straight lines but are the epicycloidal form of curve above mentioned. However, within the limits of probable dimensions which are actually involved in any cylindrical grinding operation, a graphical layout of the grain paths on a large scale (say $\frac{1}{4}$ in. = 0.0010 in.) will show that these curves are so nearly straight lines that the error is negligible. The mathematical relations are greatly simplified by this assumption. By actual trial it was found that there was seldom over 2 per cent difference in the values of maximum chip depth obtained by the use of the *actual* and the *assumed* forms of grain path.

15 Within the limits of reasonable working stresses, the wear

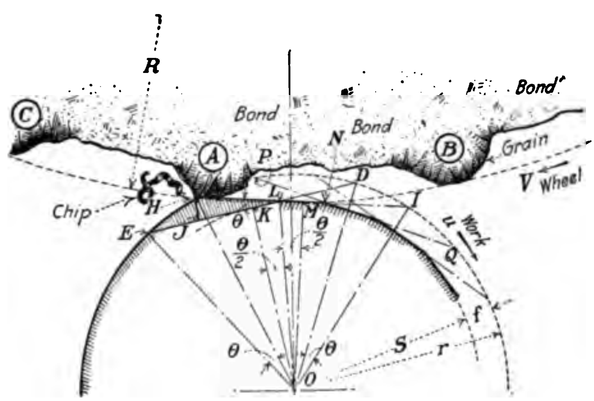

FIG. 1 DIAGRAM ILLUSTRATING CUTTING ACTION

of any cutting tool is roughly proportional to the total work done. The grain cuts until it is so dulled that impact tears it from its setting of bonding material or causes fracture, exposing fresh cutting points. The frequency with which the grain is applied, the average grain depth of cut as indicated by the chip dimensions, and the kinetic energy of the grain due to its mass and velocity, are the three factors which are to be considered as determining the wheel-wearing action. Frequency of application depends upon the revolutions per minute of the wheel. Work done is measured by the dimensions of the chip removed by each grain and the total number of chips removed. Kinetic energy is proportional to the mass and square of the velocity of the cutting particles.

16 Frequency of application of the grain $= N = \frac{12V}{2\pi R}$, and

volume of wheel wear $= \frac{12V}{2\pi R} \times 2\pi R$. Therefore wheel wear is directly proportional to $12V$; or dropping the constant, is directly proportional to V.

17 Wheel-wearing action is also proportional to grain depth of cut G, for it is this factor alone which controls the actual depth to which the grain penetrates, and therefore the rate at which the bond is worn away by mechanical rubbing against the work surface. In making this statement it is assumed that the grain and grade of the wheel are such that the true grinding action of the wheel is taking place, i.e., that the grain does not wear appreciably but dulls until the cutting becomes a rubbing and the grain is torn out from the setting of bonding material, which has been weakened by the wearing away due to contact with the work surface.

18 In Fig. 1 path QNP is generated by grain C; grain A follows at the arc distance μ and generates path ILH and is removing the chip of area PNH. The ground surface is made up of the successive paths and is not truly cylindrical, as small projections are left of the form KLM. These are immeasurably small. The grain B will generate DLE and cut chip EHL. The depth of this chip normal to the path DLE is JH, which we designate as G, grain depth of cut.

19 *Formula for Determining Depth of Cut.* To find an expression for G in terms of the variables already enumerated, we have:

HLI is tangent to circle of radius S at M; ($S = r - f$)

EJL is tangent to circle of radius S at K

HLI and ELD intersect and form angle θ

DI (arc) is the distance through which the work moves while the wheel moves through the distance μ between grains.

$$EH = \frac{\mu u}{V}$$

Other necessary nomenclature is contained in Par. 12.

Drop perpendiculars to ELD and HLI from K and M. Then

$$KO = MO = S = r - f$$

$$\text{Angle } KOM = \theta$$

$$\text{Angle } KOL = LOM = \frac{\theta}{2}$$

$$G = JH = HL \sin \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

Angle θ is very small and where expressed in radians is very nearly equal to its sine. Therefore

$$G = JH = HL \times \theta$$

But

$$\theta \text{ (in radians)} \times r = \text{arc } EH = \mu \frac{u}{V}$$

or

$$\theta = \frac{\mu u}{Vr} \quad \ldots\ldots\ldots\ldots \quad [2]$$

Therefore, from [1]

$$G = JH = HL\frac{\mu u}{Vr} \quad \ldots\ldots\ldots\ldots \quad [3]$$

In triangle OHM, $HO = r$ and $MO = S = r - f$, therefore

$$r^2 - (r-f)^2 = \overline{HM}^2$$

and

$$f\ (2r - f) = \overline{HM}^2$$

But $(2r - f) = 2r$ (very nearly), as f is rarely over one or two thousandths of an inch and is very small compared to $2r$. Therefore,

$$2rf = \overline{HM}^2 \qquad \text{and} \qquad HM = \sqrt{2rf} \quad \ldots\ldots \quad [4]$$

LM is very nearly equal to arc of radius $(r - f)$ of angle $\frac{\theta}{2}$. Assuming it to be equal, $LM = \frac{\theta}{2}(r - f)$.

As before, f may be neglected, due to its extremely small effect upon r. Then, from [2],

$$LM = \frac{\theta}{2}r = \frac{\mu u}{2V}$$

Since $HL = HM - LM$, we have from [4],

$$HL = \sqrt{2rf} - \frac{\mu u}{2V}$$

and from [3],

$$G = JH = \left[\sqrt{2rf} - \frac{\mu u}{2V}\right]\frac{\mu u}{Vr} \text{ (very nearly)} \quad \ldots\ldots \quad [5]$$

20 The above represents the grain depth of cut very closely, and is expressed in the terms which were previously chosen.

Formula [5] may be rewritten as

$$G = \frac{\mu u}{V}\sqrt{\frac{2f}{r}} - \frac{\mu^2 u^2}{2V^2 r}$$

By inspection, bearing in mind that μ is always small, say 0.05 in., u is seldom over 200, f is seldom over 0.001 inch and V usually about 6000, and r, say 1 in. to 4 in., we are justified in further reducing the expression mathematically by dropping the term $\frac{\mu^2 u^2}{2V^2 r}$ as being insignificant compared with the other two terms. Therefore

$$G = \frac{\mu u}{V}\sqrt{\frac{2f}{r}} \text{ (approximately)} \quad \ldots\ldots \quad [6]$$

21 This enables us to deduce approximate statements as follows: Grain depth of cut varies directly with grain interval, work speed, and square root of radial feed; inversely with wheel speed, and square root of work radius.

22 The relationship between work speed and feed is as determined by Mr. Guest in his Controlling Factor. The value of G also tallies with Professor Alden's results within the usual speed and dimensional limits found in cylindrical grinding.

23 *Wheel-Wearing Action.* The kinetic energy of the cutting particles increases with the square of the velocity.[1] For a certain depth of cut the resistance of the work to the grain is very nearly proportional to this depth, as the cuts are very small, even when compared to the total size of the grain. Theoretically, for some low velocity the kinetic energy of the grain should just equal the energy required to remove the chip (the feed would be reduced to maintain constant grain depth of cut under reduced wheel-velocity conditions) and wheel wear would be caused without any chip being removed up to this point. Actually the bond strength is always sufficient to prevent this condition. Our assumption is then clear:

24 Wheel wear varies inversely with the *surplus* kinetic energy of cutting particles and is inversely proportional to *effective* wheel surface velocities (wheel speed minus work speed).

25 We are now prepared to combine our factors.

a Wheel wear is proportional to the wheel speed, or to V.

b Wheel wear is proportional to the grain depth of cut, as expressed in Formula [6].

c Wheel wear is inversely proportional to surplus kinetic energy of the cutting particles, or proportional to $1/(V - u)^2$. The term u is usually negligible as compared with V and it may therefore be assumed that $1/V^2 = 1/(V - u)^2$.

26 From these relations, *a*, *b* and *c*, an index of wheel-wearing action (WW_I) due to grinding (where the wheel face is free-cutting) may be expressed as follows:

[1] See Selection of Grinding Wheels for the Foundry, *Grits and Grinds*, Jan., 1915, The Norton Company, Worcester, Mass.

$$WW_I = V\frac{\mu u}{V}\sqrt{\frac{2f}{r}} \times \frac{1}{V^2}$$

$$= \frac{\mu u}{V^2}\sqrt{\frac{2f}{r}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [7]$$

LAWS OF GRINDING FOR DRAW-IN CUTS

27 From this expression we may derive the following laws of cylindrical grinding, operative *within the usual limits of speeds and dimensions*. For straight-in (draw-in) cuts a free-cutting wheel will wear according to the effect of wheel-wearing action as outlined below:

a Wheel-wearing action increases directly as:

1 Grain interval (grain size, — inversely as grade)
2 Work speed (surface)
3 Square root of diametral cut (feed).

b Wheel-wearing action increases directly as the following are decreased:

1 Square of wheel speed (surface)
2 Square root of work radius.

28 A soft wheel is more susceptible to wheel-wearing action than a hard wheel and will more nearly follow the theoretical conditions. It is therefore more free-cutting and will cut more nearly the full chip than a hard wheel. There is consequently a gain in production as wheels of softer grades are used, all other conditions remaining constant, up to the point where the chips are geometrically perfect for a given condition of speeds and dimensions. It is clear from this standpoint that the softest possible wheel should be used. High wheel speeds up to the safe limit are of course necessary if the soft wheels are to perform in a satisfactory manner.

29 To show the practical application of Formula [7] for wheel-wearing action (Index of Wheel Wear), derived above, Table 1 is given.

30 In column 1 of the table are the work speeds provided on a recent grinding machine, given in order so that actual values can be experimentally obtained to check with the theoretical.

31 In the succeeding columns are two series of values as follows:

(1) Following the term "Production" are *approximate* values for cubic inches of metal removed per minute for a wheel

1 in. wide. These were calculated by the formula, Production = nominal work diameter $\times \pi \times$ r.p.m. $\times$ depth of cut (assumed to be 0.0005 in.), which gives values slightly less than the true values.

(2) Following the terms "Wheel Wear" are values of the *actual* wheel wear obtained by multiplying the Index of Wheel Wear, calculated by Formula [7], by a constant derived by experiment. This constant is for convenience expressed in the form $1800 \times 10.^4$

32 *Sample Calculation.* In Formula [7] assume:

u = 53 r.p.m. = 13.9 ft. per min.
r = $\frac{1}{2}$ in. (work diameter 1 in.)
f = 0.0005 in.
V = 6000
μ = 0.05
R = 9.0 in. (wheel radius)

Substituting,

$$WW_I = \frac{0.05 \times 13.9}{36{,}000{,}000}\sqrt{\frac{0.001}{0.5}}$$
$$= 0.0000000196 \times 0.047 = 9212 \times 10^{-13}$$

Multiplying by constant 1800×10^4 (see Par. 31),

$$WW_I = 0.0166 \text{ cu. in. per min. for wheel 1 in. wide.}$$

The corresponding "Production" in Table 1 is, 0.0835 cu. in. (see Par. 31).

33 It will be seen that the wheel-wear values increase rapidly as work speed increases. The value 0.0166 for work of 1 in. diameter at 53 r.p.m. becomes 0.1040 for work of 4 in. diameter at 167 r.p.m. The same wheel cannot be properly used for both cases, as one sufficiently soft to cut freely in the first case would wear so fast as to be nearly useless in the second case. However, where wheel-wear values are nearly the same we may expect the same wheel to act properly. For instance, a wheel suitable for 1-in. work at 138 r.p.m. wears at the rate of 0.0433 cu. in. per min.; and one suitable for 4-in. work at 72.5 r.p.m. wears at the rate of 0.0450 cu. in. per min. In all probability one wheel would serve for both.

SELECTION OF WHEELS FOR DRAW-IN CUTS

34 From Table 1, and by experimental tests with various wheels, certain ranges of wheel-wear values may be found which correspond

to definite grades of wheels of suitable grain size and abrasive for the material to be ground. Once this is done, the selection of proper wheels is no longer a question of guesswork, as wheel-wear indices may be computed for any set of conditions and the proper wheel selected. The question of the variation in wheels has been reduced to a point where it no longer can be a very large factor, and we may always change our speeds or feeds to correct the action in the proper direction if we understand the laws which have been previously

TABLE 1 COMPARISON BETWEEN WHEEL WEAR AND QUANTITY OF METAL REMOVED AT DIFFERENT WORK SPEEDS

Work Speed in r.p.m.		Work Diameters			
		1 in.	2 in.	3 in.	4 in.
53	Production	.0835	.1670	.2500	.3340
	Wheel Wear	.0166	.0236	.0290	.0334
72.5	Production	.1140	.2280	.3420	.4560
	Wheel Wear	.0226	.0319	.0390	.0450
87	Production	.1370	.2740	.4110	.5480
	Wheel Wear	.0273	.0387	.0475	.0546
101	Production	.1586	.3172	.4758	.6344
	Wheel Wear	.0316	.0448	.0540	.0635
138	Production	.2160	.4320	.6480	.8640
	Wheel Wear	.0433	.0612	.0645	.0865
167	Production	.2620	.5240	.7860	1.0480
	Wheel Wear	.0522	.0740	.0910	.1040

Wheel diameter 18 in.; surface speed, 6000 ft. per min.; diametral feed per work revolution (automatic feed), 0.001 in.; grain of wheel, 6646 Alundum; material, mild machinery steel.

pointed out. As an example of the possibilities of wheel selection, it has been found that for a certain machine the following wheels (Norton system of grade) are suitable for straight-in cuts on mild steel:

Wheel Designation		Wheel-Wear Values, Table 1
36 or 46 J	for values from	.0150 to .0250
36 or 46 K	for values from	.0250 to .0350
36 or 46 L	for values from	.0350 to .0450
36 or 46 M	for values of	.0450 and higher

Use no wheels harder than M. For hard materials use a grade softer and No. 60 grain. These values are for straight-in cuts only.

35 *Influence of Feed.* As an example of the effect of doubling the feed, let us assume the conditions as above, except that the feed is increased to 0.001 in. (on radius).

$$WW_I = \frac{\mu u}{V^2}\sqrt{\frac{2f}{r}} = \frac{0.05 \times 13.9}{36{,}000{,}000}\sqrt{\frac{0.002}{0.5}}$$
$$= 1240 \times 10^{-13}$$

Wheel wear $= 1800 \times 10^4 \times 1240 \times 10^{-13} = 0.0224$ cu. in. per min. per inch of wheel face. Production is approximately doubled and

$$\text{Wheel-wear increase} = \frac{0.0224 - 0.0166}{0.0166} \times 100 = 35 \text{ per cent.}$$

36 This indicates how advantageously an increase of feed operates to obtain increased production (up to the limit of grain penetration). If we had doubled work speed we would simply have doubled both production and wheel wear.

TRAVERSED CUTS

37 The most important development in recent grinding machines is the recognition and utilization of the fact that overall operating efficiency for traversed cylindrical work *increases* as *traverse speeds are increased.* (This assumes that for a given traverse speed, width of wheel and revolution of work are so related as to cause the cutting face to just cover the lead of the work.)

38 This fact is based upon certain scientific laws. It will now be shown why this is true and an expression will be derived for determining the effect of traverse speeds. Professor Alden did not go into this question, and Mr. Guest apparently did not recognize the true conditions as he considers traverse speed of negligible effect.[1]

39 It is evident what grain depth of cut is, and how it affects wheel wear. The wheel width of cut also affects wheel wear. If we consider, for instance, the resistance offered to forward motion of a snow plow, we know that if the plow is set at an angle it is easier to push, although we will have to push it further to clean off the same area of surface than when it is set exactly against the direction of the path.

40 Consider the grinding wheel as stationary, non-rotating, and the work to revolve. If the work is traversed, the path will be

[1] See Guest's Grinding Machinery, Arnold, 1915 edition, p. 80.

a helical ribbon. The width of this ribbon measured parallel to the axis of rotation of the work is the same as the cylindrical path cut by the wheel when the work is not traversed. For the same *angular* movement of the work, both paths are of equal area. The work done is the same. In the case of the traversed work, however, the resultant motion between wheel and work is at an angle to the axis of rotation. The tangent of this angle is the length of the projected arc, or circumferential distance traveled by the work, divided by the distance traversed in the same time. The actual length of the path measured in the direction of this relative motion is greater than the length of the projected arc, while the width of path perpendicular to this length is less than the width of the straight-in cut. The length of cut becomes equal to the length of equivalent straight-in cut divided by the sine of the traverse angle or lead angle of the work. The width of path is diminished in the same proportion, being the width of the straight-in cut times the sine of the traverse angle whose tangent was given above. With the wheel still at rest, we see that we have reduced the wheel width of cut by the proportion of the sine of the traverse angle. Referring to Fig. 2, let

W = width of wheel face
L = length of projected arc of rotation of work
M = actual length of path of traversed cut
B = actual width of path of traversed cut
β = traverse angle

Then $B = W \sin \beta$; $M = L/\sin \beta$; areas $W \times L$ and $B \times M$ are equal. Wheel width of cut is proportional to W for straight-in cut, but reduced to B for traversed cut.

41 It is to be noted that relative work speed has been increased over actual work speed in the inverse proportion of the sine of the traverse angle. This relative increase does not appear in the projected (end view) relations. As these alone affect individual grain depth of cut, we have gained in relative work speed to get over the longer path M, so as to cover the same area in the same time on a narrower width B, *without* increasing grain depth of cut.

42 Now if the wheel rotates, each grain cuts across the work at an angle θ, resulting from speed relations shown by Fig. 2 (vector diagram for rotating wheel). Then the individual grain paths are at a much steeper angle than the wheel path to the axis of work rotation, but we have gained in relative wheel speed *without* increasing individual grain depth of cut. Our wheel width of cut is also reduced

by a slight amount proportional to sin θ, which is of course much less than the reduction before the wheel was rotated. The net result is a reduction in wheel stress proportional to sin θ and an increase in work speed proportional to sin β, *without* increase in grain depth of cut.

43 The "head resistance" against the wheel as a whole is reduced by the sin β factor, while wheel width of cut is reduced by sin θ. These both directly affect wheel wear. For traversed-feed conditions our grain depth of cut remains the same as in straight-in feeds, but wheel wear is reduced in the proportion of sin $\beta \times$ sin θ

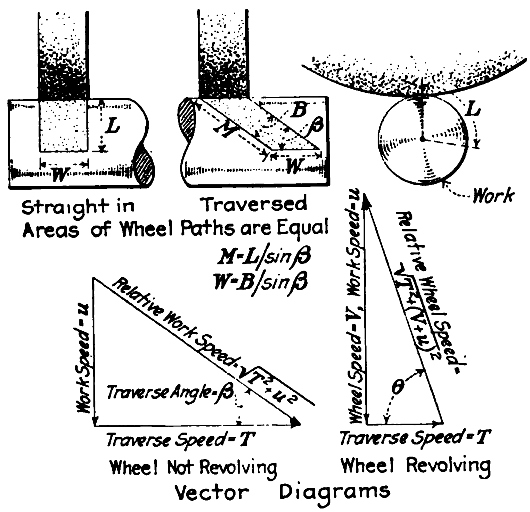

FIG. 2 SPEED RELATIONS BETWEEN WORK AND WHEEL

44 If T = traverse speed, V = wheel speed, u = work speed, then

$$WW_I = \left(\frac{u}{\sqrt{T^2 + u^2}}\right)\left(\frac{V + u}{\sqrt{T^2 + (V + u)^2}}\right)\left(\frac{\mu u}{rV^2}\right)\left(\sqrt{2rf}\right) \quad . \; . \; [8]$$

In practical cases the second term will be so nearly 1 that it may be neglected. It is evident that T affects the quantity appreciably only when u is kept relatively low. This indicates that a wide wheel should be used to get the effect of reduced wear for a given production, traversing as rapidly as possible. The above formula assumes no overlapping of the cut.

45 Now by making assumptions and neglecting values whose effect is so small as to make no appreciable difference for purposes of

comparison, we reduce the expressions for WW_I exactly as we did in the case of straight-in feeds and obtain:

$$\text{Index of wheel wear} \quad WW_I = \frac{\mu u}{V^2}\left(\frac{u}{\sqrt{T^2 + u^2}}\right)\sqrt{\frac{2f}{r}} \quad . \; . \; . \; [9]$$

LAWS OF GRINDING FOR TRAVERSED CUTS

46 From this it is evident that for traversed work:

a Wheel wear increases directly with the work speed

b Wheel wear decreases directly as the decrease in the quantity which we term "traverse factor," which is work speed divided by the square root of the sum of the squares of traverse and work speeds

c Wheel wear decreases directly as square of wheel speed increases

d Wheel wear decreases directly as the square root of the work radius increases

e Wheel wear increases directly as the square root of the feed increases.

47 The value $\frac{u}{\sqrt{T^2 + u^2}}$ in our expression for index of wheel wear is our traverse factor. Where T is large with respect to u this factor will be appreciably less than unity, but as u increases the reduction in wheel-wearing action is less pronounced.

48 Table 2 gives values of the traverse factor for a certain machine having the work speeds previously given for draw-in cuts, and the following table (traverse) speeds: 10.02, 14.00, 17.00, 19.70, 26.90 and 32.60 ft. per min. The table shows combinations of speeds giving leads from 2.3 in. upward. As the machine was designed for a wheel 2½ in. wide, combinations of speeds giving lesser leads of work are not given.

49 If we now multiply each of the traverse factors in Table 2 by the wheel-wear value for the corresponding work speed and work diameter as shown in Table 1 (draw-in cuts), we will have a table of wheel-wear values for traversed cuts. These products are given in Table 3.

50 By assigning certain wheels to specified ranges of wheel-wearing action, we may select the proper wheel through the use of Table 3 in the same way that we did from Table 1. Actually in the case of Table 3 the wheels used would be somewhat softer for the same index figure because the actual depth of cut does not follow

the indicated depth (as shown by the feeding mechanism) as closely for traversed work as for draw-in work, and therefore the actual cut is not so deep. This means, of course, that the "spring away" action of the work is more pronounced for traversed than for draw-in work. Careful use of steady rests is therefore very essential.

TABLE 2 TRAVERSE FACTORS

Traverse, ft. per min.	Work, r.p.m.	Lead, in.	Work Diameters			
			1 in.	2 in.	3 in.	4 in.
10.02	53	2.3	.810	.950	.975	.985
14.00	53	3.1	.705	.883	.946	.968
14.00	72.5	2.3	.805	.938	.968	.976
17.00	53	3.8	.632	.850	.926	.955
17.00	72.5	2.8	.730	.910	.955	.974
17.00	87	2.3	.802	.935	.972	.985
19.70	53	4.4	.576	.817	.905	.943
19.70	72.5	3.2	.691	.884	.943	.967
19.70	87	2.7	.756	.916	.961	.980
19.70	101	2.3	.802	.935	.971	.986
26.90	53	5.9	.460	.720	.839	.900
26.90	72.5	4.4	.574	.815	.902	.942
26.90	87	3.6	.646	.860	.931	.958
26.90	101	3.1	.703	.890	.950	.974
26.90	138	2.3	.801	.936	.960	.982
32.60	53	7.35	.397	.655	.790	.865
32.60	72.5	5.3	.505	.760	.867	.919
32.60	87	4.4	.580	.815	.908	.945
32.60	101	3.8	.635	.854	.927	.960
32.60	138	2.8	.748	.914	.945	.977
32.60	167	2.3	.807	.936	.967	.982

51 The following wheels are found to be satisfactory for our machine for traversed cuts on mild steel under the assumed conditions:

Wheel Designation		Wheel Wear Values Table 3
36 or 46 I	for values up to	.02500
36 or 46 J	for values from	.02500 to .04000
36 or 46 K	for values from	.04000 to .05000
36 or 46 L	for values from	.05000 to .06000
36 or 46 M	for values from	.06000

Use No. 60 grain for hard material and a grade softer; but use no wheel softer than I.

52 *Important Note.* These statements of wheel assignment are based upon assumptions of the best machine conditions and most

careful operation to prevent any abuse of the wheel. Field practice will usually show harder wheels in use due to conditions which abuse the wheel. Efficiency is sacrificed where such is the case.

PRODUCTION COSTS

53 Grinding efficiency is usually considered as (production) ÷ (wheel wear). It is expressed as cubic inches of material removed per cubic inch of wheel wear. If wheel wear were the most important

TABLE 3 WHEEL-WEAR VALUES FOR TRAVERSED CUTS

Traverse, ft. per min.	Work, r.p.m.	Lead, in.	Work Diameters			
			1 in.	2 in.	3 in.	4 in.
10.02	53	2.3	.01340	.02240	.02820	.03280
14.00	53	3.1	.01170	.02080	.02740	.03230
14.00	72.5	2.3	.01820	.02450	.03775	.04390
17.00	53	3.8	.01050	.02005	.02685	.03185
17.00	72.5	2.8	.01650	.02900	.03720	.04370
17.00	87	2.3	.02190	.03620	.04620	.05390
19.7	53	4.4	.00955	.01930	.02620	.03142
19.7	72.5	3.2	.01560	.02820	.03675	.04350
19.7	87	2.7	.02060	.03540	.04570	.05355
19.7	101	2.3	.02540	.04185	.05245	.06260
26.9	53	5.9	.00763	.01695	.02430	.03000
26.9	72.5	4.4	.01290	.02600	.03520	.04240
26.9	87	3.6	.01760	.03320	.04420	.05235
26.9	101	3.1	.02220	.03980	.05125	.06185
26.9	138	2.3	.03470	.05130	.06195	.08500
32.6	53	7.35	.00660	.01545	.02290	.02885
32.6	72.5	5.3	.01140	.02120	.03380	.04130
32.6	87	4.4	.01580	.03150	.04315	.05160
32.6	101	3.8	.02000	.03820	.05000	.06100
32.6	138	2.8	.03230	.05580	.06095	.08450
32.6	167	2.3	.04210	.06940	.08800	.10230

element of production cost, grinding efficiency might be considered on this basis without leading to the fallacy which exists today. As a matter of fact, the wheel cost is almost negligible compared to the other costs, and wheel selection should be based upon the production capacity of the wheel under the given set of grinding conditions, allowing the wheel wear to be as high as is necessary to get a free action without excessive wear (which would cause difficulty in sizing work and the need for frequent dressing). The all-important factor is the rate at which the wheel may be made to cut and still not get out of truth. This affects the grinding time, any reduction of which is of vastly more importance than an increase of wheel wear which may result from such a reduction.

54 In good practice the volume of wheel wear is approximately a tenth of the material removed (mild steel) during roughing operations. A conservative estimate of the cost of operating a modern cylindrical grinding machine (exclusive of wheel cost) is, say, five cents per minute, including labor at day-rate wages.[1] The useful content of the wheel may be assumed to be such that the wheel is actually worth seven cents per cubic inch. If production could be increased to 2 cu. in. per min. where previously we obtained $1\frac{1}{2}$ cu. in. and wheel wear jumped from 0.15 cu. in., say, to 0.25 cu. in. per minute, our first cost would be: $(0.05 + 0.15 \times 0.07) \div 1.5 = \$0.0404$ or 4.04 cents per cu. in. removed. In the second case, $(0.05 + 0.25 \times 0.07) \div 2 = \$0.0338$, or 3.38 cents per cu. in. removed.

55 Now the overall production efficiency of the machine based upon all costs involved is indicated by production-cost factors proportional to the earning rate of the machine. In the first case this factor would be (production) $\div$ (cost) $= \dfrac{1.5}{0.05 + 0.15 \times 0.07} = 24.8.$ In the second case, $\dfrac{2.0}{0.05 + 0.25 \times 0.07} = 29.6.$

56 It evidently will pay to wear the wheel more in order to obtain the higher rate as the overall gain in the foregoing is about 19 per cent. The above assumed values are reasonable and the figures given might be considered as typical. Table 4 gives these production-cost factors, or relative earning rates based upon these costs and the grinding conditions previously assumed in Tables 1, 2 and 3, the variations being due to traverse and work-speed relations only, but with proper choice of wheels in each case. Then, for the machine which we have used as an illustration it is evident that for work of a certain size and a wheel of a certain width there is one speed combination which will utilize the machine to the greatest profit. If we are able to use wheels of the width needed to cover the lead under the best speed-relation conditions, we will obtain the best earning rate of which the machine is capable. If wide wheels are not available, we still have speed-relation conditions which will give very nearly as high a rate for the narrower wheels.

57 In Table 4 the production-cost factors are given for 1-in.

[1] The figure of 5 cents a minute as time cost of operation is based upon labor at 75 cents an hour and an hourly burden of $2.25 an hour (power, invest ment, rent, heat, light, etc.).

and 4-in. diameters of work only. The factors may also be expressed as earnings per unit of time, or the relative amounts the machine would earn in equal lengths of time under the various conditions as given.

58 Note that, due to high traverse, a 2.3-in. wheel which earns $3.68 in a certain length of time when grinding 1-in. work at low traverse and work speeds, will earn $10.62 in the same length of time when using the high speeds. If the widest wheel (7.35-in. face) be used the machine will earn $11.70 in the same length of time. The grade of wheel is suitable in each case according to the

TABLE 4 PRODUCTION-COST FACTORS

Table Speed, ft. per min.	Work, r.p.m.	Lead, in.	Diameter of Work	
			1 in.	4 in.
10.02	53	2.3	3.68	13.45
14.00	53	3.1	5.16	18.80
14.00	72.5	2.3	4.95	17.85
17.00	53	3.8	6.00	21.00
17.00	72.5	2.8	6.00	21.80
17.00	87	2.3	5.85	22.00
19.70	53	4.4	6.40	23.70
19.70	72.5	3.2	6.80	23.50
19.70	87	2.7	6.84	23.45
19.70	101	2.3	6.75	23.30
26.90	53	5.9	9.05	29.80
26.90	72.5	4.4	9.34	29.85
26.90	87	3.6	9.00	29.00
26.90	101	3.1	9.03	29.10
26.90	138	2.3	9.00	29.15
32.60	53	7.35	11.70	35.70
32.60	72.5	5.3	11.30	34.60
32.60	87	4.4	11.00	33.60
32.60	101	3.8	10.90	33.40
32.60	138	2.8	10.80	33.30
32.60	167	2.3	10.62	32.90

speed changes. This clearly shows that high traverse speeds cause a corresponding increase in earnings in spite of the necessary increase in work speed.

59 Considering the cost of wheels 7.35 in. wide as compared to wheels 2.3 in. wide, and the relative power required to drive the two wheels, it is very evident that the use of a harder wheel 2.3 in. wide would offset the difference in wearing action without greatly reducing production and that the production of the 7.35-in. wheel is matched by the production of the 2.3-in. wheel using highest traverse and highest work speed.

60 The most striking conclusion from the above is the fact

that in order to use a narrow wheel and get great production the high work speed does not cause excessive wheel wear *due to the relief* on the wheel brought about by the *high traverse speed.* This traverse speed is also necessary to allow the rapid exposure of fresh work surface to the cutting face, making for a highly economical combination.

61 The foregoing should demonstrate clearly what this new development in the science of grinding may mean to the art as practiced throughout our production shops.

62 Actual grinding tests upon a machine having the high-traverse feature, and making use of the speeds shown in the table, have conclusively proven the deductions indicated by the mathematical calculations. On mild machine steel (0.15 per cent carbon) over diameters ranging from $\frac{3}{4}$ in. to 4 in., production rates as high as 2.00 cu. in. per min. have been easily attained, grinding to limits of 0.0010 in., removing 0.060 in. from the diameter over lengths of from 36 to 72 in.

63 When machine conditions, truth and balance of wheel, and proper use of steady rests and grinding compound are intelligently and skilfully handled, wheels of 36 and 46 grain size, and grades I to L (Norton system of grading) have been used in these roughing operations. Especially where the feeding is carefully regulated so as *never* to *crowd* the *radial depth* of cut to more than the possible depth of grain penetration (say not to exceed 0.0015 in. on the diameter) per traverse, it has been found that the wheel face remained sufficiently true to leave a commercial finish of fair degree absolutely free from chatter marks, feed lines or any other imperfections of real or imaginary consequence.

64 Owing to the fact that in the above the highest work and traverse speeds were used in every case, the grade (hardness) of wheel was increased as the diameter of the work increased. This is in accordance with our formula for index of wheel wear. Instead of considering the work speed and work radius independently, they were in this case tied together by the fact that a constant work r.p.m. was used. The expression for this is as follows:

$$\text{Work speed } u = \frac{2\pi r N}{12} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [10]$$

where r = work radius (inches) and N = r.p.m.

65 Formula [8] for traversed cuts, with second and third terms omitted as unimportant, becomes

$$WW_I = \frac{\mu u}{V^2}\left(\frac{u}{\sqrt{T^2 + u^2}}\right)\sqrt{\frac{2f}{r}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

Substituting [10] in [9],

$$WW_I = \frac{0.275\mu}{V^2}\left(\frac{N^2}{\sqrt{T^2 + 0.275r^2N^2}}\right)\sqrt{2fr^3} \quad . \; . \; . \quad [11]$$

66 Now if V, N, T and f are made constant, WW_I must vary as the value r appears in the formula, the greatest effect being the $\sqrt{r^3}$ factor and we would expect therefore to use harder wheels as r increases.

CONCLUSIONS

67 The conclusions to be drawn from the foregoing are definite and indicate positively how to arrive at the solution of precision grinding problems, provided one has an intelligent knowledge of abrasive wheels and the usual materials ground.

68 *Theoretical Conclusions* (*Cylindrical Grinding*). Greatest grinding efficiency is obtained by the use of the *softest wheels* suited to the nature of the material ground. This efficiency is dependent upon the control of the dimension and speed relations between the wheel and the work so that the individual chip may have the minimum depth for a given volume determined by the maximum allowable radial depth of cut. This means long arc of contact, low work speeds and maximum feeds.

69 With the above conditions established, *increase of traverse speed increases production without increase* of *wheel-wearing action.*

70 *Practical Conclusions.* Machine conditions must be such as to maintain as accurately as it is reasonably possible the speed and dimensional relations of the wheel and the work. This includes a great number of individual factors, any one of which may serve to entirely or partially impair the successful operation of the machine in following the theoretical fundamental laws as previously stated. It is entirely worth while to emphasize here some of the most important of these.

a The power drive must not allow speed variations unless under the wilful control of the operator

b The wheel must be in good running balance and in absolute truth, and must be held in its position relative to the work within the closest possible limits

c The work must be accurately held with respect to the wheel and must be *uniformly* rotated. Relative traverse between work and wheel must be uniform *d* The work must be rigidly supported over its entire length and no vibration allowed to occur between centers

e Feed control must be sensitive and accurate and feeding must be at a rate such that the feed increment *never exceeds the maximum grain penetration.* This is the most frequently violated of the factors involved

f Work must be kept at a uniform temperature and local heating prevented at all times. A copious supply of grinding compound should be directed to the *arc of contact* at all times. Eccentric work due to bowing from heat effects is the usual result of failure to supply sufficient compound at the arc of contact. In truing the wheel with a diamond, the use of the cooling medium is vital to accuracy. Save the diamond by using a dresser to roughly true a wheel, using the diamond only when necessary to put the wheel in exact truth for precision work. Never feed a diamond over 0.001 in. per traverse

g Select wheels intelligently, and do not try to use a single wheel for a variety of work sizes or materials unless the job is too short to warrant efficient grinding as compared with the time of changing wheels. It is seldom that this is true. Hard work, (hardened steel, manganese steel, stellite, etc.) requires soft, rather fine wheels. The fine grain reduces the chip size but removes more chips, and wheel wear is not so likely to cause a pounding of the wheel, nor will the small grain cause the glazing which is bound to occur with coarse, hard wheels

h An accurate finish requires the use of a soft, free-cutting wheel so controlled that the chips are very small (light feed and slow work speed with traverse to make the wheel face just cover the lead)

j A burnished finish may be obtained by a peening action of a coarse, hard wheel trued dead smooth. Heating and inaccuracies of surface are likely to occur

k Cases are rare where a wheel harder than grade M may be properly used in cylindrical grinding. Low wheel speeds, too small a wheel, too high a work speed and abusive feeding — these are usually the causes for the use of hard wheels. Mention should also be made of failure to take draw-in cuts to establish traverse limits, thus requiring too hard a wheel in order to make the "corner stand up."

Along this line may properly be mentioned design and drafting-room practice in calling for sharp shoulders where a generous fillet might be allowed. This makes needless difficulty in grinding and weakens the piece ground

l The contact of the wheel face with dry work at any time immediately ruins its value for finishing. The common practice of just touching the dry work when bringing up the wheel to contact is wrong, due to the charging of the wheel face with the uncooled chips (loading).

71 When the machine is properly designed and operated the grinding action will closely follow the theoretical laws. These laws are always at work, and when there is apparent conflict between the theoretical and the actual, there exists some fault in the machine or with its operator. There is an enormous field for the industrial engineer in obtaining better production at less cost in grinding operations. There is a definite science involved, there is no mystery about it. The proper appreciation of the laws of grinding by machine designers and operators is rapidly increasing and we may expect a constant improvement in the art as a consequence.

DISCUSSION

Henry J. Eberhardt. Is it not true that the geometrical rules for grinding and milling are the same? Also, will not the author point out the main difference between milling and grinding insofar as his conclusions are concerned? The question is asked because apparently there will be a gain in efficiency, and in the length of life of the cutter in proportion to the production by increasing the feed and leaving the speed of cutting at the normal rate.

The Author replied that the grinding wheel was, for purposes of studying the geometric relations, the same as the milling cutter in action. The dimensions and velocities are radically different, but the laws which control milling cuts must also control grinding cuts, if it is assumed that the relation of the cutting tool to the work is always kept within reasonably rigid limits.

Charles H. Norton. Back of the question of the first speaker is, perhaps, the thought of the relation in production between grinding and milling. If there is a small amount of metal to be removed, it can be removed faster with a grinding wheel than with a milling cutter. If roughing cuts on a forging are made with the milling cutter, precision will be obtained more quickly with a grinding wheel than with the milling cutter. Irregular forms and brittle pieces, like a camshaft or a crankshaft or something difficult to hold rigidly enough to use a milling cutter or turning tool, can be shaped from the rough faster with the grinding wheel than with the milling cutter because the grinding wheel removes more metal in a given time. A shaft a mile long and half an inch in diameter could not be turned perfectly round its entire length, but it could be ground to size within a quarter of a thousandth of an inch.

A grinding wheel cannot cut efficiently any deeper than the projection of its teeth beyond their setting will accommodate. Any attempt to force it beyond this means inefficiency. The more of these teeth brought into contact with the surface per minute, the more work will be done. This is accomplished by rapid revolution, a broad wheel and rapid traverses. When 3 cubic inches of steel are removed per minute, work is being accomplished at a rapid rate, and when 20 or 30 cubic inches of steel are removed for every cubic inch of wheel wear, it is being accomplished efficiently.

THE AUTHOR. In addition to what has been presented in the paper is a point which has since been discovered. It is a mathematical relation. In traversed work, where the work speed of revolution is chosen to make the wheel face just cover the lead of the work, so that the width of the wheel and the work speed are tied together in intimate relation, the production is directly proportional to the traverse speed. If in each traverse only very thin chips can be removed, the traverse speed determines the rate at which removal takes place.

Also, as has been pointed out, the depth of the cut is limited for each of the chips, and the thickness of the chip is generally proportional to the wear of the wheel. Therefore, the greater number of traverses made from end to end in a minute, the greater and more efficient the production. The mathematical relation would be

$$24\pi r f T = P$$

where

r = nominal work radius, inches
f = radial depth of feed, inches
T = traverse speed in feet per minute
P = production in cubic inches per minute.

Wheel wearing action is proportional to

$$WW_I = \frac{\mu u}{V^2}\left(\frac{u}{\sqrt{T^2 + u^2}}\right)\sqrt{\frac{2f}{r}} \quad \left(\text{see Eq. [9]}\right)$$

Then any increase of T increases production and decreases wheel wearing action at the same time.

No. 1765

CALIBRATION OF NOZZLES FOR THE MEASUREMENT OF AIR FLOWING INTO A VACUUM

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The following paper is a discussion of the flow of an elastic fluid through nozzles having well-rounded entrances. The effects of frictional resistance and of moisture in the atmosphere when the fluid is atmospheric air are presented and it is shown that the rate of flow of dry air decreases with increase of moisture. This result is based upon a series of tests on metal nozzles. The paper also includes a description of the nozzles, the method of calibrating them, and the results obtained. Formulæ employed in the calculations are also derived and the theory involved is outlined.

DURING some tests of air pumps, condensers, etc., conducted at the U. S. Naval Engineering Experiment Station, Annapolis, Md., it was desired to measure quantities of air by means of nozzles, the air flowing from the atmosphere through the nozzle into a vacuum. The problem was therefore to make a number of nozzles having orifices of suitable areas and to calibrate them, rather than to investigate correction factors for nozzles of different shapes. The following paper is based upon a report submitted on this work, and includes not only a description of the nozzles constructed, the method of calibration and the results obtained, but also a brief outline of the theory involved and the formulæ and calculations to be used.

DESCRIPTION OF THE NOZZLES

2 Twenty monel-metal nozzle plugs were made of the general shape shown in Fig. 1-A. Monel metal was selected as it is practically non-corrodible and takes a high polish. The plugs were threaded with a 1-in. pipe thread, a standard ring thread gage being used to secure interchangeability. It was intended to use these plugs in a

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soft brass bushing, as shown in Fig. 1-B, in order to preserve the threads as long as possible. The threaded joint between plug and bushing was made tight with asphaltum.

3 Each plug was $1\frac{1}{4}$ in. long, and the 1-in. pipe thread permitted the use of a maximum diameter of 1 in. for the orifice through the plug. An entrance curve to the orifice was selected as shown in Fig. 1-C, which is constructed of two arcs and is approximately elliptical. This entrance curve takes up three-tenths of the length of the nozzle, leaving in the case of the 1-in. orifice a straight portion seven-tenths of the nozzle length. The cross-sections of all nozzles having orifices less than 1 in. in diameter are made geo-

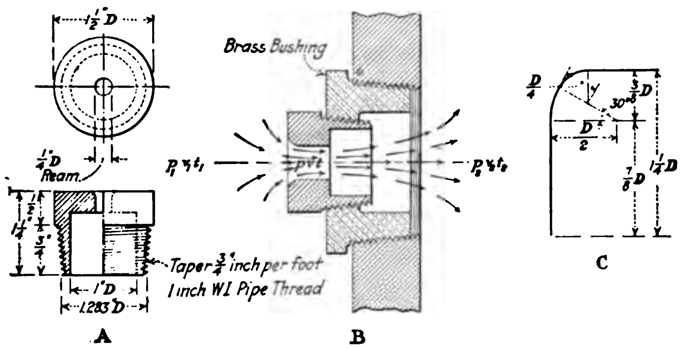

FIG. 1 DETAILS OF NOZZLE PLUGS

A, Cross-section of nozzle; B, Method of inserting nozzle in tank; C, Construction of entrance curve.

metrically similar to the 1-in. nozzle by counterboring the plug. In all nozzles, therefore, the length of the straight portion is seven-eighths its diameter, and the length of the entrance portion is three-eighths its diameter.

4 The diameters of the orifices were selected to correspond to standard drill sizes and to give convenient increments in area as given in Table 1. Except for the sizes below $\frac{1}{4}$ in., they were drilled $\frac{1}{32}$ in. smaller in diameter and reamed to size. The entrance surface was rubbed down with emery cloth to templet, giving the nozzle a highly polished appearance.

THEORY OF AIR FLOW THROUGH NOZZLES

5 Referring to Fig. 1-B, showing a fluid flowing through a nozzle, let the static pressure of the fluid in pounds per square foot

absolute be denoted by p, its specific volume in cubic feet per pound by v, and its temperature in degrees fahrenheit by t. Let the initial condition of the fluid be denoted by letters with the subscript 1, its final condition by letters with the subscript 2, and its condition within the orifice by letters without subscripts.

6 With a well-rounded entrance before the orifice, there will be no contraction of the jet and its minimum area will equal the area of the orifice; consequently,

$$G = \frac{Fw}{v} \qquad [1]$$

TABLE 1 NOZZLE SIZES AND CAPACITIES

Nozzle number	Drill size	Diam. by measurement, in.	Capacity, lb. per hr.	Capacity, cu. ft. per min.
1	No. 46	0.08537	6.79	1.51
2	No. 33	0.11239	11.77	2.62
3	No. 21	0.15683	22.93	5.10
4	No. 9	0.19653	36.01	8.01
5	No. 1	0.22671	47.92	10.70
6	1/4 in.	0.24944	58.00	12.90
7	9/32 in.	0.27721	71.63	15.90
8	5/16 in.	0.31527	92.65	20.60
9	11/32 in.	0.34385	110.20	24.50
10	3/8 in.	0.37490	131.00	29.10
11	13/32 in.	0.40600	153.70	34.20
12	7/16 in.	0.43822	179.00	39.80
13	15/32 in.	0.46755	203.80	45.30
14	1/2 in.	0.49895	232.10	51.60
15	9/16 in.	0.56823	301.00	67.00
16	5/8 in.	0.61981	358.10	79.70
17	11/16 in.	0.68735	403.90	89.90
18	3/4 in.	0.74950	523.70	116.50
19	7/8 in.	0.87595	715.20	159.10
20	1 in.	0.99885	930.00	206.90

NOTE. — Capacity based on dry air at 70 deg. fahr. and with an initial pressure of 29.92 in mercury and a final pressure less than 15 in. mercury.

where

F = area of orifice, sq. ft.

v = specific volume of the fluid within the orifice, cu. ft. per lb.

w = average velocity within the orifice, ft. per sec.

G = rate of flow of fluid through the nozzle, lb. per sec.

7 With frictionless flow and no heat imparted to or received

from the flowing fluid by the nozzle, the fluid expands adiabatically, and for a gas we have the relation

$$p_1 v_1^k = pv^k \quad \ldots\ldots\ldots\ldots [2]$$

in which k = the abiabatic exponent (for atmospheric air, k = 1.40).

8 The work done in increasing the kinetic energy during the adiabatic expansion of one pound of gas is represented by the shaded area 1234 of Fig. 2 and is equal to

$$H = \frac{k}{k-1}\left[p_1 v_1 - pv\right] \quad \ldots\ldots\ldots [3]$$

Substituting v from Equation [2],

$$H = \frac{k}{k-1}\, p_1 v_1 \left[1 - \left(\frac{p}{p_1}\right)^{\frac{k-1}{k}}\right] \quad \ldots\ldots\ldots [4]$$

With the gas originally at rest, the increase in kinetic energy of one pound is

$$H = \frac{w^2}{2g} \quad \ldots\ldots\ldots\ldots [5]$$

Equating Equations [4] and [5], we obtain the velocity

$$w = \sqrt{\frac{2\,gk}{k-1}\, p_1 v_1 \left[1 - \left(\frac{p}{p_1}\right)^{\frac{k-1}{k}}\right]} \quad \ldots\ldots [6]$$

Substituting Equations [2] and [6] in [1], the rate of weight flow is found to be

$$G = F\sqrt{\frac{2\,gk}{k-1}\,\frac{p_1}{v_1}\left[\left(\frac{p}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p}{p_1}\right)^{\frac{k+1}{k}}\right]} \quad \ldots\ldots [7]$$

9 Now, if the flow is not frictionless, part of the kinetic energy theoretically available by adiabatic expansion is used up in overcoming frictional resistances, and the actual kinetic energy developed, the velocity and the rate of flow will all be less than given by the above formulæ. The actual increase in kinetic energy H_e may be represented in Fig. 2 by the area 1534 less than the area 1234. The actual expansion of the gas, however, is along the line 16 to a final volume v_a greater than v, since the work of friction generates an equivalent quantity of heat which is taken up by the flowing gas. The area 165 represents the work W done against friction, and it is interesting to note that the total work H_a done in overcoming friction and in increasing velocity, area 1634, is greater than the work H done during adiabatic expansion to the same lower pressure without frictional resistances, area 1234. The additional work done in this case over that theoretically available without

frictional resistances comes from the heat generated by friction and returned immediately to the fluid.

10 When a fluid flows with velocity w through a tube of diameter d and length dx, the loss of energy dW in consequence of friction may be calculated from

$$dW = \xi \frac{dx}{d} \frac{w^2}{2g} = \xi \frac{dx}{d} H_e \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

when ξ is the coefficient of friction. Hence for a nozzle we may write for the total frictional resistance

$$W = \theta H_e = H_a - H_e \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

where θ represents the coefficient of resistance of the whole nozzle and is a constant.

11 With a constant coefficient of resistance θ,

$$p_1 v_1^n = p v_a^n \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [10]$$

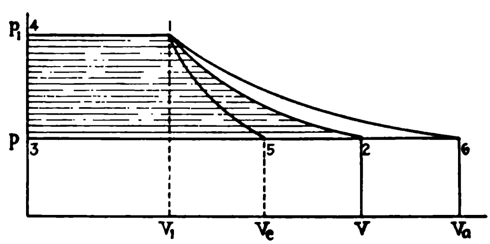

FIG. 2 PRESSURE-VOLUME DIAGRAM

where v_a is the final volume. The actual increase in kinetic energy is therefore given by

$$H_e = \frac{1}{k-1}\left[p_1 v_1 - p v_a\right] + \left[p_1 v_1 - p v_a\right] = \frac{k}{k-1}\left[p_1 v_1 - p v_a\right] \; . \; . \quad [11]$$

Or by combination with Equation [10]

$$H_e = \frac{k}{k-1} p_1 v_1 \left[1 - \left(\frac{p}{p_1}\right)^{\frac{n-1}{n}}\right] \quad . \; . \; . \; . \; . \quad [12]$$

The actual velocity w_e is given by

$$w_e = \sqrt{\frac{2\,gk}{k-1} p_1 v_1 \left[1 - \left(\frac{p}{p_1}\right)^{\frac{n-1}{n}}\right]} \quad . \; . \; . \; . \; . \quad [13]$$

The actual rate of weight flow is

$$G = F \sqrt{\frac{2\,gk}{k-1} \frac{p_1}{v_1} \left[\left(\frac{p}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p}{p_1}\right)^{\frac{n+1}{n}}\right]} \quad . \; . \; . \; . \quad [14]$$

12 The relation between the coefficient θ of Equation [**9**] and the exponent n of Equation [**10**] may be obtained by substituting the values of H_e and H_a in Equation [**9**]. The value of H_e is given in Equation [**11**] and that of H_a is equal to the area 1634, in Fig. 2, or

$$H_a = \frac{n}{n-1}\left[p_1v_1 - pv_a\right] \quad . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{15}]$$

Thus,

$$\theta H_e = H_a - H_e \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{16}]$$

And

$$\theta \frac{k}{k-1}\left[p_1v_1 - pv_a\right] = \frac{n}{n-1}\left[p_1v_1 - pv_a\right] - \frac{k}{k-1}\left[p_1v_1 - pv_a\right]$$

Whence

$$\theta = \frac{k-n}{k(n-1)} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{17}]$$

and

$$n = \frac{k(\theta+1)}{\theta k + 1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [\mathbf{18}]$$

That is, if θ is constant, the exponent n is constant, assuming k constant. Referring to Equations [**8**] and [**9**], it is seen that the coefficient θ depends upon the diameter and length of the nozzle; but for a given nozzle the coefficient θ and the exponent n can be taken as constants and independent of the pressure range. Also the coefficient θ and the exponent n should evidently be the same for geometrically similar nozzles.

13 The efficiency η of the nozzle may be defined as the ratio of the kinetic energy actually possessed by the flowing mass to that theoretically available by adiabatic expansion through the same pressure range. Thus,

$$\eta = \frac{H_e}{H} = \frac{\dfrac{k}{k-1}p_1v_1\left[1 - \left(\dfrac{p}{p_1}\right)^{\frac{n-1}{n}}\right]}{\dfrac{k}{k-1}p_1v_1\left[1 - \left(\dfrac{p}{p_1}\right)^{\frac{k-1}{k}}\right]} = \frac{1 - \left(\dfrac{p}{p_1}\right)^{\frac{n-1}{n}}}{1 - \left(\dfrac{p}{p_1}\right)^{\frac{k-1}{k}}} \quad . \; . \; . \quad [\mathbf{19}]$$

The efficiency η is therefore dependent upon the ratio of expansion $\left(\frac{p}{p_1}\right)$ as well as upon the exponent of expansion n.

14 If the rate of flow G be worked out from Equation [**14**] for various pressure ratios $\left(\frac{p}{p_1}\right)$, it will be found to have a maximum

value for a certain ratio. In fact, by differentiation this critical pressure ratio may be found to be

$$\frac{p}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad \ldots\ldots\ldots\ldots \quad [20][1]$$

From this follows the critical volume ratio

$$\frac{v}{v_1} = \left(\frac{n+1}{2}\right)^{\frac{1}{n-1}} \quad \ldots\ldots\ldots\ldots \quad [21]$$

and the critical velocity

$$w = \sqrt{\frac{2gk}{k-1}\left(\frac{n-1}{n+1}\right)p_1v_1} \quad \ldots\ldots\ldots \quad [22]$$

The maximum rate of flow is

$$G_m = F\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}}\sqrt{\frac{2gk}{k-1}\left(\frac{n-1}{n+1}\right)\frac{p_1}{v_1}} \quad \ldots\ldots \quad [23]$$

If the final pressure p_2 beyond the nozzle is less than the critical pressure p as given by Equation [20], the above maximum rate of flow will obtain.

15 Equation [23] may be put into a form somewhat more convenient for calculation with calibrated nozzles, for by letting

$$\psi = \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}}\sqrt{\frac{2gk}{k-1}\left(\frac{n-1}{n+1}\right)} \quad \ldots\ldots \quad [24]$$

we may then write

$$\frac{G_m}{F} = \psi\sqrt{p_1d_1} \quad \ldots\ldots\ldots\ldots \quad [25]$$

where d_1 is the initial density of the gas (the reciprocal of the specific volume v_1). Also, since for air we have the absolute temperature constant 458.5,

$$\frac{p_1}{d_1} = p_1v_1 = R_1(458.5 + t_1) \quad \ldots\ldots\ldots \quad [26]$$

where t_1 is the initial temperature in degrees fahrenheit and

$$\frac{G_m}{F} = \frac{\psi}{\sqrt{R_1}}\,\frac{p_1}{\sqrt{458.5 + t_1}} \quad \ldots\ldots\ldots \quad [27]$$

16 If it is desired to express the flow as Q cubic feet per minute instead of G pounds per hour then Equations [25] and [27] become

$$\frac{Q_m}{F} = \frac{\psi}{60}\sqrt{\frac{p_1}{d_1}} = \frac{\psi}{60}\sqrt{p_1v_1} \quad \ldots\ldots\ldots \quad [28]$$

and

$$\frac{Q_m}{F} = \frac{\psi}{60}\sqrt{R(458.5 + t_1)} \quad \ldots\ldots\ldots \quad [29]$$

[1] The critical pressure ratio for expansion without frictional resistance is obtained by substituting k for n.

17 The factor $\psi = \frac{G_m}{F\sqrt{p_1 d_1}}$ of Equation [25] and also the values of $\frac{p}{p_1}$ from Equation [20], $\frac{v}{v_1}$ from Equation [21] and $\frac{w}{\sqrt{p_1 v_1}}$ from Equation [22] depend upon the exponent n as well as upon k.

18 If the ratio of the final pressure beyond the nozzle to the initial pressure is *less* than the ratio $\frac{p}{p_1}$ then the rate of flow through the nozzle is equal to G_m and may be calculated by Equation [25] or [27]. If, however, the ratio is *greater* than $\frac{p}{p_1}$ then Equation [14] must be used. Calculations by Equation [14] may be simplified by putting it in a form similar to Equation [25]; that is, if we let

$$\psi_1 = \sqrt{\frac{2gk}{k-1}\left[\left(\frac{p}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p}{p_1}\right)^{\frac{n+1}{n}}\right]} \quad . \ . \ . \ . \ . \ [30]$$

we may then write

$$\frac{G}{F} = \psi_1\sqrt{p_1 d_1} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ [31]$$

The factor ψ_1 of course depends upon the pressure ratio $\frac{p}{p_1}$ as well as the exponent n.

19 The values of R_1 in Equation [27] and d_1 in Equation [25] depend upon the ratio Z of moisture and dry air in the atmospheric air flowing through the nozzle. To determine this ratio, the following formula may be employed

$$Z = \frac{r_2 Z_2 - C_{pa}(t_1 - t_2)}{r_2 + C_{ps}(t_1 - t_2)} \quad . \ . \ . \ . \ . \ . \ . \ [32]^1$$

in which Z = lb. water vapor per lb. of dry air in the mixture

t_1 = initial air temperature, in deg. fahr.

t_2 = wet-bulb temperature, in deg. fahr.

C_{pa} = 0.24112 + 0.0000045 $(t_1 + t_2)$, mean specific heat of dry air in B.t.u. per lb. per deg. fahr. temperature change between t_1 and t_2

C_{ps} = 0.4423 + 0.00009 $(t_1 + t_2)$, mean specific heat of superheated steam in B.t.u. per lb. per deg. fahr. temperature change between t_1 and t_2

r_2 = latent heat of steam at t_2 in B.t.u. per lb.

[1] Rational Psychrometric Formulæ, W. H. Carrier, Trans. A.S.M.E., 1911 vol. 33, pp. 1005–1053.

$Z_2 = d_2 \times 0.7556 \times \dfrac{458.5 + t_2}{b - p_2}$, dry saturated water vapor per lb. of dry air at t_2 in lb.

d_2 = density of dry saturated steam at t_2 in lb. per cu. ft.

p_2 = absolute pressure of saturated steam at t_2 in inches of mercury

b = barometric pressure in inches of mercury.

FIG. 3 APPARATUS USED FOR TESTS

A, Tank; *M*, Mercury Gage; *N*, Nozzle; *H*, Exhauster

The value of the gas constant R_1 in Equation [26] may be obtained from the relation

$$R_1 = \frac{R_a + R_s Z}{1 + Z} \quad \ldots\ldots\ldots\ldots \quad [33]$$

in which R_a = 53.45 for dry air, R_s = 85.89 for water vapor, and Z is determined by Equation [32].

APPLICATION OF FORMULÆ DERIVED

20 Having determined by calibration the values of ψ from Equation [24] and of n from Equation [22] for a particular nozzle this nozzle may then be used to accurately measure the air drawn

through it from the atmosphere, provided that the following are measured:

b = barometric pressure, lb. per sq. ft.
t_1 = air temperature (dry bulb), deg. fahr.
t_2 = wet-bulb temperature, deg. fahr.
p_2 = pressure beyond nozzle, lb. per sq. ft.

21 If the pressure ratio of $\frac{p_2}{b}$ is less than $\frac{p}{p_1}$ from Equation [20] corresponding to the value of n for the particular nozzle in use, then Equations [25] or [27] may be employed direct, the values of d_1 or R_1 being obtained by means of Equations [33] and [26]. If, how-

FIG. 4 SPECIAL VALVE FOR CLOSING NOZZLE

ever, the pressure ratio is greater than $\frac{p}{p_1}$, then Equation [31] must be used in which the value of ψ_1 may be obtained by means of Equation [30].

DESCRIPTION OF APPARATUS USED

22 The nozzle to be calibrated was screwed into a brass bushing mounted at N (Fig. 3) in the side of the sheet-steel tank A. This tank was 8 ft. in diameter and 16 ft. high. A special cover or valve was arranged to close the nozzle. This valve, shown in its open position in Fig. 4, consisted of a rubber-covered plate mounted on trunnions in a flat bar hinged at one end to the tank and having a

handle at the other end. With the nozzle plug in place, the two screws at the back of the valve were adjusted so that its rubber-covered face fitted evenly over the face of the plug. The valve was held tightly closed by a hand nut screwed on a stud in the tank.

23 The atmospheric pressure was indicated by a mercury barometer hung on a nearby wall, and the vacuum within the tank was measured by a mercury gage *M* (Fig. 3). The latter was of the floating-scale type, consisting of a cast-iron float with a brass scale attached thereto. The float rises or falls with the mercury in the cistern so that the zero of the scale always corresponds with the sur-

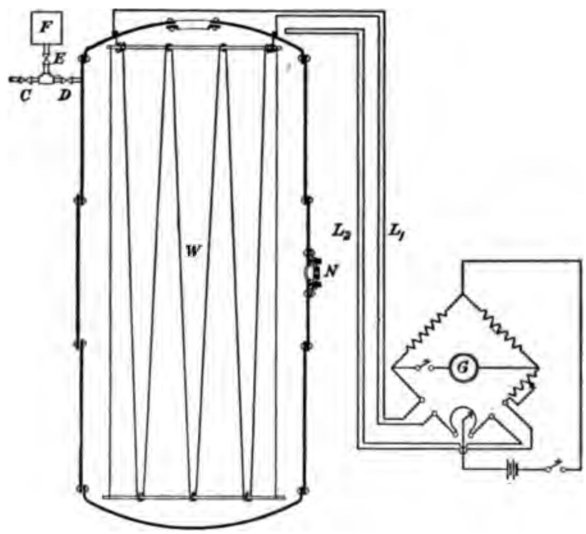

FIG. 5 DIAGRAM SHOWING METHOD EMPLOYED TO MEASURE TEMPERATURE WITHIN TANK

face of the mercury. The difference between atmospheric pressure and the absolute pressure in the tank was therefore obtained by a single reading — the height of the upper end of the column of mercury in the gage glass, obtained by means of a sliding index. Only one side of the column was used, it being of the duplex type. A thermometer was hung on the column to obtain its temperature.

24 The temperature within the tank was obtained from the resistance of a copper wire (No. 36 B. & S. Gage) strung from top to bottom of the tank six times and distributed across a diameter as shown at *W* in Fig. 5. This figure also shows diagrammatically the arrangement of the bridge used to measure the resistance of the

wire W. The leads L_1 were of No. 12 copper wire, with compensating leads L_2 of the same size and length.

25 The temperature and humidity of the atmospheric air flowing into the nozzle under calibration were determined from readings of dry- and wet-bulb thermometers placed in the suction duct of a small exhauster H (Fig. 3). The wet bulb was moistened by a cotton wick which dipped into one leg of a U-tube kept filled with water. By raising or lowering this U-tube, the cotton wick could be kept just sufficiently moist around the thermometer bulb, a drop of water occasionally falling from its lower end.

26 The evacuation of the tank was accomplished by a Westinghouse-Leblanc air ejector connected to the tank at D (Fig. 5). While the tank was being exhausted, valves C and D were open and valve E was closed. When the desired vacuum was attained, valves C and D were closed and valve E was opened to admit water from the vessel F to water-seal valve D.

METHOD OF CONDUCTING TESTS

27 The nozzle to be calibrated was first mounted in place, the valve adjusted to cover the nozzle squarely and the hand nut screwed down to hold the valve tightly closed. The tank was then exhausted by the ejector to a vacuum of 28 to 29 in. of mercury and the intervening valves closed and water-sealed. The tank was then allowed to stand overnight in this exhausted condition in order to insure that a portion of the water previously put into the tank would have time to evaporate and saturate the interior space. It was found necessary to add water in order to replace the moisture withdrawn with the air by the ejector; and if sufficient time were not allowed for the water to evaporate, the vapor pressure taken from steam tables to correspond to the temperature within the tank would be greater than the actual existing pressure, thus indicating that the space was not saturated with water vapor.

28 In order to obtain some idea of the weight of water required to be put into the tank, a curve was derived to give the actual volume of air removed at a continuously decreasing pressure according to the relation

$$\frac{v_r}{v_1} = \log_e \frac{p_1}{p_2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [34]$$

in which v_r = total volume of air removed

v_1 = volume of tank

p_1 = initial pressure in tank

p_2 = final pressure in tank.

29 The application of this formula may be shown by an example. Thus, if the tank is exhausted from an atmospheric pressure of 30 in. mercury to a vacuum of 29 in. mercury, we shall have

$$\frac{p}{p_1} = \frac{30}{30 - 29} = 30$$

and from the formula

$$\frac{v_r}{v_1} = 3.41$$

The volume of the tank as determined by calibration with water, was about 865 cu. ft. Hence

$$v_r = 3.41 \times 865 = 2950 \text{ cu. ft.}$$

This will also be the volume of the water vapor removed; and assuming it to be saturated steam at a temperature of 70 deg. fahr. with a corresponding density of 0.001148 lb. per cu. ft. (taken from steam tables), we find 3.4 lb. are withdrawn. Accordingly, about 5 lb. of water was put in the tank before exhausting each time.

30 Allowing the exhausted tank to stand over night did not always, however, insure saturation, and before taking readings the nozzle was opened to admit atmospheric air until the absolute pressure within the tank doubled; that is, the vacuum dropped from say 29 to 28 in. mercury. The moisture brought in from the atmosphere together with the vapor in the tank insured saturation.

31 Each calibration run, during which the air nozzle under calibration was open, was preceded and followed by a leakage period During the initial leakage period, readings of the barometric pressure, the vacuum within the tank and the temperature within the tank were taken at intervals of 3 min. for 30 min. Then the nozzle was quickly opened at a given second to start the run, and readings of the barometer, mercury column, and wet- and dry-bulb thermometers taken at intervals of 30 sec. The run was ended by quickly closing the nozzle a certain number of minutes after opening it, the length of the run depending upon the size of the nozzle. A short run was necessary with the larger nozzles in order to prevent the pressure within the tank rising above the critical value. The final leakage period began at the time the nozzle was closed and continued for 30 min.

CALCULATION OF RESULTS

32 The methods employed in correcting the data and deducing the results are illustrated and explained in the Appendix for

Run 1 of Nozzle 1. To obtain the dry-air density in the tank as tabulated under Items 10, 11, 14, and 15, of the Appendix, the temperature and pressure observations for the leakage intervals preceding and following each run were plotted as shown in Fig. 6, and the calculations then based on these curves rather than directly upon the observations. A curve was also plotted showing the relation of weight-flow factor ψ to expansion exponent n so as to enable Item 25 being obtained from Item 24, and in Table 2 there have been tabulated for each nozzle the average values of the weight-flow factor ψ, the expansion exponent, and the weight-flow efficiency. The number of runs made on each nozzle is also given. It will be

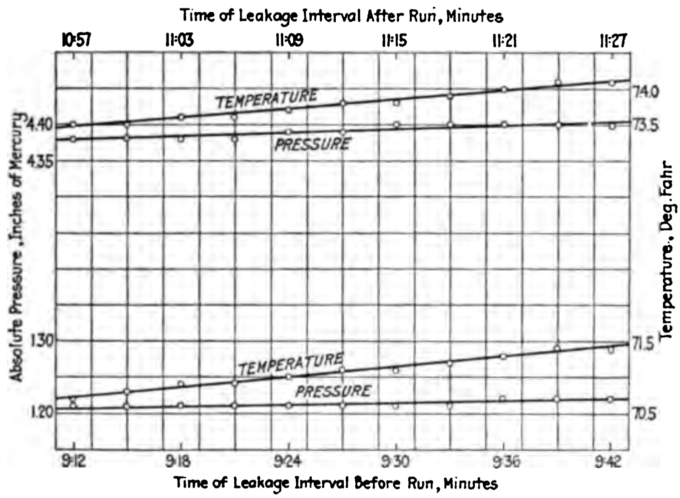

FIG. 6 CURVES SHOWING ABSOLUTE PRESSURE AND TEMPERATURE FOR LEAKAGE INTERVALS BOTH BEFORE AND AFTER RUN NO. 1 ON NOZZLE NO. 1

noted that the calculations were based upon the pressure and temperature within the tank before the nozzle was opened and after it was closed, when the conditions were static, rather than upon the pressure and temperature, while the air was flowing into the tank.

33 Assuming all the individual runs to be of equal weight in applying the formula for the probable error of the mean value, we obtain the expansion exponent $n = 1.3747 \pm 0.0013$, and assuming the average values in Table 2 for the several nozzles to be of equal weight, we obtain $n = 1.3753 \pm 0.0018$. The probable errors for a single run and for a single nozzle are, thereby, respectively $\pm$ 0.0107 and $\pm$ 0.0078. That is, for carefully made nozzles geometrically

similar to those tested, the value of n will probably be within 0.010 of 1.375 if the observations are taken as carefully as during this calibration test. The corresponding value and accuracy of the weight-flow efficiency are 97.1 ± 1.2 per cent.

34 Corresponding to the expansion exponent $n = 1.375$, the numerical value of the weight-flow factor ψ in the formula

$$\frac{G}{F} = \psi\sqrt{pd}$$

will depend upon the units in which the several quantities G, F, p and d are expressed, as indicated in Table 3. For final pressures greater than the critical pressure, that is $\frac{p}{p_1}$ greater than 0.533, the

TABLE 2 SUMMARY OF RESULTS

Nozzle Number	Nozzle diam., in.	Number of runs	Average weight-flow factor	Average expansion exponent	Average weight-flow efficiency per cent
1	0.08537	9	3.646	1.349	93.86
2	0.11239	6	3.786	1.378	97.48
3	0.15683	12	3.770	1.375	97.08
4	0.19653	4	3.824	1.387	98.46
5	0.22671	5	3.853	1.393	99.20
6	0.24944	2	3.768	1.375	97.02
7	0.27721	4	3.834	1.389	98.72
8	0.31527	2	3.748	1.370	96.49
9	0.34385	4	3.791	1.380	97.61
10	0.37490	2	3.776	1.376	97.22
11	0.40600	2	3.779	1.377	97.28
12	0.43822	4	3.767	1.374	96.98
13	0.46755	2	3.774	1.376	97.16
14	0.49895	2	3.781	1.377	97.35
15	0.56823	2	3.759	1.373	96.77
16	0.51961	2	3.831	1.388	98.64
17	0.68735	2	3.791	1.379	97.61
18	0.74950	1	3.636	1.346	93.61
19	0.87595	1	3.790	1.379	97.58
20	0.99885	1	3.728	1.366	95.98

factor will be reduced as given by the solid curve in Fig. 7, for the units in the sixth line of Table 3.

35 Changing the above formula into the form

$$\frac{G}{F} = \frac{\psi}{\sqrt{R}} \frac{p}{\sqrt{458.5 + t}}$$

also changes in certain cases the numerical value of ψ for the same units of G, F, p and t, as shown in Table 4. For pressure ratios greater than 0.533, the dotted curve in Fig. 7 gives the reduced values of ψ corresponding to the units in the sixth line of Table 4.

36 For moist air, the value of the gas constant R is larger than for dry air, as may be seen by an inspection of Equation [33]. Consequently, the factor $\phi = \dfrac{\psi}{\sqrt{R}}$ in the formula of Par. 35 is less

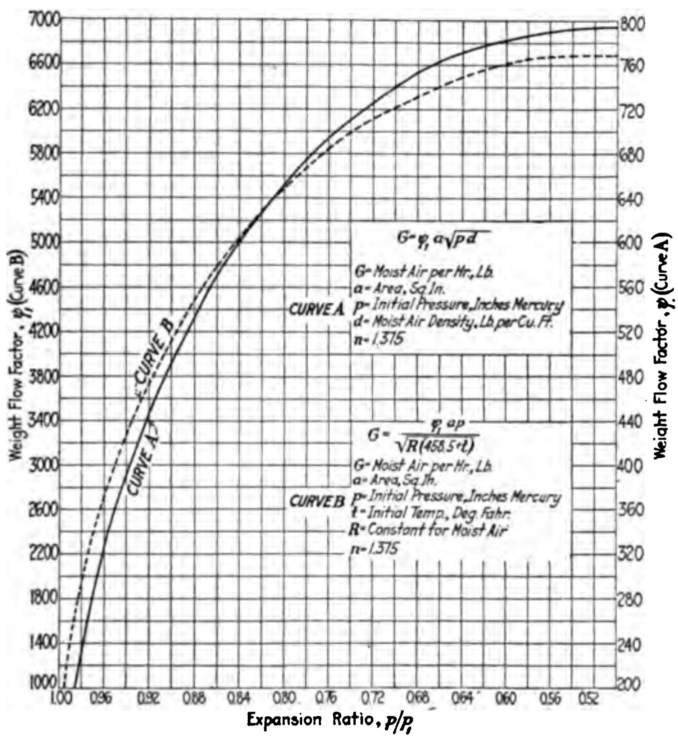

FIG. 7 CURVES OF WEIGHT-FLOW FACTOR VS. EXPANSION RATIO FOR $n = 1.375$

for moist air than for dry air. The weight of moist air flowing through a nozzle is therefore less than of dry air under the same initial pressure and temperature. Incidentally, moist air is less dense than dry air under the same pressure and temperature. The reduction in weight flow is given by the "moist air fraction" in Table 6.

37 In testing air pumps for condenser service, the weight of dry air flowing through a nozzle in a given time is desired, rather than the total weight of moist air flowing. The weight of dry air in a mixture of Z pounds of water vapor to one pound of dry air is equal to the weight of moist air divided by $(1 + Z)$. The dry air factor θ in Table 6 is therefore obtained by dividing ϕ by $(1+Z)$. If the weight of dry air that would flow through a nozzle in a given time is known, the weight of dry air in a mixture of water vapor and dry air flowing under the same initial pressure and temperature is obtained by multiplying by the "dry air factor" in Table 6.

38 Based on an expansion exponent of $n = 1.375$, the rates of flow of dry air containing no moisture under the normal conditions of 70 deg. fahr. and 29.92 in. mercury pressure, 0.07492 lb. per cu. ft. density, have been calculated for the nozzles calibrated, and the results are tabulated in Table 1. If the air contains moisture, the values in Table 1 may be multiplied by the fractions in Table 6.

TABLE 3 VALUES OF WEIGHT-FLOW FACTOR FOR VARIOUS UNITS

Weight flow, G	Orifice area, F	Initial pressure, p	Initial density, d	Numerical value of factor, Ψ
lb. per sec.	sq. ft.	lb. per sq. ft.	lb. per cu. ft.	3.771
lb. per sec.	sq. in.	lb. per sq. in.	lb. per cu. ft.	0.3143
lb. per sec.	sq. in.	inches mercury	lb. per cu. ft.	0.2203
lb. per hr.	sq. ft.	lb. per sq. ft	lb. per cu. ft.	13576
lb. per hr.	sq. in.	lb. per sq. in.	lb. per cu. ft.	1131.3
lb. per hr.	sq. in.	inches mercury	lb. per cu. ft.	792.9
kg. per hr.	sq. cm.	mm. mercury	kg. per cu. m.	11.061

TABLE 4 VALUES OF WEIGHT-FLOW FACTOR FOR VARIOUS UNITS

Weight flow, G	Orifice area, F	Initial pressure, p	Initial temperature, t	Numerical value of factor, Ψ
lb. per sec.	sq. ft.	lb. per sq. ft.	deg. fahr.	3.771
lb. per sec.	sq. in.	lb. per sq. in.	deg. fahr.	3.771
lb. per sec.	sq. in.	inches mercury	deg. fahr.	1.852
lb. per hr.	sq. ft.	lb. per sq. ft.	deg. fahr.	13576
lb. per hr.	sq. in.	lb. per sq. in.	deg. fahr.	13576
lb. per hr.	sq. in.	inches mercury	deg. fahr.	6668
kg. per hr.	sq. cm.	mm. mercury	deg. cent	18.458

TABLE 5. DATA OF TESTS

Reference number	Date, 1916	Duration of run with nozzle open, minutes	Barometric pressure, inches mercury	Room temperature (dry bulb), degrees fahr.	Wet bulb temperature, degrees fahr.	Water vapor per pound of dry air, lb.	Density of moist air, lb. per cu.ft.	Duration of leakage interval before run, minutes	Dry air density in tank at beginning of interval, lb. per cu.ft.	Dry air density in tank at end of interval, lb. per cu.ft.	Density change per minute, lb. per cu.ft.	Duration of leakage interval after run, minutes	Dry air density in tank at beginning of interval, lb. per cu.ft.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Sept.26	60.0	30.12	70.0	59.4	0.00828	0.07502	30	0.00112	0.00112	0.000000	30	0.00863
2	"	59.0	30.11	71.3	61.4	0.00924	0.07479	30	0.00863	0.00863	0.000000	30	0.01641
3	"	59.0	30.06	72.1	61.2	0.00900	0.07464	30	0.01641	0.01641	0.000000	30	0.02396
4	Dec.19	60.0	29.95	63.3	52.2	0.00374	0.07565	30	0.00617	0.00618	0.000000	30	0.01388
5	"	60.0	29.88	63.7	53.2	0.00621	0.07545	30	0.01368	0.01292	0.000001	30	0.02158
6	"	60.0	29.85	64.1	54.1	0.00661	0.07528	30	0.02158	0.02165	0.000002	30	0.02980
7	Dec.23	60.0	30.21	67.8	55.2	0.00629	0.07568	30	0.00161	0.00157	-0.000001	30	0.00932
8	"	60.0	30.16	71.0	56.2	0.00614	0.07507	30	0.00932	0.00934	0.000001	30	0.01691
9	"	60.0	30.12	72.2	55.9	0.00371	0.07482	30	0.01691	0.01706	0.000005	30	0.02489
10	Sept.28	45.0	30.09	76.5	70.2	0.01424	0.07379	30	0.00028	0.00085	-0.000001	30	0.01039
11	"	46.0	30.07	79.1	71.1	0.01436	0.07339	30	0.01039	0.01038	0.000000	30	0.02067
12	"	45.0	30.04	77.8	71.4	0.01491	0.07346	30	0.02067	0.02061	-0.000002	30	0.03063
13	Dec.20	46.0	30.08	64.6	52.9	0.00579	0.07585	30	0.00124	0.00134	0.000000	30	0.01192
14	"	44.0	30.01	67.2	54.3	0.00597	0.07527	30	0.01191	0.01196	0.000001	30	0.02183
15	"	43.0	30.01	69.5	55.7	0.00625	0.07493	30	0.02182	0.02185	0.000001	30	0.03152
16	July 17	38.0	29.90	81.4	73.8	0.01612	0.07260	30	0.00053	0.00044	-0.000002	30	0.01473
17	"	35.0	29.88	81.2	73.2	0.01567	0.07260	30	0.01473	0.01482	0.000003	30	0.02945
18	July 19	33.0	30.10	81.5	76.7	0.01853	0.07298	30	-0.00012	-0.00012	0.000000	30	0.01412
19	"	32.0	30.08	83.2	76.1	0.01758	0.07273	30	0.01412	0.01412	0.000000	30	0.02803
20	Aug.24	37.0	29.87	74.0	64.4	0.01069	0.07375	30	0.00116	0.00114	-0.000001	30	0.01674
21	"	36.0	29.86	75.1	64.8	0.01073	0.07359	30	0.01674	0.01677	0.000001	30	0.03185
22	Sept.29	23.0	29.83	78.2	70.8	0.01467	0.07326	30	0.00078	0.00078	0.000000	30	0.01080
23	"	20.0	29.85	71.7	66.5	0.01272	0.07392	30	0.01080	0.01083	0.000001	30	0.01961
24	"	16.0	29.83	67.0	62.0	0.01072	0.07467	30	0.01961	0.01963	0.000001	30	0.02672
25	Oct.14	23.0	30.23	73.7	63.9	0.01028	0.07471	30	0.00168	0.00171	0.000001	30	0.01197
26	"	23.0	30.21	73.4	64.7	0.01043	0.07441	30	0.01187	0.01180	-0.000002	30	0.02193
27	"	22.0	30.20	74.0	61.5	0.00863	0.07464	30	0.02193	0.02197	0.000001	30	0.02172
28	Sept.30	22.0	30.21	72.2	58.8	0.00736	0.07500	30	0.00090	0.00082	-0.000003	30	0.01624
29	"	23.0	30.20	72.2	57.3	0.00648	0.07499	30	0.01624	0.01615	-0.000003	30	0.03230
30	Dec.21	21.0	30.01	69.5	58.9	0.00811	0.07485	30	0.00180	0.00147	-0.000001	30	0.01603
31	"	21.0	29.99	71.9	60.7	0.00870	0.07443	30	0.01603	0.01599	-0.000001	30	0.03052
32	July 12	13.0	30.06	84.8	77.2	0.01829	0.07248	30	0.01965	0.01965	0.000000	30	0.02175
33	July 13	10.0	29.95	86.6	78.3	0.01889	0.07189	30	0.00130	0.00134	0.000001	30	0.01765
34	"	14.0	29.95	88.1	79.3	0.01947	0.07167	30	0.01765	0.01760	-0.000002	30	0.03028
35	Aug.28	17.0	30.02	76.2	71.5	0.01536	0.07364	30	0.00032	0.00029	-0.000001	30	0.01588
36	"	17.0	30.05	79.5	75.2	0.01771	0.07313	30	0.01588	0.01581	-0.000002	30	0.03134
37	Oct.3	14.0	30.25	74.3	60.3	0.00790	0.07477	30	0.00003	0.00008	-0.000002	30	0.01563
38	"	14.0	30.23	74.6	60.6	0.00790	0.07466	30	0.01563	0.01570	0.000002	30	0.03139
39	June 21	16.0	29.90	76.4	71.4	0.01532	0.07222	30	0.00283	0.00286	0.000001	30	0.02494
40	June 22	19.0	30.19	69.3	57.9	0.00749	0.07536	30	0.00062	0.00062	0.000000	30	0.02626
41	Aug.30	11.0	30.18	73.5	65.5	0.01142	0.07434	30	0.00057	0.00057	0.000000	30	0.01620
42	"	11.0	30.18	73.9	65.6	0.01141	0.07448	30	0.01620	0.01619	0.000000	30	0.03168
43	Oct.5	9.0	30.41	73.8	69.7	0.01432	0.07497	30	-0.00025	-0.00022	0.000001	30	0.01572
44	"	8.0	30.41	73.0	69.3	0.01422	0.07508	30	0.01572	0.01579	0.000002	30	0.02996
45	June 1	13.0	30.17	77.0	64.3	0.00977	0.07413	30	0.00086	0.00083	-0.000001	15	0.02814
46	June 2	13.0	30.10	72.9	63.8	0.01042	0.07431	30	0.00037	0.00034	-0.000001	15	0.02914
47	Sept.1	8.0	30.19	79.8	72.9	0.01564	0.07317	30	0.00029	0.00034	0.000002	30	0.01712
48	"	6.0	30.16	81.6	72.5	0.01404	0.07327	30	0.01712	0.01714	0.000001	30	0.02972
49	Oct.7	6.0	30.21	76.8	70.3	0.01419	0.07407	30	0.00253	0.00253	0.000000	30	0.01748
50	"	6.0	30.20	76.4	69.2	0.01345	0.07410	30	0.01748	0.01747	0.000000	30	0.03244
51	Oct.11	5.0	30.28	71.0	62.1	0.00968	0.07522	30	0.00861	0.00235	-0.000002	30	0.01738
52	"	5.0	30.27	72.0	64.6	0.01113	0.07501	30	0.01738	0.01738	0.000000	30	0.03213
53	May 29	6.0	29.92	76.1	70.8	0.01486	0.07342	30	0.00135	0.00130	-0.000002	22	0.02800
54	May 31	8.0	29.98	77.1	64.5	0.01000	0.07350	30	0.00132	0.00130	-0.000001	24	0.02819
55	Sept.5	4.0	30.12	76.3	69.8	0.01391	0.07391	30	0.00066	0.00067	0.000000	30	0.01425
56	"	5.0	30.12	78.3	70.7	0.01421	0.07360	30	0.01433	0.01431	-0.000001	30	0.02121
57	Oct.12	4.0	30.24	74.4	59.9	0.00751	0.07474	30	0.00147	0.00157	0.000003	30	0.01728
58	"	4.0	30.23	74.5	60.0	0.00756	0.07471	30	0.01731	0.01736	0.000002	30	0.03310
59	Oct.17	3.5	29.91	75.0	65.3	0.01108	0.07369	30	0.00106	0.00101	-0.000002	30	0.01642
60	"	2.5	29.94	73.2	61.9	0.00917	0.07412	30	0.01642	0.01638	-0.000001	30	0.03192
61	Sept.7	2.0	30.02	76.1	72.7	0.01638	0.07363	30	0.00057	0.00054	-0.000001	30	0.01750
62	"	2.5	30.02	77.1	73.1	0.01647	0.07360	30	0.01750	0.01710	0.000000	30	0.03160
63	Oct.18	2.0	30.48	74.2	61.3	0.00836	0.07533	30	0.00232	0.00233	0.000000	30	0.01642
64	"	2.0	30.46	72.8	56.9	0.00720	0.07532	30	0.01642	0.01641	0.000000	30	0.03090
65	Sept.11	4.0	30.40	71.8	61.7	0.00916	0.07563	30	0.00026	0.00030	0.000001	30	0.03403
66	Dec.22	3.5	29.82	67.6	56.9	0.00757	0.07365	30	0.00196	0.00193	-0.000001	30	0.03089
67	Oct.19	3.0	29.88	72.0	58.0	0.01374	0.07394	30	0.00224	0.00226	0.000001	30	0.03207
68	Sept.13	2.0	30.18	76.2	69.0	0.01334	0.07407	30	0.00071	0.00071	0.000000	30	0.02803
69	Oct.20	1.5	29.78	77.0	72.1	0.01504	0.07281	30	0.00089	0.00089	0.000000	30	0.02643

TABLE 5. (*concl'd*) DATA OF TESTS

Dry air density in tank at end of interval, lb. per cu.ft.	Density change per minute, lb. per cu. ft.	Dry air density change during run, lb. per cu.ft.	Dry air through nozzle during run, lb.	Dry air per hour, lb.	Moist air per hour, lb.	Nozzle number	Nozzle diameter, inches	Nozzle area, sq. in.	Weight flow factor, ψ'.	Expansion exponent, n.	Weight flow efficiency, per cent.
15	16	17	18	19	20	21	22	23	24	25	26
0.00883	0.000000	0.00771	6.668	6.669	6.723	1	0.08537	0.00572	3.720	4.364	95.79
0.01641	0.000000	0.00758	6.556	6.667	6.729	1	"	"	3.731	1.366	96.06
0.02397	0.000001	0.00747	6.461	6.371	6.630	1	"	"	3.683	1.356	94.82
0.01392	0.000001	0.00762	6.590	6.590	6.638	1	"	"	3.460	1.351	94.23
0.02185	0.000002	0.00751	6.495	6.473	6.535	1	"	"	3.623	1.344	93.29
0.02935	0.000002	0.00750	6.487	6.487	6.530	1	"	"	3.628	1.345	93.41
0.00984	0.000001	0.00775	6.702	6.723	6.745	1	"	"	3.710	1.362	95.52
0.01704	0.000006	0.00734	6.348	6.348	6.367	1	"	"	3.534	1.326	90.99
0.02462	0.000031	0.00730	6.314	6.314	6.330	1	"	"	3.521	1.323	90.63
0.01030	0.000000	0.01320	8.822	11.76	11.95	2	0.11229	0.00992	3.840	1.390	98.87
0.02061	0.000002	0.01035	8.952	11.68	11.85	2	"	"	3.823	1.386	98.43
0.02073	0.000002	0.01007	8.709	11.41	11.78	2	"	"	3.800	1.381	97.84
0.01152	0.000000	0.01058	9.150	11.93	12.00	2	"	"	3.810	1.382	98.09
0.02185	0.000001	0.00981	8.484	11.37	11.64	2	"	"	3.712	1.362	95.57
0.03152	0.000000	0.00961	8.311	11.60	11.67	2	"	"	3.731	1.366	96.06
0.01482	0.000003	0.01422	12.299	22.36	22.72	3	0.15683	0.01932	3.795	1.380	97.71
0.03000	0.000002	0.01498	12.936	22.21	22.56	3	"	"	3.771	1.375	97.09
0.01412	0.000000	0.01401	12.117	22.03	22.44	3	"	"	3.726	1.365	95.93
0.02803	0.000000	0.01391	12.030	22.96	22.96	3	"	"	3.819	1.386	98.33
0.01674	0.000000	0.01565	13.535	21.95	22.10	3	"	"	3.678	1.356	94.70
0.03191	0.000002	0.01496	12.936	21.99	21.82	3	"	"	3.623	1.344	93.28
0.01083	0.000001	0.00998	8.632	22.52	22.83	3	"	"	3.801	1.381	97.86
0.01963	0.000001	0.00874	7.559	22.68	22.97	3	"	"	3.807	1.383	98.02
0.02669	-0.000001	0.00709	6.132	23.00	23.28	3	"	"	3.833	1.389	98.69
0.01190	-0.000002	0.01020	8.822	23.01	23.25	3	"	"	3.808	1.382	98.04
0.02197	0.000001	0.01017	8.746	22.95	23.19	3	"	"	3.807	1.383	98.02
0.03177	0.000002	0.00968	8.372	23.03	23.32	3	"	"	3.777	1.375	97.23
0.01615	-0.000003	0.01553	13.432	36.63	36.90	4	0.19653	0.03034	3.843	1.391	98.94
0.03228	-0.000001	0.01623	14.037	36.63	36.86	4	"	"	3.840	1.390	98.87
0.01599	-0.000001	0.01460	12.627	36.08	36.37	4	"	"	3.804	1.382	97.94
0.03032	0.000000	0.01437	12.601	36.07	36.31	4	"	"	3.810	1.384	98.09
0.03175	0.000000	0.01190	10.292	47.50	48.27	5	0.22671	0.04037	3.860	1.395	99.38
0.01760	-0.000003	0.01634	14.132	47.11	[illegible]	5	"	"	3.855	1.394	99.25
0.03029	0.000000	0.01271	10.993	47.11	48.02	5	"	"	3.864	1.395	99.49
0.01581	-0.000002	0.01365	13.575	47.77	48.50	5	"	"	3.843	1.391	98.94
0.03134	0.000001	0.01516	12.458	47.50	48.24	5	"	"	3.843	1.391	98.94
0.01570	0.000002	0.01549	13.377	57.47	57.87	6	0.24944	0.04887	3.746	1.370	96.43
0.03136	-0.000001	0.01366	13.344	58.08	58.51	6	"	"	3.790	1.379	97.58
0.02484	0.000000	0.02195	18.994	71.19	72.28	7	0.27721	0.06035	3.847	1.392	99.05
0.04622	-0.000002	0.02569	21.219	70.16	70.68	7	"	"	3.693	1.358	95.08
0.01619	0.000000	0.01563	13.518	73.72	74.57	7	"	"	3.919	1.407	100.90
0.03171	0.000001	0.01536	13.271	72.93	73.76	7	"	"	3.878	1.399	99.85
0.01579	0.000002	0.01589	13.742	91.62	92.92	8	0.31527	0.07806	3.749	1.370	96.52
0.02999	0.000001	0.01412	12.212	91.59	93.69	8	"	"	3.746	1.370	96.45
0.02816	0.000001	0.02731	23.620	109.71	110.08	9	0.34285	0.09286	3.770	1.375	97.06
0.02314	0.000000	0.02763	23.897	110.29	111.44	9	"	"	3.810	1.383	98.09
0.01714	0.000001	0.01673	14.469	108.53	110.22	9	"	"	3.768	1.379	97.53
0.02976	0.000001	0.01256	10.862	108.62	110.24	9	"	"	3.797	1.381	97.76
0.01747	0.000000	0.01495	12.930	129.30	131.13	10	0.37490	0.11039	3.777	1.376	97.25
0.03249	0.000002	0.01495	12.930	129.30	131.04	10	"	"	3.775	1.376	97.19
0.01730	0.000000	0.01485	12.843	154.12	155.61	11	0.40600	0.12946	3.789	1.379	97.55
0.03219	0.000002	0.01473	12.740	152.88	154.56	11	"	"	3.766	1.374	97.01
0.02736	-0.000002	0.02675	23.136	173.52	176.09	12	0.43832	0.15089	3.746	1.370	96.45
0.02818	0.000000	0.02691	23.273	174.56	176.31	12	"	"	3.749	1.370	96.52
0.01431	-0.000001	0.01370	11.849	177.74	180.21	12	"	"	3.808	1.383	98.04
0.03127	0.000002	0.01686	14.799	175.19	177.68	12	"	"	3.764	1.372	96.91
0.01736	0.000003	0.01563	12.335	203.03	204.55	13	0.46755	0.17169	3.770	1.375	97.06
0.03323	0.000004	0.01568	13.561	203.42	204.96	13	"	"	3.777	1.376	97.25
0.01626	-0.000001	0.01537	13.293	227.88	230.40	14	0.49895	0.19553	3.775	1.376	97.19
0.03165	-0.000002	0.01550	13.406	229.82	231.93	14	"	"	3.787	1.378	97.50
0.01760	0.000000	0.01698	14.686	[illegible]	[illegible]	15	0.56823	0.25359	3.766	1.374	96.96
0.03163	0.000001	0.01468	13.176	[illegible]	[illegible]	15	"	"	3.751	1.371	96.58
0.01641	0.000000	0.01409	12.186	365.58	368.64	16	0.61981	0.30172	3.835	1.389	98.74
0.03049	0.000000	0.01409	12.186	365.58	363.21	16	"	"	3.827	1.387	98.33
0.03487	0.000001	0.03451	29.847	447.71	451.81	17	0.68728	0.37106	3.814	1.384	98.20
0.03091	0.000001	0.02856	25.047	429.29	432.63	17	"	"	3.766	1.374	97.01
0.03212	0.000002	0.02975	23.730	494.60	501.47	18	0.74950	0.44120	3.636	1.346	93.61
0.02807	0.000001	0.02730	23.611	708.36	717.81	19	0.87595	0.60263	3.790	1.379	97.58
0.02665	0.000001	0.02572	22.245	839.80	905.89	20	0.99885	0.78369	3.728	1.366	95.78
							Average		3.7691	1.3747	97.040

TABLE 6 FACTORS FOR MOIST AIR

$\frac{\text{Water Vapor}}{\text{Dry Air}} = Z$	Moist Air Factor $\phi = \frac{\psi}{\sqrt{R}}$	Moist Air Fraction	Dry Air Factor $\theta = \frac{\phi}{1 + Z}$	Dry Air Fraction
0.000	912.3	1.0000	912.3	1.0000
0.002	911.8	0.9994	910.0	0.9974
0.004	911.2	0.9988	907.6	0.9948
0.006	910.7	0.9982	905.3	0.9923
0.008	910.1	0.9976	902.9	0.9897
0.010	909.6	0.9970	900.6	0.9872
0.012	909.0	0.9964	898.3	0.9846
0.014	908.5	0.9958	896.0	0.9821
0.016	908.0	0.9953	893.7	0.9796
0.018	907.5	0.9947	891.4	0.9771
0.020	906.9	0.9941	889.1	0.9746

APPENDIX

METHOD OF CALCULATING RESULTS FROM OBSERVED DATA

Item

(1) Reference number 1

(2) Date . . . . Sept. 26, 1916

(3) Duration of run with nozzle open, min 60.0

(4) Barometric pressure, in. mercury 30.12

Corrected for scale error and reduced to a mercury density of 0.4912 lb. per cu. in.

(5) Room temperature (dry bulb), deg. fahr 70.0

Corrected for calibration error.

(6) Wet-bulb temperature, deg. fahr 59.4

Corrected for calibration error and for exposed stem.

(7) Water vapor per lb. of dry air, lb 0.00828

From Equation [32] for values corresponding to b = Item 4; t_1 = Item 5; and t_2 = Item 6.

(8) Density of moist air, lb. per cu. ft 0.007502

$$\text{From the formula } d = \frac{b \times 0.4912 \times 144}{R\,(458.5 + t_1)}$$

in which b = Item 4; t_1 = Item 5; and R = gas constant corresponding to Item 7.

(9) Duration of leakage interval before run, min 30.0

(10) Dry-air density in tank at beginning of interval, lb. per cu. ft 0.00112

$$\text{From the formula } d = 1.3234\,\frac{p_1 - p_s}{458.5 + t_1}$$

in which p_1 = absolute pressure in in. mercury at beginning of interval
t_1 = temperature in deg. fahr. at beginning of interval
p_s = absolute steam pressure from steam tables corresponding to t_1.

(11) Dry-air density in tank at end of interval, lb. per cu. ft 0.00112

$$\text{From the formula } d_2 = 1.3234\,\frac{p_2 - p_s}{458.5 + t_2}$$

in which p_2 = absolute pressure in in. at end of interval
t_2 = temperature in deg. fahr. at end of interval
p_s = absolute steam pressure from steam tables corresponding to t_2.

(12) Density change per min., lb. per cu. ft 0.000000

$$\text{From the expression } \frac{\text{Item 11} - \text{Item 10}}{\text{Item 9}}$$

(13) Duration of leakage interval after run, min . . . . 30.

(14) Dry-air density in tank at beginning of interval, lb. per cu. ft 0.0088

From the formula $d_3 = 1.3234 \dfrac{p_3 - p_s}{458.5 + t_3}$

in which p_3 = absolute pressure in in. mercury at beginning of interval
t_3 = temperature in deg. fahr. at beginning of interval
p_s = absolute steam pressure from steam tables corresponding to t_3.

(15) Dry-air density in tank at end of interval, lb. per cu. ft. 0.00883

From the formula $d_4 = 1.3234 \dfrac{p_4 - p_s}{458.5 + t_4}$

in which p_4 = absolute pressure in in. mercury at end of interval
t_4 = temperature in deg. fahr. at end of interval
p_s = absolute steam pressure from steam tables corresponding to t_4.

(16) Density change per min., lb. per cu. ft. 0.000000

From the expression $\dfrac{\text{Item 15} - \text{Item 14}}{\text{Item 13}}$

(17) Dry-air density change during run, lb. per cu. ft. 0.0077
From the expression

$$(\text{Item 14} - \text{Item 11}) - \frac{\text{Item 12} + \text{Item 16}}{2}\ (\text{Item 3} + 15)$$

in which 15 min. is the interval from the time of closing the nozzle and the time of reading the pressure and temperature from which Item 14 is calculated.

(18) Dry air through nozzle during run, lb. 6.668
Item 17 multiplied by 864.88 cu. ft., volume of tank by calibration with water.

(19) Dry air per hour, lb. 6.668
Item 18 divided by Item 3 and multiplied by 60 min. per hr.

(20) Moist air per hour, lb. 6.723
From the expression Item 19 × (1 + Item 7).

(21) Nozzle number 1

(22) Nozzle diameter, in. 0.08537
From measurements with micrometer slide and microscope.

(23) Nozzle area, sq. in. 0.00572
Item 22 squared and multiplied by 0.7854.

(24) Discharge factor, ψ 3.72

From the formula $G = \psi F \sqrt{p_1 d_1}$

in which p_1 = Item 4 × 0.4912 × 144
d_1 = Item 8
F = Item 23 ÷ 144
G = Item 20 ÷ 3600

(25) Expansion exponent, n 1.364
Corresponding to Item 24 from Equation [**24**].

(26) Weight flow efficiency, per cent 0.9578
Item 24 divided by discharge factor ψ corresponding to frictionless flow.

DISCUSSION

S. A. Moss. The author has handled some difficult experiments in an admirable way. Those of us who have previously calibrated nozzles can appreciate the methods for handling the temperature and humidity of the air within the tank.

There are tabulated in the last column of Table 2 the real final results of the paper which are called "weight-flow efficiencies." The writer understands these to be rates of actual flow of moist atmospheric air to theoretical adiabatic flow of a perfect gas of the same density. This has customarily been called a "velocity coefficient" or "discharge coefficient" (with stream-line nozzles). It is customary to call the square of this number the "efficiency," since the square approximates an energy ratio.

It is to be noted that the general formulae which are given may be expressed in many other forms. (See *American Machinist*, Sept. 20 and 27, 1906.)

An interesting but very minor point is the effect of moisture upon the coefficient k of adiabatic expansion. It can be shown that a mixture of practically perfect gases expands with values of specific heats which are computed in proportion to the respective weights of the individual gases. (See *Sibley Journal of Engineering*, May 1905.)

If the author has any values showing the exact amount of this small effect, it would be interesting to give them. It is probably negligible compared with the direct effect of moisture on the density. It is to be noted that even the latter effect is small. The chart in the paper of Mr. Carrier cited, with the usual value of about half of full humidity, shows a difference in density of about one per cent. The flow varies with the square root of the density, so that flow would be in error about one-half of one per cent if computed on a basis of dry-air density, neglecting humidity for all days except very humid ones.

The values of the weight-flow coefficient or velocity coefficient average very close to 0.97 except in the case of the very smallest nozzle where the value is about 0.94. The value of 0.936 for the three-quarter-inch nozzle is undoubtedly a wild point. It has customarily been assumed that this coefficient varies with diameter on account of the form of the friction equation, given by the author as Equation [8], but the decrease with small diameters has always been supposed more rapid than the author's values indicate. Many experiments with orifices larger than 3 or 4 inches show values of

0.99. There is no doubt but that the author's values give valuable points on a curve of velocity coefficient against flow.

One hesitates to find fault with such an admirable paper, yet the writer feels that all the mathematical work connected with the average expansion exponent n is incorrect. A gas in a nozzle does not expand in an average way as indicated by Fig. 2, and the author's n has no real meaning for this case. The main body of gas expands according to the strict adiabatic law, and there is a thin skin along the nozzle walls which follows the author's friction Equation [8], and has reduced velocity. This is shown by impact tube traverses,[1] and by the fact that the actual jet in an orifice in a thin plate has the full adiabatic velocity.

One of the consequences of the author's n is the fact that if it existed, the energy efficiency of a nozzle would vary with the pressure ratio, as shown by Equation [19]. An enormous amount of work with steam turbine nozzles by many investigators shows that this is not the case.

The use of a velocity coefficient, as given in the last column of Table 2, has for many years been a satisfactory way of handling the flow through nozzles, and the author gives no good reason for changing the practice. Hence his very valuable contribution will be made more valuable if he uses the customary method and omits the non-essential n for which he gives no justification.

G. L. E. Kothny. The author in presenting this paper has made a very valuable contribution to the Society, and should be congratulated for his work, which is very much appreciated. The C. H. Wheeler Manufacturing Company for many years used air orifices of a similar type to those shown in the author's paper for testing air ejectors and dry vacuum pumps. This method is considered the most accurate for determining the actual air-handling capacity of these pumps.

Since the author has done considerable research work, the results of which are unquestionable, it is suggested that the Power Test Code Committee of the Society adopt standard types of air orifice, and that the efficiencies of vacuum pumps should be based on air orifice tests.

Sanford A. Moss. In connection with the suggestion made by G. L. E. Kothny that the Committee of the Society adopt some

[1] The Impact Tube, Trans., Am., Soc. M. E., 1916.

standard orifice, certain rules for orifices have been presented and undoubtedly a standardization will result.

Willis H. Carrier asked whether the formula for losses in the nozzles should be based upon friction or upon the elements of contraction. They had, he believed, a similar effect, but it was his understanding that in short nozzles of this type the most serious losses were introduced by the form of flow rather than by the resistance of the surface itself. The uniform proportions would seem to him to be required primarily to secure uniform coefficient of contraction, and uniform velocity coefficient, rather than uniform surface friction, which, he thought, would be a relatively small element entering into the nozzle coefficient.

R. C. H. Heck. Under the conditions to which Mr. Carrier refers, namely, with the low velocities and small pressure drops encountered in work with fans, velocity of approach will undoubtedly have an appreciable effect: relative magnitudes are here much like those met with in the flow of water. When, however, the pressure drop becomes relatively large, even from an initial pressure near to atmospheric, velocities in the nozzle will be so big that any ordinary variation in the small velocity of approach will have an almost imperceptible effect.

John Hodgson.[1] In this set of experiments which were obviously carried out with great care, and for which the nozzles were obviously made as nearly geometrically similar as possible, the experimental discharge coefficient or flow efficiency varies by about 2½ per cent. In his own work, the writer always uses geometrically similar square-edged orifices for standard measurements as these are easy to reproduce with accuracy.

The writer finds that such orifices can be reproduced so as to give coefficients of discharge that agree within one-half of one per cent, and that calibration of a small orifice, say, one inch in diameter, can be applied with accuracy (i.e., to within plus or minus ½ per cent) to orifices in the neighborhood of 30 in. in diameter.

The writer's tests on shaped nozzles fully bear out the author's results. It is impossible to predict the coefficient of discharge of such a nozzle within 2 or 3 per cent.

[1] George Kent, Limited, Biscot Road Works, Luton, Eng.

THE AUTHOR. With reference to the discussion of S. A. Moss, the "weight-flow efficiency" in Table 2, is obtained as indicated in item (26) of the Appendix. It is the ratio of the flow under actual conditions to the flow that would occur without friction but with a fluid of the same density. The variation of the exponent k with the moisture in atmospheric air is negligible; for example, with dry air it is 1.40 and with moist air having a ratio of water vapor to dry air of 0.02, it is 1.398, assuming $k = 1.30$ for the water vapor. Table 6 has been added to show the effect of moisture on the flow of air through a nozzle. While the total mass flow is affected only to one-half of one per cent as stated by Mr. Moss, the actual weight of dry air flowing is reduced to two per cent with humid air as compared with moisture-free air.

In the use of the exponent n the theory of Zeuner has been followed, but without making the erroneous assumption that the velocity in the orifice does not exceed the acoustic velocity. With small orifices, such as the ones experimented on, friction against the side walls has a greater effect upon the central core than with orifices 4 inches in diameter and larger, investigated by Mr. Moss. With an orifice in a thin plate, this frictional resistance is practically zero, so that $n = k$. But with the nozzles in steam turbines, it is customary for designers to assume an energy efficiency of 90 to 95 per cent, thus practically accepting the theory given to represent average conditions. If n were equal to k, the efficiency of all steam nozzles would be 100 per cent, as indicated by Formula [**10**]. The efficiency as given by this formula varies but little with the pressure ratio. In fact, substituting $n = 1.375$ and $k = 1.4$, η varies only from 0.955 to 0.967 with a variation of p/p_1 from 0.9 to 0.1. It is admitted that the theory submitted is not exact and represents only approximately the average conditions in the flowing stream; however, it is believed to represent the conditions more closely than any other theory at present available.

No. 1766

EFFECT OF FITTINGS ON FLOW OF FLUIDS THROUGH PIPE LINES

INCLUDING CHART FOR FLOW OF SATURATED AND SUPERHEATED STEAM IN PIPES

By Dean E. Foster,[1] Tulsa, Okla.
Member of the Society

The tables and chart developed in this paper afford a means of making proper allowance for resistance offered by fittings to the flow of liquids and gases through pipe lines. The tables which are based upon equations developed from formulæ giving flow of water and of high-pressure steam through pipes and the resistances offered to flow by fittings, give equivalent lengths of standard pipe to allow for various screw fittings in conduits carrying non-viscous liquids, steam, air or gas. The chart furnishes a means of rapidly solving problems involving the carrying of either dry saturated or superheated steam. Examples are given showing the use of the tables and chart.

IN the layout of piping in power plants, heating systems, refineries and other projects, the engineer is frequently confronted with the problem of making proper allowance for the resistance offered by fittings to the flow of liquids and gases. A number of formulæ and tables have been published giving such information for globe valves and elbows, but none for fittings in general. Konrad Meier, Mem. Am.Soc.M.E., in his excellent treatise entitled The Mechanics of Heating and Ventilation, has given the subject of the effect of fittings on resistance to flow a very careful study and has presented data thereon in the form of seven charts. In his work of the past several years the writer has found these charts to be very useful in the design of piping for uniform distribution of liquids and gases and in the selection of equipment for producing the necessary force for moving fluids, his only objection to these charts being the time

[1] Consulting Petroleum Engineer.

Presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

required to make analyses with them. After some study of the equations used in their construction it was found to be a comparatively easy matter to formulate the following relations between the frictional resistance of pipes of various sizes and the various screwed fittings commonly used, and from these relations to establish tables showing the lengths of pipe equivalent in resistance to that offered by fittings.

2 For the flow of water through pipe lines Mr. Meier uses the formula —

$$H_f = \frac{0.0257}{2g} \times v^{1.86} \times \frac{L}{d^{1.25}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

where H_f = loss of pressure in ft.
v = velocity in ft. per sec.
d = diameter in ft.
L = length in ft.

Equation [1] agrees within 5 per cent with Gardner S. Williams formula, which is so widely used in this country.

3 For resistance offered to flow by fittings, Mr. Meier gives the following:

$$H_r = 1.38r \times \frac{v^{1.86}}{2g} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [2]$$

where H_r = ft. of pressure loss per fitting
r = a factor of resistance whose value depends upon the shape of the fitting
v = velocity of the fluid in ft. per sec.

4 By combining Equations [1] and [2] we obtain

$$L_e = 53.75r \times d^{1.25} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [3]$$

where L_e represents the number of feet of pipe equivalent in resistance to a fitting and the other factors as before. Table 1 has been made up from this equation and may be used for water, crude or refined petroleum oils of 32 deg. B. gravity or lighter, or other non-viscous liquids.

5 For the flow of high-pressure steam through pipes Mr. Meier uses the equation:

$$P_f = \frac{W}{144} \times 0.0257 \times \frac{v^{1.95}}{2g} \times \frac{L}{d^{1.2}} \quad . \; . \; . \; . \; . \; . \quad [4]$$

where P_f = pressure in lb. per sq. in. to overcome frictional loss.

6 For resistance offered to flow by fittings he gives

$$P_r = \frac{W}{144} \times 1.12r \times \frac{v^{1.95}}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

7 By combining Equations [4] and [5] we obtain

$$L_e = 43.7rd^{1.2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

TABLE 1 EQUIVALENT LENGTHS OF STANDARD PIPE TO ALLOW FOR VARIOUS SCREW FITTINGS IN CONDUITS CARRYING NON-VISCOUS LIQUIDS
(Calculated from Equation [3])

Nominal Pipe Size, in.	Actual Inside Diameter, in.	Form of Resistance: Gate Valve	Long-Sweep Elbow or on Run of Standard Tee	Medium-Sweep Elbow or on Run of Tee Reduced in Size ¼	Standard Elbow or on Run of Tee Reduced in Size ½	Angle Valve	Close Return Bend	Tee through Side Outlet	Globe Valve
Factor of Resistance		0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00
½	0.622	0.335	0.442	0.56	0.89	1.20	1.34	1.78	2.68
¾	0.824	0.475	0.627	0.79	1.27	1.71	1.90	2.52	3.80
1	1.049	0.610	0.844	1.07	1.72	2.30	2.56	3.40	5.12
1¼	1.38	0.902	1.19	1.51	2.42	3.24	3.61	4.80	7.22
1½	1.61	1.09	1.43	1.83	2.92	3.92	4.36	5.79	8.72
2	2.06	1.49	1.96	2.50	3.99	5.36	5.96	7.92	11.92
2½	2.46	1.86	2.46	3.13	5.00	6.72	7.47	9.93	14.94
3	3.06	2.46	3.25	4.11	6.66	8.87	9.86	13.11	19.72
3½	3.54	2.92	3.80	4.91	7.84	10.53	11.70	15.56	23.40
4	4.026	3.44	4.53	5.77	9.22	12.37	13.70	18.28	27.50
4½	4.506	3.95	5.20	6.63	10.60	14.22	15.80	21.01	31.60
5	5.047	4.57	6.00	7.68	12.20	16.47	18.30	24.33	36.60
6	6.065	5.72	7.55	9.61	15.30	20.61	22.90	30.45	45.80
7	7.024	6.90	9.10	11.59	18.50	24.84	27.60	36.70	55.20
8	7.981	8.10	10.69	13.60	21.70	29.16	32.40	43.09	64.80
10	10.020	10.70	14.10	17.97	28.70	38.52	42.80	56.92	85.60
12	12.090	12.50	17.80	22.68	36.20	48.60	54.00	71.82	108.00

8 There is no reason to assume that the flow of gas or air through fittings should occasion losses differing from those caused by the flow of steam. Hence the writer assumes that the equivalents of Table 2 may be used for all three fluids.

9 To show the use of Table 2 an example has been taken of a 6-in. steam line 1000 ft. long, containing 5 gate valves, 3 angle valves, 20 standard tees and 10 standard ells. From Table 2 for steam, air and gas, these fittings are found to be equivalent to 332.10 ft. of 6-in. pipe, as shown on the top of following page.

5 Gate valves, each 4.81	24.05
3 Angle valves, each 17.35	52.05
20 Standard tees on run, each 6.35	127.00
10 Standard ells, each 12.90	129.00
Total allowance	332.10 ft.
Actual pipe length	1000.00
Equivalent total length	1332.10 ft.

TABLE 2 LENGTHS OF STANDARD PIPE TO ALLOW FOR VARIOUS SCREW FITTINGS IN CONDUITS CARRYING STEAM, AIR OR GAS

(Calculated from Equation [6])

Nominal Pipe Size, in.	Actual Inside Diameter, in.	Form of Resistance							
		Gate Valve	Long-Sweep Elbow or on Run of Standard Tee	Medium-Sweep Elbow or on Run of Tee Reduced in Size ¼	Standard Elbow or on Run of Tee Reduced in Size ½	Angle Valve	Close Return Bend	Tee through Side Outlet	Globe Valve
Factor of Resistance		0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00
½	0.622	0.031	0.41	0.52	0.84	1.12	1.25	1.66	2.50
¾	0.824	0 044	0 57	0.73	1.17	1.57	1.75	2.33	3.50
1	1.049	0.057	0.77	0.98	1.57	2.11	2.34	3.11	4.68
1¼	1.380	0.082	1.07	1.37	2.19	2.94	3.27	4.35	6.54
1½	1.610	0 098	1.29	1.64	2.63	3.52	3.92	5.21	7.84
2	2.067	1.32	1.74	2.23	3.55	4.77	5.30	7.05	10.60
2½	2.469	1.64	2.16	2.75	4.39	5.91	6.56	8.71	13.12
3	3.068	2.13	2.81	3.59	5.72	7.69	8.54	11.40	17.08
3½	3.548	2 53	3.34	4.26	6.80	9.10	10.13	13.50	20 26
4	4.026	2.96	3.90	4.97	7.94	10.65	11.84	15.75	23.68
4½	4.506	3.27	4.45	5.66	9.05	12.14	13.50	17.95	27.00
5	5.047	3.88	5.11	6.52	10.40	13.95	15.51	20.60	31.02
6	6.068	4.81	6.35	8.09	12.90	17.35	19.27	25.60	38.54
7	7.023	5.75	7.59	9.66	15.40	20 70	23.02	30.60	46.08
8	7.981	6.70	8.85	11 20	17.90	24.10	26.80	35 60	53.60
10	10 02	8.75	11.54	14.70	23.40	31.50	35.00	46.60	70.00
12	12.09	10.90	14.40	18.35	29.30	39.30	43.70	58.10	87.40

10 The chart for flow of steam, described in Pars. 15, 16 and 17, shows that with an initial pressure of 150 lb. and a total loss of 5 lb. pressure or 0.375 lb. per 100 ft. this line will transmit about 225 lb. of steam per min. If the effect of the valves and fittings had been ignored, the calculations would show a capacity of 260 lb. of steam per min. This would have indicated a flow of 11½ per cent more than the real capacity of the line. This error in capacity will be the same no matter what pressure drop is allowed through the line.

11 Use of Table 1 is illustrated by the solution of the following problem which assumes that a 4-in. line is to handle 26.2 cu. ft. of water per min. through a horizontal run of 50 ft., at which point the line rises at right angles 35 ft., thence runs horizontally 120 ft. to a tee heading a manifold consisting of five pieces of 4-in. pipe 12 ft. long connected by 4-in. tees, each delivering one-sixth of the total quantity of water. A centrifugal pump is to be selected and properly speeded to handle this quantity of water. The pump is to be located 10 ft. above the water surface.

12 Since the frictional resistance of pipe varies approximately as the square of the quantity of water passing through it, it will be convenient to combine the last 12 ft. of horizontal run with the first tee. This will give us 6 sections of 4-in. pipe each 12 ft. long and having a connecting tee. From Table 1 for liquids we find a 4-in. tee equivalent to 4.53 ft. of 4-in. pipe making each section the equivalent of 16.53 ft. in length. From Cox's tables for flow of water through pipe lines we find a loss of 2.563 ft. per 100 ft. for a 4-in. line handling 26.2 cu. ft. of water per min. The loss for the first section which carries all the water will be $16.53 \times 2.563/100$ or 0.424 ft. For the second section which carries $\frac{5}{6}$ of the total the loss will be as the square of the quantity carried, or $\frac{5}{6} \times \frac{5}{6} \times 0.424 = 0.295$ ft.

13 For the entire manifold the losses will be as follows:

Section 1 $(\frac{6}{6})^2 \times 0.424$	0.424
Section 2 $(\frac{5}{6})^2 \times 0.424$	0.295
Section 3 $(\frac{4}{6})^2 \times 0.424$	0.198
Section 4 $(\frac{3}{6})^2 \times 0.424$	0.131
Section 5 $(\frac{2}{6})^2 \times 0.424$	0.047
Section 6 $(\frac{1}{6})^2 \times 0.424$	0.012
Total loss for manifold	1.107 ft.

For the remainder of the line we have:

Suction line	10
300 diameters for pump	100
First horizontal run	50
Riser	35
Second horizontal run	108
Three 4-in. standard elbows, each 9.22	27.66
Total equivalent length of pipe to Section 1 of manifold	330.66 ft.

And from the foregoing:

Loss on 330.66 ft., 2.563/100 per ft.	8.47
Loss through manifold	1.11
Total loss	9.58 ft.

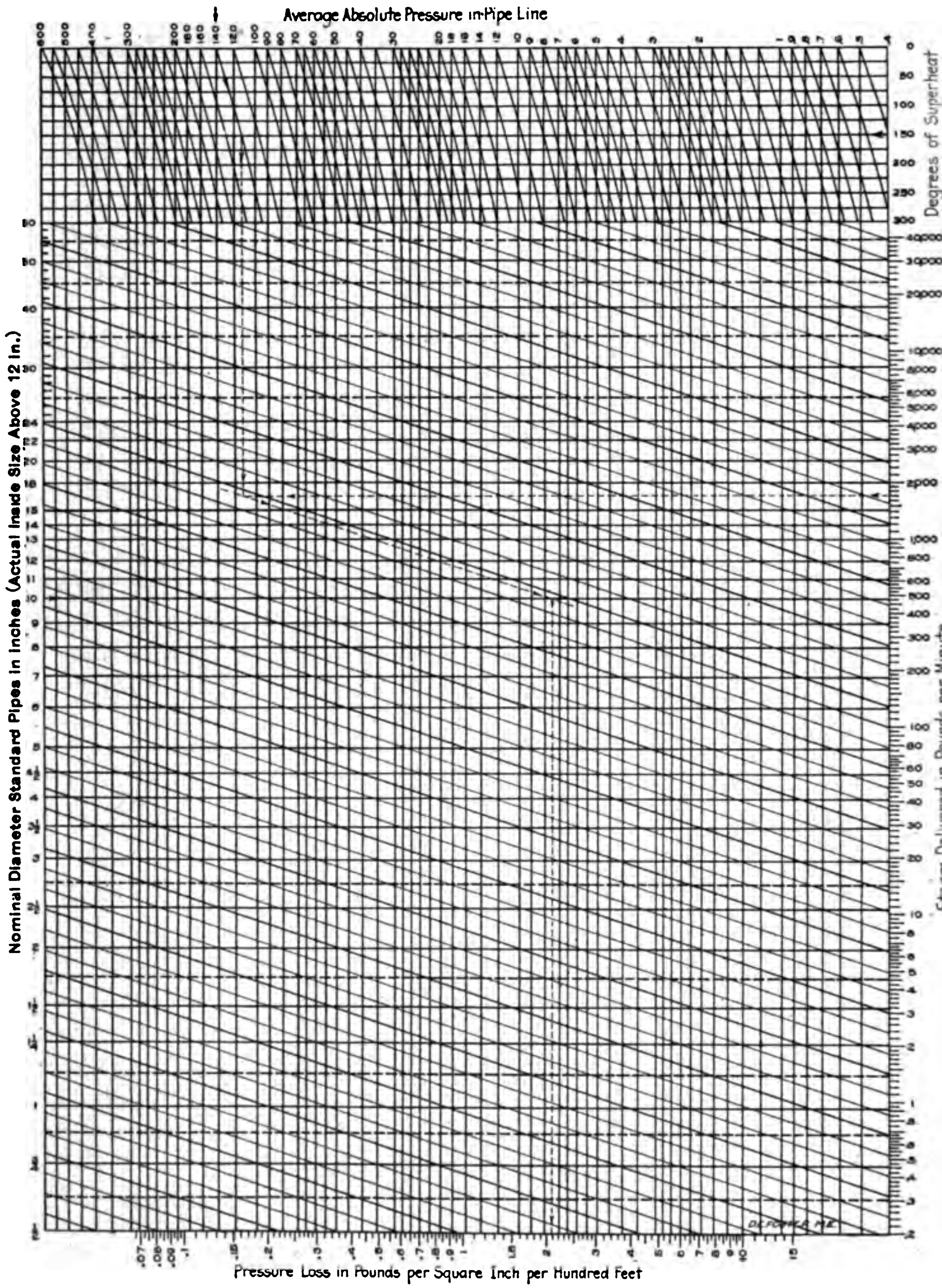

FIG. 1 GRAPHICAL SOLUTION OF BABCOCK'S FORMULA FOR FLOW OF STEAM IN PIPE LINES

14 The total pumping head then will be as follows:

Suction head	10.00
Elevation	35.00
Friction head	9.58
Total head	54.58 ft.

15 Many charts and diagrams for the quick determination of steam-pipe sizes have been published but they have all been constructed under the assumption that dry saturated steam was the fluid to be carried. The accompanying chart, Fig. 1, however, solves problems involving the carrying of either dry saturated or superheated steam. In order that its use may be clear the following case has been assumed, the pressure loss in pounds per square inch per hundred feet being desired:

Average steam pressure in line	140 lb. abs.
Superheat	150 deg.
Amount of steam to be delivered	1700 lb. per min.
Size of pipe	10 in. standard

16 In the solution shown in Fig. 1 the chart is entered on the horizontal line representing 150 deg. of superheat. This line is followed to the left until it intersects the curve representing 140 lb. pressure. From this intersection the dashed vertical line is followed down to its intersection with the horizontal dashed line representing 1700 lb. of steam. From this intersection the inclined dashed line is followed to its intersection with the horizontal line representing 10-in. pipe. The vertical thus determined gives the loss required as 2.08 lb. per 100 ft. Had a 12-in. pipe line been used, the loss would have been only 0.81 lb.

17 Fig. 1 can be used for the solution of any case where four of the variables are known and the fifth is to be determined. For the solution of problems involving dry saturated steam enter the chart at the top at the point representing the pressure assumed and follow a vertical line from this point to its intersection with a horizontal representing the desired quantity of steam. The remainder of the solution will be the same as outlined for superheated steam. If the part representing superheated steam were omitted, this chart would be similar to one constructed by H. V. Carpenter, Mem. Am. Soc. M. E., and published several years ago in *Power*.

DISCUSSION

H. H. Suplee. The paper is devoted entirely to the subject of the determination of the resistances offered to the flow of liquids and gases through various fittings of existing design, and as such it is a welcome contribution to technical literature. It should also serve to direct attention to the opportunity for improvement in the design of such fittings, in order that they should be made more efficient in this respect and that the losses due to their resistance should be minimized.

Experience during the past few years in connection with the resistance of the air to the motion of bodies of various forms has resulted in the development of the science of aerodynamics; and the use of the wind-tunnel has become generally accepted for the experimental determination of such resistances as occur in the various parts of airplanes. These tests have demonstrated the great importance of minimizing the production of eddy currents and similar forms of internal work, and the beneficial effects of "stream-lining," or "profiling," as the French call it, are fully appreciated.

It is well known that much of the resistance to the flow of fluids through fittings is due to the production of such eddy reactions, and the difference shown by the author between the resistance of a standard elbow and a long-sweep elbow indicates the extent to which this source of loss may be controlled.

Many years ago Professor Reuleaux suggested that a marked improvement might be made in standard fittings by removing an obvious cause of eddy currents, that is, by avoiding abrupt changes in cross section.

Thus, in a standard elbow, there is a sudden increase in diameter in the interior of the fitting over the size of the threaded portion, this being doubtless for convenience in tapping the thread. The effect of this change of section is the production of eddies, as shown in Fig. 2, and a consequent development of resistance to flow, quite analogous to the excessive resistance found under similar conditions in the wind tunnel.

Professor Reuleaux[1] suggested the modification in design of such an elbow fitting as shown in Fig. 3, in which the change in section is avoided and the hurtful resistances diminished. A slight clearance is necessary to facilitate cutting the thread, but this is so

[1] Reuleaux: Der Konstrukteur, p. 1018; Am. Ed., p. 251.

nearly filled by the pipe, and so soon filled up by sediment, etc., that the operative section is practically constant, and the occasion for the formation of eddies removed. The resistance of such an elbow is less than one-fifth that shown in Fig. 2, as computed for water flowing at a velocity of 2 metres per second, or about 400 feet per minute. Similar reductions may be effected in other fittings by using the same principle in their design.

Attention should be called to the fact that eddy losses in fittings are much greater than friction losses, properly so called, and that in all cases abrupt changes in cross section should be avoided, and stream-lined profiles introduced, if practicable.

F. M. VanDeventer. It should be particularly noted that the author's data apply only to standard weight pipe and screwed

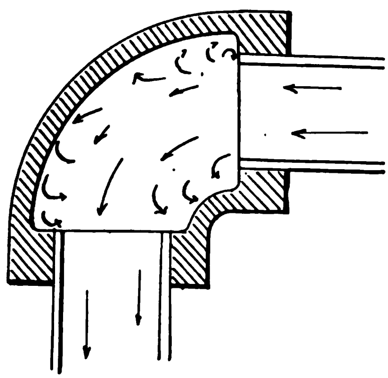

Fig. 2 Ordinary Elbow Fitting

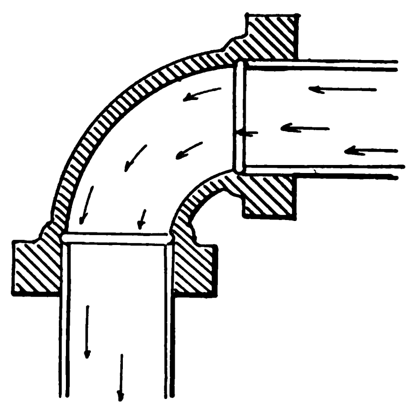

Fig. 3 Reuleaux Elbow Fitting

fittings. The amount of these two classes of material used in the modern power plant equipped for high-pressure and high-temperature steam is so small that it is almost negligible when compared to the amount of extra-heavy pipe and flanged extra-heavy fittings. For instance, in the author's sample calculation for a six-inch steam line 1000 feet long containing valves, tees, elbows, etc., surely few engineers would install screwed fittings in such a line, which is subject to the most extreme expansion and contraction strains; and if flanged fittings be substituted, the accuracy of the tabular data as applying to the problem is doubtful, although it is true that the tabular data may be used for extra-heavy *screwed* fittings with slight error. Most flanged fittings have a smooth bore, e.g. elbows and tees, and since they do not have a change in diameter such as the screwed variety have, there is not as much chance for eddying and other phenomena which produce pressure drop. This difference must be considerable; for the tables show that the pressure loss in a standard elbow is 16 per cent greater than in a medium-sweep elbow, and the writer is of the opinion that the difference between flanged and screwed connections would be more marked than the difference between standard and medium-sweep elbows. Are there any data available from which this correction can be obtained?

The pressure loss chart, as published by the author, is not interchangeable, and should not be used for extra-heavy pipe. The error resulting from its use would amount to − 10 per cent for 12-inch pipe and − 53 per cent for one-half-inch pipe, with an average error of − 28 per cent for all sizes from one-half to 12-inch inclusive. The accompanying form of the Carpenter chart, Fig. 4, is recommended, as it is universal in its application. The only essential difference between it and the Foster chart is that the single pipe size scale is replaced by three scales, one of which is laid off for standard weight pipe, another for extra-heavy pipe, and the third for actual inside diameters above 20-inch. The latter scale may be used for cast iron, horn welded, and similar kinds of pipe in such uses as exhaust steam lines from turbines to barometric condenser heads, etc., where large sizes of pipe are required; this being a frequent and important application of the pressure loss calculation. The standard weight scale from 1- to 20-inch and the extra-heavy scale from 1- to 24-inch are based upon the actual diameters and gages as regularly manufactured.[1]

Contrary to the natural conclusion, there are conditions under which the "formula and slide-rule" method of calculation is more speedy than the chart method, at the same time affording greater accuracy. Some time ago the writer was concerned with a comparison of the pressure loss characteristics of a complicated steam header layout with different sizes of pipe, for the purpose of determining the economical size. There were more than 250 separate pressure loss determinations in all, as the quantity of steam flow was changing at nearly every fitting. At first the Carpenter chart was used, the writer having appended a set of superheat correction lines almost identical with the author's, but this method was very tedious and tiring, requiring three distinct operations for each calculation, even after eliminating one operation by drawing in a

[1] Data from National Pipe Standards, National Tube Co.

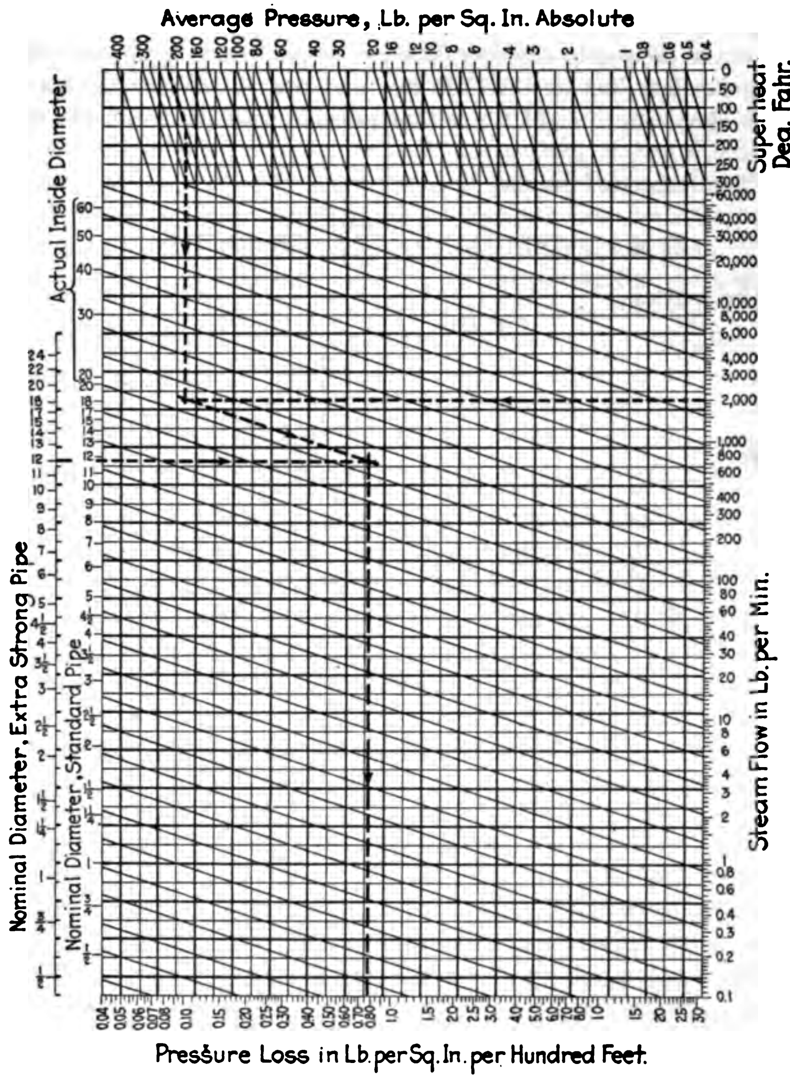

FIG. 4 CHART FOR FLOW OF STEAM IN PIPES

vertical line to represent the constant steam condition. By adopting the slide-rule method, which will be described, it was possible to reduce the process to one operation per calculation — merely moving along the hair line on the slide rule and reading off the figures — and a speed six to eight times that possible with the chart was obtained.

The Babcock formula,

$$p = .000131\left(1 + \frac{3.6}{d}\right)\frac{w^2L}{Dd^5} \text{ (formula } A\text{)}$$

may be written

$$p = \frac{w^2L}{D}F \text{ (formula } B\text{)}$$

or

$$p = w^2LVF \text{ (formula } C\text{)}$$

where

p = pressure loss, lb. per sq. in.
w = steam flow, lb. per min.
L = length of pipe or section, ft.
D = specific density of steam, lb. per cu. ft.
V = specific volume of steam, cu. ft. per lb.
Note: for superheated steam use D and V for the *superheated* steam, not the saturated value.
d = inside diameter of pipe, in.
F = a factor which is a function of the inside pipe diameter only, $= .000131\,\frac{d + 3.6}{d^6}$. See Table 3 for tabulated values.

An example will illustrate the method. Assuming a 12-inch extra-heavy steam header is to carry 250 lb. per sq. in. gage pressure and 150 deg. fahr. superheat; from the table, $F = 764.1 \times 10^{-12}$, and from the steam tables $V = 2.18$ or $D = .454$. Formula B then becomes: $p = w^2\ L \times 2.18 \times 764.1 \times 10^{-12}$. By letting p_{100} represent the pressure loss per 100 feet, and collecting the constants, the simplified form becomes: $p_{100} = w^2 \times 1.68 \times 10^{-7}$. Assuming further that a tabulation has been prepared which indicates the quantity of steam flow in each section, and provides a blank column for the pressure loss per 100 feet, the procedure is as follows: Set the left-hand index of the B scale of the slide rule under 1.68 on the left-hand section of the A scale; then run the hair line successively to the various rates of flow on the C scale and for each setting read the corresponding pressure loss on the A scale. Thus, set the line at 2000 lb. per min. and read .67 lb. per sq. in. per 100 ft., slide along to 2750 lb. per min. and read 1.27 lb. per sq. in., etc. The hair line can be moved with the left hand, leaving the right hand free to write down the figures without interruption. It is then necessary to run down the tabulation and multiply the pressure loss per 100 ft. in each section by the length, in order to obtain the net pressure loss in that section, but this procedure is also necessary when the chart is used. It is obvious that this method is many times faster than the chart when a long tabulation is to be calculated in which steam flow and length are the variables, but of course, for single set-ups and miscellaneous steam conditions and pipe sizes, the chart is much to be preferred.

The question may advisedly be raised as to the adaptability of the Babcock formula to superheated steam. The formula is a derivation of Unwin's $h = f\,\frac{2L}{D} \times \frac{v^2}{g}$, which is developed into $p = .04839\,K\left(1 + \frac{3.6}{d}\right)\frac{w^2L}{d^5D}$, in which K is a constant to be determined experimentally. Babcock assigned the value .0027 and R. C. Carpenter reported .0026, these values being for saturated steam. The question is, then, does this coefficient hold for superheated steam? There are two conditions that differ for wet and superheated steam, viz., surface condition and velocity. An engineer for a company which manufactures superheaters claims that the frictional resistance of superheated steam at a given pressure is actually less than that of saturated steam under the same pressure, due to the fact that the pipe surface is dry instead of being slushed with a film of water as is the case with wet steam. By considering the effect of velocity, the fallacy of this statement becomes so apparent that it does not warrant further consideration. It is possible that in the smallest sizes of pipe the surface condition might have an appreciable effect, but on the larger sizes where the mean hydraulic radius is greater, it is doubtful whether the difference in surface conditions for saturated and superheated steam is worth considering.

The effect of higher velocity of superheated steam is the important factor. The relation between friction and velocity is not a simple one, as a study of the Babcock formula will demonstrate. A glance at the formula indicates that if steam flow is the only variable, the friction varies as the square of the velocity. By plotting factor F against velocity it can be shown that if the pipe size be the only variable, then the friction varies approximately as the 2.6 power of the velocity. This may be explained by the smaller mean hydraulic radius and hence greater comparative frictional surface for small pipes. Further, a set of calculations with saturated steam, making steam pressure the only variable, indicates that friction varies as the first power of velocity. The explanation of this unexpected fact probably lies in the fact that for lower pressures, although the specific volume and hence velocity is greater, the density is correspondingly less, and the number of molecules of steam in contact with unit area of pipe is proportionately less. In view of the latter fact, it is the logical conclusion that, if the surface condition can be ignored, the friction for various amounts of superheat will vary as the first power of the velocity. The author's chart conforms to this conclusion. The correction for superheat may be made in either of two ways. (1) The formula or chart can be solved for saturated steam and the pressure drop so obtained multiplied by the ratio of the specific volumes of superheated and saturated steam at that pressure. A careful examination of the properties of superheated steam indicates that the increase in volume at any given pressure is very nearly 16 per cent for every 100 degrees of superheat. This is so nearly exact that the worst error due to this assumption is less about two per cent when referred to the final result — friction loss. This accuracy is well within that of the experimental data upon which the formula is based, as there is 4 per cent variance between Babcock's and Carpenter's values for K. Then, to use any chart or formula designed for saturated steam, it is only necessary to increase the pressure loss or velocity by 16 per cent for every 100 degrees of superheat to obtain a reasonably correct figure. (2) The Babcock formula lends itself to solutions for superheated steam without any changes or corrections by merely using formula B or C together with Table 3 and inserting for specific density D or specific volume V the actual value for the degree of superheat dealt with.

The deductions made in this study are necessarily theoretical, as the writer has been unable to find any experimental data on pressure loss for superheated steam. If such data are available, they would be a valuable addition to the present paper in settling the questionable factors. The same is true of pressure loss due to flanged fittings, non-return and triple-duty check valves for boiler outlets, etc. Experimentation and research along such lines should be encouraged, as modern engineering demands economic analyses

TABLE 3 STEAM FLOW — LOSS OF PRESSURE IN PIPES
FACTORS FOR SOLUTION OF FORMULA

Formula: $p = w^2 LVF$

Standard Weight Pipe			Extra Heavy Pipe		
Nominal Size, Inches	Actual Inside Diam.	Pressure Loss Factor, F	Nominal Size, Inches	Actual Inside Diam.	Pressure Loss Factor, F
½	0.622	9551. $\times 10^{-6}$	½	0.546	20.51 $\times 10^{-3}$
¾	0.824	1847. "	¾	0.742	3408. 10^{-6}
1	1.049	457.1 "	1	0.957	777.1 "
1¼	1.380	94.32 "	1¼	1.278	146.7 "
1½	1.610	39.14 "	1½	1.500	58.65 "
2	2.067	9.519 "	2	1.939	13.65 "
2½	2.469	3510. $\times 10^{-9}$	2½	2.323	4938. $\times 10^{-9}$
3	3.068	1047. "	3	2.900	1432. "
3½	3.548	469.4 "	3½	3.364	629.5 "
4	4.026	234.6 "	4	3.826	310.1 "
4½	4.506	126.9 "	4½	4.290	165.8 "
5	5.047	68.54 "	5	4.813	88.66 "
6	6.065	25.44 "	6	5.761	33.54 "
7	7.023	11.60 "	7	6.625	15.84 "
8	8.071	5531. $\times 10^{-12}$	8	7.625	7482. $\times 10^{-12}$
8	7.981	5870. "	9	8.625	3890. "
9	8.941	3216. "	10	9.750	2036. "
10	10.192	1612. "	11	10.750	1217. "
10	10.136	1659. "	12	11.750	764.1 "
10	10.020	1763. "	13	13.000	450.5 "
11	11.000	1080. "	14	14.000	306.2 "
12	12.090	658.2 "	15	15.000	213.9 "
12	12.000	684.4 "	17 OD	16.000	153.0 "
13	13.250	407.9 "	18 "	17.000	111.8 "
14	14.250	279.2 "	20 "	19.000	62.93 "
15	15.250	196.3 "	22 "	21.000	37.57 "
17 OD	16.214	143.0 "	24 "	23.000	23.54 "
18 "	17.182	105.8 "			
20 "	19.182	59.91 "			

of individual problems taking into account existing conditions, rather than following the "rule of thumb" methods of the past.

F. E. Giesecke. Tables, like those prepared by the author, showing the length of pipe equivalent in friction to various fittings, are very valuable to the engineer. The writer believes, however, that some of the data from which the author's tables were calculated and which were taken from Meier's Mechanics of Heating and Ventilation are not as accurate as they should be for such a purpose.

When Mr. Meier's book was written, no comprehensive tests of the friction of water in pipe fittings had been made and published;

at any rate the writer was not able to find any, so he conducted a series of tests in 1911 and 1912, at the Texas Agricultural and Mechanical College, and a second series in 1916 and 1917, at the University of Texas. The results of those tests, in so far as they apply to pipes and elbows, are published in University of Texas Bulletin No. 1759, The Friction of Water in Pipes and Fittings.

According to the tests, the friction of water in one foot of clean iron pipes is $0.00685 \frac{v^{1.77}}{d^{1.275}}$ feet; v being expressed in feet per second, and d in inches; and the friction in one standard short radius 90-deg. elbow is $0.0141 \frac{v^{1.96}}{d^{0.26}}$ feet; v being expressed in feet per second, and d being the diameter in inches of the pipe to which the elbow is screwed.

Since the exponents of v in the two cases are not alike, it is not possible to express the friction of one elbow in terms of that of a pipe of definite length, without specifying the velocity of flow at which the comparison is made.

Fig. 5 shows the lengths of pipe in feet equivalent in friction to one elbow, according to the results of the writer's tests. Comparing Fig. 5 with Table 1, it appears that for a 4-in. pipe, for example, the length shown in the table, 9.2 ft., is correct for a velocity of about 1½ ft. per sec., but not for any other velocity. For a 1-in. pipe, the length shown in the table, 1.72 ft., is correct for a velocity of about 3/10 ft. per sec.

The relative values assigned by the author to the friction in gate valves, long-radius elbows, short-radius elbows, tees, and return bends agree well with the results of the writer's tests, but those for a globe valve are only about one-fourth as high as those determined by the writer.

The writer's experimental values for various fittings were published in *Domestic Engineering*, Nov. 2, 1912. They may also be seen in Harding and Willard's Mechanical Equipment of Buildings, Vol. 1, p. 258; in Halsey's Handbook for Machine Designers and Draftsmen, p. 205, and in Cosgrove's Principles and Practices of Plumbing, p. 198.

Konrad Meier. The author's paper presents two tables for the equivalent lengths of standard pipe to allow for various screw fittings, calculated from a combination of the two equations, which form the basis of the charts for determining pipe sizes in Mechanics of Heating and Ventilating. These tables are easier of application than similar data in text books giving the equivalent in pipe diameter, since they save the conversion.

It seems proper to mention, however, that the formula $H_r = 1.38\, r \times v^{1.86}/2g$ is an approximation resorted to mainly for convenience. The resistance head is generally expressed by the Weisbach formula $H_r = r \times v^2/2g$, giving the resistance in terms of

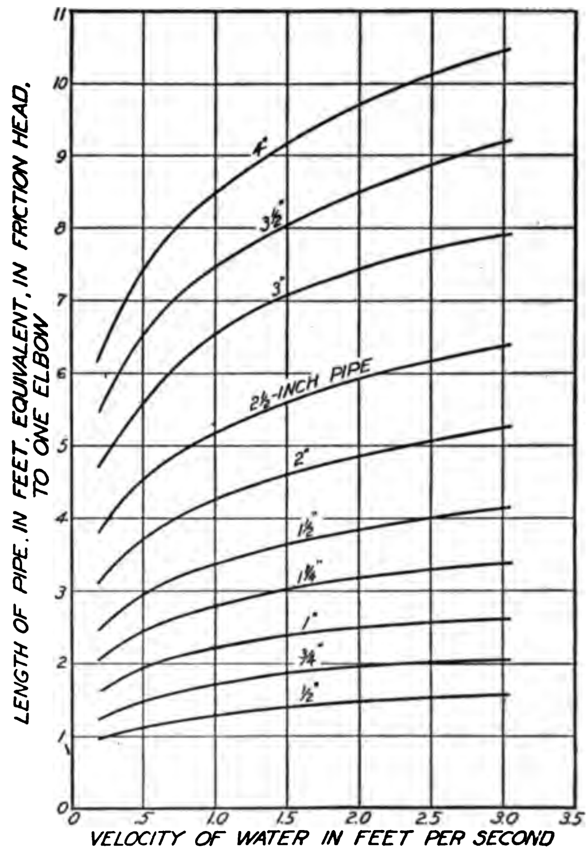

FIG. 5 LENGTHS OF PIPE IN FEET EQUIVALENT IN FRICTION TO ONE ELBOW.

velocity head. Weisbach found r to be constant, but this can be true only when the obstructing shape retains the same proportions for all diameters. In commercial fittings the recesses and changes of velocity are relatively greater for the smaller sizes, and recent experiments have shown how the coefficient r varies with the diameter, while the exponent of velocity remains practically at or near the square. The accurate calculation would therefore involve two charts or auxiliary tables. In practice this means the frequent neglect of local resistances, which really ought to receive equal consideration as the friction. In order to bring this about, the resistances were charted together with the friction losses on the assumption that they are subject to similar laws. The constant 1.38 will make 1.38 $v^{1.86}$ greater than v^2 for velocities up to 10 ft. and thus provides a margin of safety. The exponent 1.86 allows at least indirectly for the effect of diameter, since it gives relatively higher values for smaller pipes insofar as they involve lower velocities. For the same rate of friction, for instance of 1 ft. head per 100 ft. length, the velocity in a 1-in. pipe will be 1.1 ft. per sec., in a 3-in. pipe, 2.3 ft., and in a 10-in. pipe, 5 ft., for which the allowances amount to 35, 20 and 10 per cent respectively. This corresponds about with the variation of the true coefficient for fittings as near as we know it. The comparative table given herewith shows how the results from the author's tables and the charts of the writer agree with the latest research. They are, in any event, within the limits of error for such calculations.

TABLE 4 EQUIVALENT LENGTH IN FEET AND RESISTANCE HEADS OF SHORT SCREW ELBOWS ACCORDING TO VARIOUS FORMULAE FOR THE FLOW OF WATER

Diameter of pipe, in.	1				3				10			
Velocity of water, ft./sec.	½		2		2		4		4		8	
Equivalent length and resistance	L_e	H_r	L_e	H_r	L_e	H_r	L_e	H_r	L_e	H_r	L_e	H_r
1. Weisbach, r = .98	*	0.0038		0.06		0.06		0.24		0.24		0.96
2. Brabbée, r = 2 and 1.25	*	0.005		0.08		0.045		0.18		—		—
r = 1.3 and 0.75	*	0.0033		0.052		0.027		0.11		—		—
3. Foster, $L_e = 53.8r \times d$ 1.25	1.7	0.0038	1.7	0.051	6.7	0.051	6.7	0.185	28.7	0.185	28.7	0.70
4. Meier, r = 0.67	*	0.0038		0.051		0.051		0.185		0.185		0.70
5. Giesecke's chart	1.9	0.0033	2.5	0.058	7.4	0.045	8.1	0.18	—	—	—	—
6. Allen & Walker	2	0.0038	2.0	0.048	6.0	0.048	6.0	0.17	20	—	20	—
7. Schoder (Marks)	2.5	0.006	2.5	0.08	7.5	0.06	7.5	0.22	25	0.16	25	0.60

* Calculation of resistance head direct by formula or chart.

The Weisbach figures refer to a sharp turn (two intersecting cylinders), which may be about equivalent to the resistance of short elbows of small size, but would not be as good as those for large pipe. Brabbée gives figures for knees and bends of foreign standards, the radius of which is not stated. F. E. Giesecke, according to the tests published in the *Transactions* of the Am. Soc. H and V. E., 1917, finds the equivalent to be greater than the author has calculated them, but for the final result the discrepancy is practically offset by the smaller coefficient of friction used by Giesecke, which allows less for couplings and other incidentals. This applies also to Allen and Walker's figures, while Schoder makes considerable allowance all around for fouling, which is not always a factor. Since the equivalent increases as the friction decreases, the results will depend on the coefficient of friction or the exponents of v and d in use. Hence it is safer to calculate direct by formula or by chart giving both together.

While the losses of head by friction and the resistances of standard elbows are now fairly well known for various conditions and classes of pipe work, there is still considerable doubt about other common factors, such as branch tees of different kinds. The value for $r = 1.33$ given on the charts and taken by the author as a basis for equivalent pipe lengths must be regarded as an average applying only when there are no extreme changes of velocity. In unusual situations and for poorly selected or proportioned fittings the resistance may run up to several velocity heads. Likewise for bull-head arrangement. On the other hand, there are cases where the flow in one branch induces motion in another and r assumes a negative value. There is acceleration instead of retarding resistance. It makes also more or less difference, whether the streams are separating or joining. In fact, the losses of head at junctions is a study in itself. The essential facts should be made available, even though it is not really practicable to give or to use coefficients for all possible conditions and relations, as attempted by German professors. Since they could not apply to all styles and sizes of fittings, such arrays of figures are more likely to create confusion. Nor could any reliable data be given for welded branches, the resistance of which will always be uncertain. For the present it will be best to take conservative average values for r as far as known, and at the same time to avoid extreme changes in velocity and uncertain factors generally whenever possible. A standardization of practice in that sense would go far in meeting the difficulty.

It should always be borne in mind, that the exact value of coefficients is not the essential need for calculation of this kind, since they represent only one item among others that are also more or less uncertain. Of greater importance is the intelligent consideration of all the main factors entering into a problem and the proper selection and use of such data as are available. It will often mislead, for instance, to make arbitrary allowances for fittings as a percentage of the friction head, because that part of the resistance may be a negligible item in one case, while it becomes predominating in another and determines the pipe size. Hence the desirability of simplifying the calculation and presenting the data in the handiest form for practical service. To this end the average values for the resistance of fittings have been charted, so that the head may be obtained for each run together with that for friction. This feature of the charts is explained in the book referred to and mentioned also in Marks' Mechanical Engineers Handbook containing an extract for the flow of water and steam in the section for Heating and Ventilating.

E. H. Foster. The author should be commended for his valuable contribution to the Society's literature. It will be useful, at least, in calling attention to the losses due to friction, a point which is often overlooked in more or less carelessly designed pipe lines.

Theoretically, it would appear that there is a greater loss due to friction in pipes carrying superheated steam than in those carrying saturated steam. If the author's curve is used for designing pipe lines, the writer is convinced that an error will be made.

In certain cases which have come to the writer's attention, pressures at the end of a pipe line have been increased by installing superheaters. One was in the case of the assembling shop of a bridge works. A pressure gage, about 700 feet from the boilers in which steam was generated at a pressure of 100 lb., indicated 60 lb. before installing superheaters and 80 lb. thereafter, although it was determined that other conditions were the same and approximately the same amount of steam was used. Another case was of a 14-in. steam line, 1400 feet long, which showed an increase in pressure of 8 lb. after a separately-fired superheater was installed, other conditions being the same.

Carl V. Serbell pointed out that there was a pressure drop through the superheater itself which might easily be confused with the frictional resistance existing subsequently in the pipe line if the pressure were read at the exit of the boiler, rather than at the exit of the superheater.

Sanford A. Moss. In the table given by the author there seems to be a remarkable increase in the equivalent length of pipe corresponding to a fitting, as the pipe size increases. Will not the author explain the season for this increase?

Leo Loeb. In connection with extra-heavy fittings, it should be borne in mind that the usual installation of pipe line does not contain them. Thus in certain high-pressure line there may greater frictional losses due to the use of such fittings, but the number of such installations is restricted, while in the smaller sizes they are used for drip lines in which frictional losses are of less importance.

E. M. Ferris spoke of the practice of using high-pressure boilers and reducing valves in heating installations, thus increasing the velocity of the steam, using smaller pipes and saving greatly in the cost of piping. In such an installation the loss in pressure due to friction was unimportant as the pressure was subsequently lowered by a reducing valve.

The Author. It is gratifying that the paper has had the desired effect in bringing out a considerable amount of discussion. It is the new data which will be brought to light by the paper which is its chief justification.

Attention has been drawn to the difference between long-sweep and standard elbows, and it is not amiss to call attention to the fact that the effect of increasing the radius of the elbow is to reduce the frictional resistance through the elbow up to a certain point, after which the reduction ceases and there comes a point where a further increase in the radius of the elbow will increase the frictional loss. This has been brought out in at least two instances. The first was a report of Gardner S. Williams in connection with the Detroit Water Works system, several years ago, when he determined that with the same length of pipe the loss was greater with a long curvature in the pipe than with a short curvature. The reason is that the turning creates a greater velocity on the outside of the curve of the pipe than on the inside, and the disturbance persists for a greater length of time in pipes with curves of long radius than in pipes with curves of short radius.

The same fact was brought out by some experimental work by Professor Geo. W. Davis, done at the University of Wisconsin.

In answer to the question by Sanford A. Moss, the loss in a small pipe with fittings is due principally to friction in the pipe itself, while that in an elbow is due to eddy currents. The friction loss for a given velocity in a large pipe decreases inversely as approximately the 1.2 power of the diameter of the pipe, whereas the loss in the fitting remains the same.

Attention was also called to the relative diameters of the pipe and the fitting. In most fittings the diameter is larger than in the pipe. There is the so-called drainage type of fitting in which the inside diameter is the same as that of the pipe. The frictional loss in this fitting is less than in other fittings.

Professor Giesecke criticizes the paper because there is given no definite relation between loss in fittings and equivalent pipe lengths. It is impossible to formulate the relation between loss in fittings and equivalent pipe length, because the loss through fittings follows one law and the loss through the pipe another. Some allowance must be made, and for rapid determinations the work presented in the paper is satisfactory. Where close calculations must be made, it would be well to revert to the fundamental laws for the flow of fluids through pipes and fittings and to calculate each separately.

F. M. Van Deventer refers to some work on extra-heavy and double-extra-heavy pipe on which the author is uninformed. The author did not deem it of sufficient importance to include these grades of pipe in his paper.

If Mr. Van Deventer will take into consideration the re-location of pipe size, due to the actual inside diameter of the extra-heavy pipe, the chart will check correctly.

The author does not believe that the Babcock formula for the flow of steam through pipe lines is correct. Much must be done before the true loss in the flow of steam through pipes is known. The formula assumes that the losses vary as the square of the flow, whereas it is fairly certain that, as with water, the exponent is approximately 1.87, 1.90 or 1.96.

In answer to a question by A. A. Adler, the author replied that his chart did not take into consideration the pressure drop in fittings entering or leaving another main, because these losses depend upon the pressure drop occasioned by the division into streams or the amalgamation of streams, depending not only upon the character of junction and type of fitting, but also upon the relative size of the streams.

In the tables presented in this paper, reference has been made only to losses in screwed fittings. Mr. Suplee and Mr. Van Deventer both call attention to saving in pressure losses to be made by use of screwed drainage fittings and flanged fittings. Konrad Meier suggests using three-quarters of the screwed-fitting loss when making calculations for flanged fittings, and the same of course would apply to calculations for drainage fittings. The actual loss would be less probably than three-fourths of the screwed-fitting losses, but the error of this assumption would be on the side of safety.

In further answer to Mr. Van Deventer's discussion the sizes of standard pipe up to twelve inches has been located according to their actual inside diameters and it was the intention of the author that users of extra-heavy pipe should make due allowances in entering the chart for the smaller size of this pipe as compared with standard pipe.

No. 1767

THE CONSTITUTION AND PROPERTIES OF BOILER TUBES

By Albert E. White,[1] Ann Arbor, Mich.
Non-Member

The investigation recorded in this paper was carried out because of defects which developed in tubes of the boilers at the Park Place Heating Plant of The Detroit Edison Company requiring replacement of the defective tubes and consequent shutting down of the boilers. Investigation was made of the question of water softening for the prevention of scale, changes in baffling for the better distribution of the heat in the boilers, and of the constitution and properties of the metal of the tubes. The paper relates mainly to the third part of the investigation.

The author believes that tube failures, aside from those due to imperfect heat transmission that is mainly the result of scale, are due to: (1) Tube brittleness resulting from absorption by the metal of hydrogen and usually attributable to faulty boiler-feedwater treatment; (2) blowholes or other imperfections in the metal; (3) recrystallization of the metal.

The paper goes into detail only on the last two causes, evidence regarding which is presented in the form of photomicrographs.

All of these conditions encountered lead the author to suggest that tubes with a carbon content of from 0.30 to 0.35 per cent would have better durability than tubes with a carbon content of from 0.08 to 0.18 per cent.

DURING the winters of 1913–1914 and 1914–1915 considerable difficulty was experienced by the Park Place Heating Plant of The Detroit Edison Company in maintaining continuity of boiler operation. This difficulty arose because of the frequent shutdowns necessitated when boiler tubes bore evidence of being or becoming defective, requiring in consequence the temporary closing down of the boiler or boilers until the tubes in question could be replaced. Tubes in the front bank of the boilers were particularly prone to develop defects, and since this condition was experienced in its

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most aggravating form at a time when the boilers were most needed, namely, during the various cold snaps of the winter, investigations were started so that suitable steps might be taken to combat the trouble.

2 The necessity for obtaining relief was emphasized in June 1915, when No. 7 tube in the third row of No. 7 boiler at the Park Place Heating Plant let go. The appearance of this tube after the accident is given in Fig. 1. This tube failure made necessary the rebuilding of most of the boiler and especially the replacement of nearly all of the tubes in the boiler. Fortunately no injuries were sustained by any of the men in the plant. Also, the failure occurred at a time of the year when the plant was not operating at full load, so that the customers were not subjected to any inconvenience through lack of steam.

3 The accident, however, indicated the need for prompt reilef and investigations were started along three lines: (*a*) water softening, (*b*) rearrangement of baffling and (*c*) considerations relating to the composition and constitution of boiler tubes.

WATER SOFTENING

4 With reference to water softening, it was noted that practically all of the tubes which were replaced showed a thin but tough scale on the water side. This scale was calcium sulphate and was thin simply because the quantity of this salt in the boiler feedwater was relatively small and in plants operating under normal load conditions would have warranted no consideration. In these boilers, however, which were of the Stirling water-tube type of 750 hp., the long periods of cold often necessitated a load averaging 80 per cent above the nominal rating for periods extending at times into as many as six days. These load conditions, while not apparently unusually exacting, were in reality very severe when cognizance is taken of the fact that 98 per cent of the boiler feedwater was raw water drawn from the city mains.

5 There was no question but that the thin scale seriously affected heat transmission and was one of the contributing factors leading to tube failure. Much research work was done on this phase of the problem and it resulted in a decision to treat all of the water in the plant with soda ash. Since this treatment has been in use practically all of the insolubles are caught in the live-steam purifier. There is no scale in any of the tubes, although there are flocculent

particles of insoluble calcium carbonate circulating with the water in the boilers. This product, together with the other minor insoluble ones that may be present and the soluble salts, especially sodium

FIG. 1 BURST TUBE FROM NO. 7 BOILER, PARK PLACE HEATING PLANT, DETROIT EDISON COMPANY

sulphate, are kept down to harmless percentages by frequent boiler blow-offs. In this connection the dionic tester is used as a guide, experience indicating that when it reads under 1800 no fear need be had of trouble resulting from foaming and priming.

CHANGES IN BAFFLING

6 Steps were also instituted with reference to a study of the baffling. At the time when the trouble was most pronounced the front baffles were between the first and second rows of tubes. This arrangement had been adopted since it was believed that it would produce higher temperatures with consequent fuel economy and abatement of the smoke nuisance.

7 The investigations showed that most of the tube replacements were from the front row. For the purpose, therefore, of distributing the radiant heat among a greater number of tubes, hoping in this way to increase tube life without, however, carrying the same to such an extent as to decrease the fuel economy or run into the smoke troubles coming from incomplete combustion, the front baffling was changed so as to lie beween the second and third rows rather than between the first and second rows. This arrangement has been most beneficial. There has been no decrease in fuel economy nor increase in smoke nuisance and tube replacements have been materially lowered.

CONSTITUTION OF TUBES

8 Under the third line of investigation listed above —the composition and constitution of boiler tubes — the first step was a study of the constitution of the metal in the tubes. This was undertaken for the purpose of ascertaining the effect of service conditions upon the types of tubes now employed. The metal in a very considerable number of tubes was examined metallographically, the etching mediums being the common nitric acid and the so-called Rosenhain. This latter deposits copper over the surface of the metal, giving an even coating when the distribution of the metalloids is uniform; but when such is not the case, disclosing minutest segregations and imperfections most clearly by variations in the depth of the plating.

9 In Figs. 2 to 7, inclusive, are reproductions of six photomicrographs showing the structure of three pieces of boiler-tube metal after having been etched with nitric acid and with the Rosenhain medium. Figs. 2 and 3 represent the average of the best sections of the several hundred that were examined; Figs. 4 and 5, the average of the majority of the sections; and Figs. 6 and 7, the average of the poor sections.

10 In Figs. 2 and 3 it will be noted that the etching is uniformly distributed, indicating that the constituents are evenly dispersed throughout the metal. It is too much to hope for a 100 per cent production of metal of this quality, but the more nearly this can be approached, the less chance there will be of failure resulting from imperfect stock.

11 In Figs. 4 and 5 a clearly defined ghost line resulting from phosphorus segregation is brought out, although it is more apparent in the one resulting from etching with the Rosenhain medium than in the one resulting from etching with nitric acid. In the former, also, there are strikingly evident cavities, or segregations which the writer has detected in many of the samples he has examined. He considers this condition most serious, but will defer the discussion of this to a later portion of the paper.

12 In Figs. 6 and 7 the defects are not as evident in the sample etched with nitric acid as in the one etched with the Rosenhain reagent. Both structures, however, came from the same section of the same piece so the difference is due more to the character of the reagents than to differences in the specimen. The wide ghost line and the numerous cavities, or segregations, are most emphatically repugnant to those desirous of clean stock and would make one fearful of consequences if too great a tonnage of this class of material were placed in our boilers.

CAUSES OF TUBE FAILURE

13 On the completion of this preliminary survey attention was directed to the causes of tube failure. Excluding that due to imperfect heat transmission resulting from scale, the writer believes that the principal causes can be listed under the following heads:

(*a*) Failure due to tube brittleness resulting from absorption by the metal of hydrogen and usually attributable to faulty boiler-feedwater treatment

(*b*) Failure due to blowholes or other imperfections in the metal

(*c*) Failure due to recrystallization of the metal.

14 *Tube Brittleness Resulting from Hydrogen Absorption.* The first of these causes, namely, that due to tube brittleness resulting from the absorption by the metal of hydrogen, will not be developed in this paper. The facts are outstanding that contained hydrogen in metal makes it extremely brittle. The facts are further outstanding that certain types of water improperly treated, or all water ex-

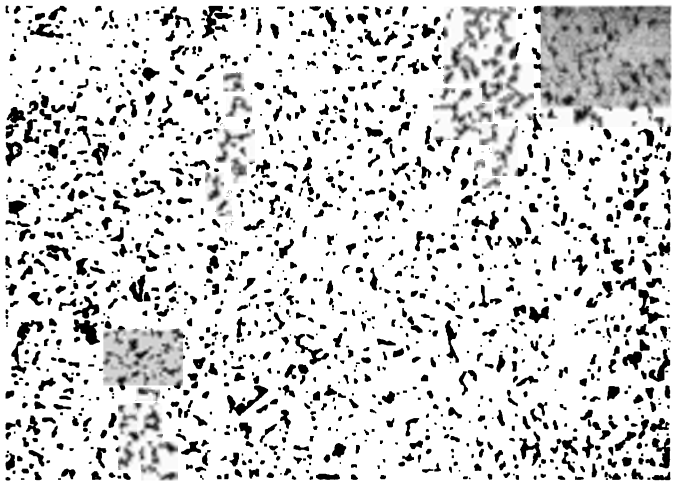

Fig. 2 Etched with Nitric Acid

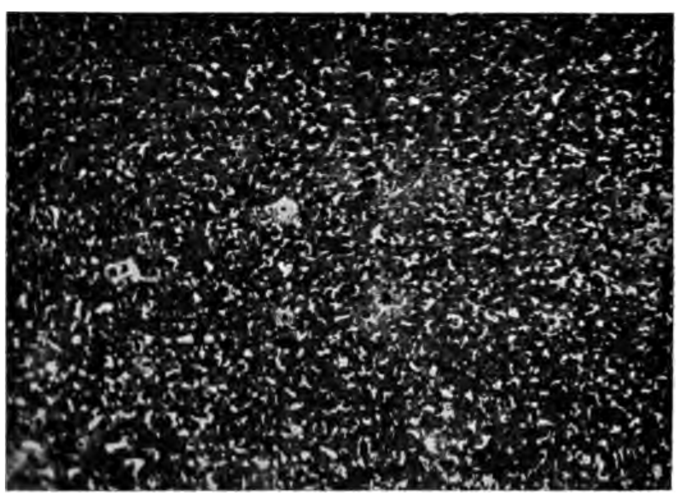

Fig. 3 Etched with Rosenhain Medium

FIGS. 2 AND 3 SPECIMENS SHOWING BEST STRUCTURE IN BOILER TUBES; MAGNIFICATION 100

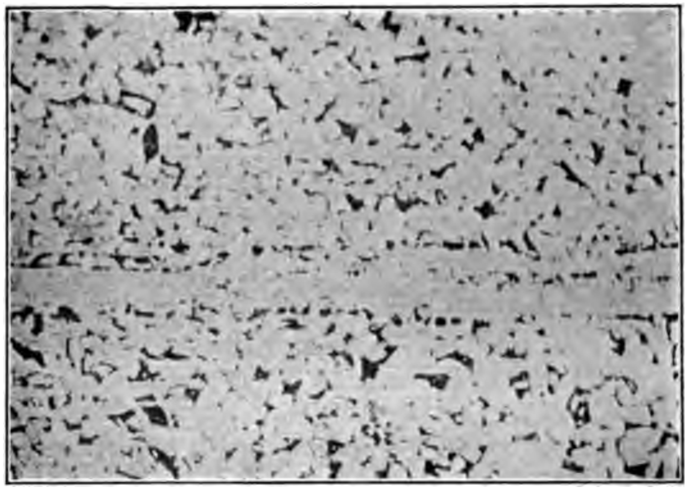

Fig. 4 Etched with Nitric Acid

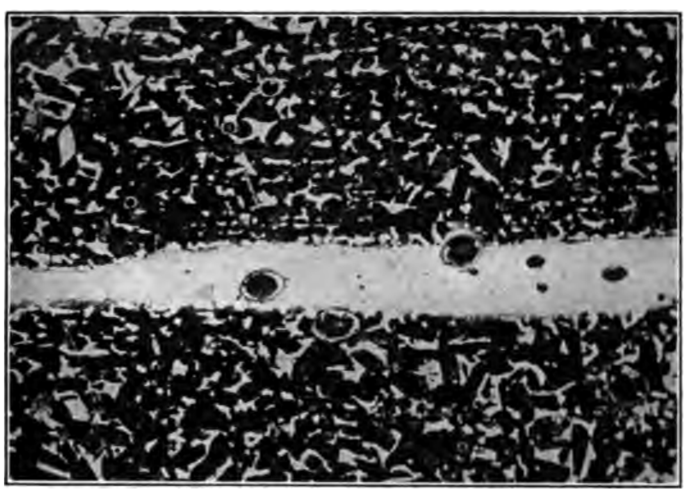

Fig. 5 Etched with Rosenhain Medium

FIGS. 4 AND 5 SPECIMENS SHOWING AVERAGE STRUCTURE IN BOILER TUBES; MAGNIFICATION 100

Fig. 6 Etched with Nitric Acid

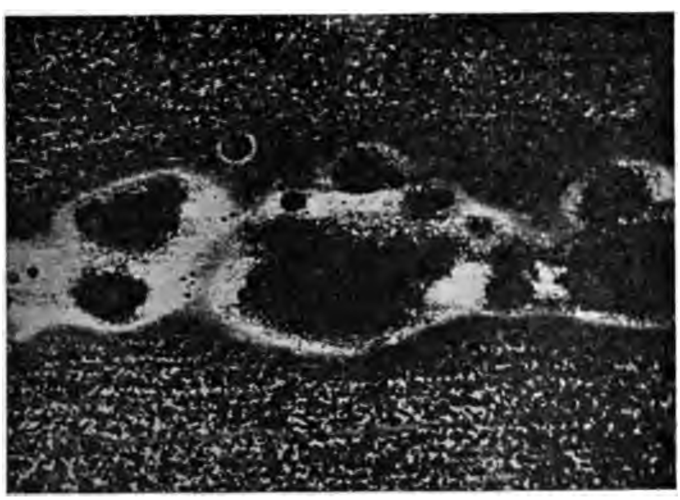

Fig. 7 Etched with Rosenhain Medium

FIGS. 6 AND 7 SPECIMENS SHOWING POOR STRUCTURE IN BOILER TUBES; MAGNIFICATION 100 cessively treated with certain types of boiler compounds, will cause the tubes to absorb hydrogen and become brittle. With intelligent treatment, however, there need be no cause for concern over tube failure from this source.

15 *Failure from Blowholes.* Failure from blowholes and other imperfections, the writer believes, should receive greater consideration in the future than has been accorded it in the past. The matter of course goes back to the steel mill and calls for greater emphasis on quality and less on tonnage.

16 Four different samples of metal are shown in Figs. 8 to 11, inclusive. Ghost lines are evident in all except Fig. 11, and in this latter section the crater or cavity is so large that the ghost lines, which visual examination showed were in the metal, could not be brought out in a photograph of the size submitted. Even a casual scrutiny of the photomicrographs reveals radiating lines issuing from the center of the cavities. This condition is manifest in all of the views. These lines indicate a lack of continuity of the metal, and should service conditions cause a tube to bag, and should a cavity of the types shown occur at the nipple of the bag, there is every reason to anticipate a bursting of the tube, with all of the attending dangers and expense.

17 *Failure from Recrystallization.* This matter merits much consideration. Recrystallization is accompanied by a marked decrease in the elastic limit and fatigue-resisting properties of the metal manifesting this phenomenon. It will occur if steel with a low carbon content which has previously been mechanically deformed at a temperature below the critical is later heated for a sufficient time to any temperature below that at the critical. The common composition for boiler tubes is such that this class of metal is especially susceptible to this phenomenon. Mechanical deformation to some degree is unfortunately assured by present-day methods of handling tubes; for at the mill tubes are straightened and in many cases actually brought to final size when below the critical temperature. Tubes are often bent during fabrication and erection and are universally rolled into the tube sheet when cold and the methods of cleaning tubes in service often employ forms of apparatus which produce local deformation by repeated hammer blows. Finally, the time and temperature conditions required for recrystallization are present.

18 Is it any wonder, therefore, that we hear of the development of brittleness in boiler material when the factors so essential for this

Fig. 8

Fig. 9

FIGS. 8 AND 9 EXAMPLES OF BLOWHOLES; MAGNIFICATION 100; ETCHED WITH ROSENHAIN MEDIUM

Fig. 10

Fig. 11

FIGS. 10 AND 11 EXAMPLES OF BLOWHOLES; MAGNIFICATION 100; ETCHED WITH ROSENHAIN MEDIUM phenomenon are so perfectly present: namely, metal of the right composition, metal that has been mechanically deformed at temperatures below the critical temperature, and suitable time and temperature conditions?

TIME-TEMPERATURE CRITERION FOR CRYSTAL GROWTH

19 Heating of deformed metal to temperatures approaching the critical, causes crystal growth in very short time periods. Corresponding growth occurs more slowly at lower temperatures, the time periods required for the same increasing very rapidly as the temperature to which the material is heated decreases. Values experimentally determined for temperatures from 550 deg. cent. (1022 deg. fahr.) to 675 deg. cent. (1247 deg. fahr.) for one set of conditions, are given by the following equations:

T (minutes) $= 8$ for 675 deg. cent. (1247 deg. fahr.)
T (minutes) $= 8 \times 3$ for 650 deg. cent. (1202 deg. fahr.)
T (minutes) $= 8 \times 3^2$ for 625 deg. cent. (1157 deg. fahr.)
T (minutes) $= 8 \times 3^3$ for 600 deg. cent. (1112 deg. fahr.)[1]

or, in general, for temperatures below 675 deg. cent. (1247 deg. fahr.),

$$T = 8 \times 3^n$$

where T = time in minutes
t = temperature in degrees centigrade
$n = (675 - t)/25$

20 That the normal method of handling boiler tubes results in mechanical deformation and that this metal as a result, under the proper conditions of time and temperature, will develop large crystals, is shown in Figs. 12 to 15, inclusive. Of these photomicrographs one was taken preceding a heat treatment that would quickly develop grain growth and three after special treatment for the quick development of grain growth at temperatures below the critical. A comparison of the grain sizes of the four samples, taking into account

[1] Recrystallization as a Factor in the Failure of Boiler Tubes, White and Wood. Proc. American Society for Testing Materials, vol. xvi, pp. 82–116.

the magnification employed in the case of each one, indicates clearly that the tubes have been sufficiently deformed so that they respond to the laws of grain growth when the proper conditions for this development are present.

PHYSICAL TESTS

21 Both tension tests and fatigue tests were made to determine the effect of coarse grains on the physical properties of the tubes. In Table 1 are presented the results of tension tests and in Table 2 the results of fatigue tests. In each of the tension tests three samples per set were employed: the first on metal as received, the second on metal annealed for 10 minutes at 950 deg. cent. or 1742 deg. fahr. and then cooled in the furnace and the third on coarse-grained metal produced by stressing all of the test specimens the same amount

TABLE 1 RESULTS OF TENSION TESTS OF BOILER TUBES

Sample No.	Treatment	Elastic Limit, lb. per sq. in.	Tensile Strength, lb. per sq. in.	Ultimate Strength, lb. per sq. in.	Elongation in 6 in., per cent	Reduction of Area, per cent
1	As received	33,190	55,250	42,300	13.30	56.1
2	As received	35,650	53,400	41,500	14.50	55.2
3	As received	34,410	54,700	42,500	15.30	54.8
	Average	34,420	54,450	42,100	14.37	55.4
4	Annealed	22,550	51,620	36,250	21.66	54.5
5	Annealed	22,100	50,000	38,150	18.32	54.4
6	Annealed	20,450	49,900	36,180	21.35	54.6
	Average	21,700	50,507	36,860	20.44	54.5
7	Coarse-grain	14,700	39,700	31,750	17.15	54.7
8	Coarse-grain	12,850	39,650	27,900	19.50	55.4
9	Coarse-grain	14,800	39,560	32,100	19.68	55.7
	Average	14,117	39,637	30,583	18.78	55.3

and in all cases past the elastic limit and followed by an annealing for three hours at a temperature of 800 deg. cent. or 1472 deg. fahr. In the fatigue tests ten samples per set were used because of the greater difficulties in a test of this kind in securing check results.

22 The physical properties of the "as received" samples showed satisfactory metal. The average tensile-strength and elastic-

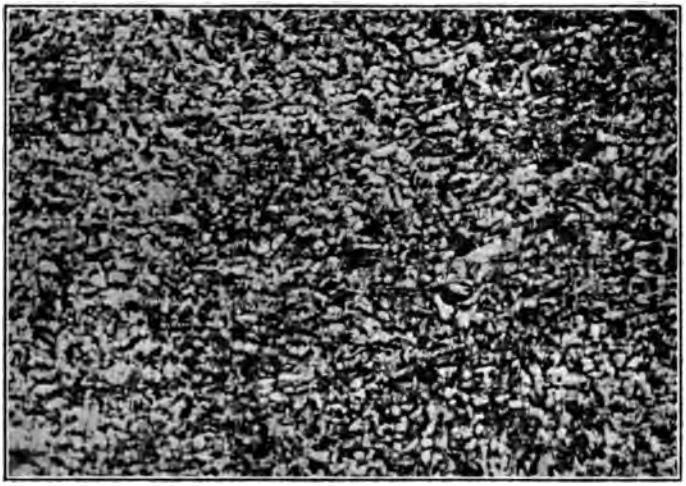

Fig. 12 Untreated; Magnification 100

Fig. 13 Heated at 650 Deg. Cent. for 15 Hr.; Magnification 100

FIGS. 12 AND 13 EFFECT OF HEAT TREATMENT ON GRAIN SIZE OF BOILER TUBES; SECTIONS ADJACENT TO TUBE SHEET; ETCHED WITH NITRIC ACID

Fig. 14 Heated at 650 Deg. Cent. for 15 Hr.; Magnification 50

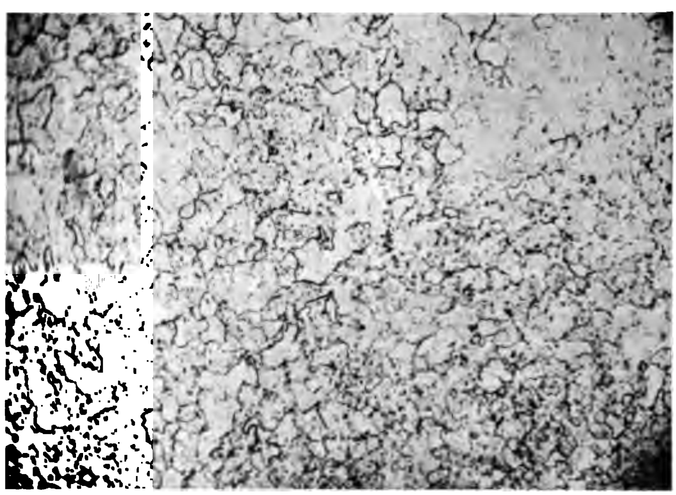

Fig. 15 Heated at 650 Deg. Cent. for 15 Hr.; Magnification 50

FIGS. 14 AND 15 EFFECT OF HEAT TREATMENT ON GRAIN SIZE OF BOILER TUBES; SECTIONS ADJACENT TO TUBE SHEET; ETCHED WITH NITRIC ACID limit values of the "coarse-grain" samples were 27.2 and 58.9 per cent lower than they were for the "as received" samples and indicate, therefore, a decidedly inferior grade of metal. In the fatigue tests,

TABLE 2 RESULTS OF FATIGUE TESTS OF BOILER TUBES

Treatment	Sample No.	Cycles for Fracture
As received	1	405
	2	420
	3	475
	4	445
	5	458
	6	430
	7	464
	8	485
	9	496
	10	479
		Average 456
Annealed	11	705
	12	830
	13	756
	14	706
	15	726
	16	792
	17	725
	18	722
	19	737
	20	751
		Average 745
Coarse-grain	21	419
	22	345
	23	375
	24	392
	25	368
	26	415
	27	405
	28	355
	29	397
	30	351
		Average 382

NOTE: — Dimensions of all specimens: width, 1 in.; thickness, 0.127 in. Throw of machine, 2 in. Speed of machine, 267 r.p.m.

also, the "coarse-grain" metal was 16.2 per cent poorer than the "as received," and 48.70 per cent poorer than the "annealed" metal.

COMPOSITION OF TUBES

23 In view of all of the conditions above pointed out there arose a question as to whether or not the present composition of boiler tubes, from the consumer's standpoint, was the most acceptable. Would there be a composition as easy to make from the producer's viewpoint, as easy to install, as resistant to the absorption of hydrogen, more strong, as free, if not freer, from blowholes, and above all less subject to recrystallization?

24 This is a formidable set of conditions, and yet do not tubes with a carbon content between 0.30 to 0.35 per cent more perfectly meet all of the above conditions than tubes with a carbon content ranging between 0.08 to 0.18 per cent?

25 Tubes with this higher carbon range will not be appreciably more difficult to manufacture or to install; there is nothing to indicate that they will absorb hydrogen more readily; it should be possible to make them as free of blowholes; there is no question but that they are at least 40 per cent stronger as measured by tensile-strength and elastic-limit tests throughout all working temperatures with no detrimental decrease in elongation or reduction; and finally, and most important of all, tubes with the higher carbon range are not subject to recrystallization.

26 On this last point the literature is suggestive, although there have been no pieces of work as yet published as far as the writer knows which give direct proof.

27 In view of this condition, therefore, the following test was carried out to ascertain roughly the carbon range in which grain growth on deformed iron [1] when heated at temperatures below the critical range, occurred. Five irons were used with carbon varying from 0.006 to 0.315 per cent. Each of the samples was annealed so as to secure freedom from strains and obtain minimum grain size. This treatment was then followed by impressing a 5-mm. ball on each specimen under a load of 3000 kg. Each sample was then heated at 675 deg. cent., or 1247 deg. fahr., a temperature considerably below the critical for all of the carbon ranges present for four hours. The specimens were then examined and the average grain size in the section undeformed and in that portion of the deformed area showing maximum grain size compared.

28 The results given in Table 3 and shown in Figs. 16 to 25,

[1] The word "iron" is used in its generic sense and is intended to include steel and what is commonly called ingot iron.

inclusive, indicate that iron ranges in carbon from 0.006 to 0.251 per cent, inclusive, undergoes a perceptible grain growth when treated as just described; also that iron with a carbon content of 0.315 per cent is not thus visibly subject to grain growth. Not only was there evident a marked growth in the ferrite grains for the irons ranging in carbon content from 0.006 to 0.251 per cent, inclusive, but in the irons in this range where there was a visible quantity of carbon existing as pearlite, very apparent agglomeration or balling up of this constituent was in evidence.

SERVICE COMPARISON BETWEEN MEDIUM-HIGH AND LOW-CARBON BOILER TUBES

29 Not only do all theoretical considerations point to the procurement of an increased life for boiler tubes through a raising of

TABLE 3 EFFECT OF CARBON CONTENT ON GRAIN GROWTH IN DEFORMED IRON WHEN HEATED BELOW THE CRITICAL TEMPERATURE

Carbon Content, Per cent	Number of Ferrite Grains per Square Inch	
	Undeformed	Deformed
0.006	3.9	1.9
0.103	25.4	13.4
0.203	63.4	23.0
0.251	26.3	16.0
0.315	(a)	(a)

(a) Grains too small to count. Photomicrographs from both the undeformed and deformed areas show no appreciable difference in grain size.

the carbon content, but the results of some actual tests on which data are now available seem to indicate and confirm this claim.

30 This test was started in 1916–1917 by placing in the front row of four of the 750-hp. Stirling boilers at the Park Place Heating Plant of The Detroit Edison Co. tubes of a medium-high-carbon content averaging around 0.30 per cent carbon, and in four other boilers at the same plant operating under the same loads at about the same time tubes with a carbon content between 0.08 and 0.18 per cent.

31 The results given in the "Service Test" (Table 4) speak for themselves, for of the medium-high-carbon tubes only about one-half have been replaced, and of the low-carbon tubes, more than a 100 per cent replacement has been necessary.

TABLE 4 SERVICE TEST ON MEDIUM-HIGH- AND LOW-CARBON BOILER TUBES, PARK PLACE HEATING PLANT, DETROIT, MICH.

Boiler Number	Installation		Replacement	
	Number	Date	Number	Date
High-Carbon Tubes in Front Row				
3	29	June 19, 1916	27	July 27, 1918
			15	Summer, 1920
6	29	May 9, 1916	13	Sept. 11, 1917
			5	July 15, 1919
			1	Summer, 1920
7	29	June 19, 1917		
8	29	June 9, 1917		
	Totals 116		61	
Low-Carbon Tubes in Front Row				
1	29	Sept. 23, 1916	27	July 12, 1918
			5	Summer, 1920
2	29	Sept. 18, 1916	10	Sept. 28, 1917
			27	July 11, 1920
			7	Summer, 1920
4	29	June 14, 1916	18	Jan. 6, 1917
			5	July 27, 1918
			13	June 17, 1919
			5	Summer, 1920
5	10	June 5, 1916		
	19	Aug. 3, 1917	10	August 3, 1917
	Totals 116		127	

CONCLUSION

32 This paper has been prepared, not for the purpose of suggesting radical changes in boiler-tube composition nor for the purpose of criticizing present boiler-tube manufacturing practice, for, all things considered, it is on a very high plane with respect to quality.

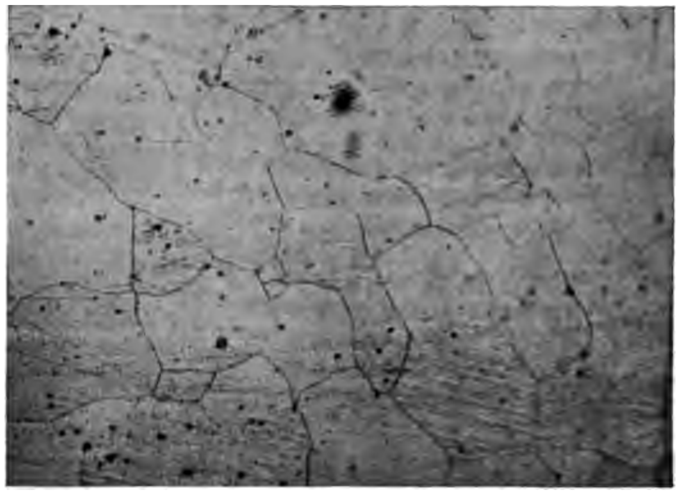

Fig. 16 Carbon 0.006 Per Cent, Undeformed

Fig. 17 Carbon 0.006 Per Cent, Deformed

FIGS. 16 AND 17 EFFECT OF CARBON CONTENT ON GRAIN GROWTH IN DEFORMED IRON WHEN HEATED BELOW THE CRITICAL TEMPERATURE; MAGNIFICATION 100; ETCHED WITH NITRIC ACID

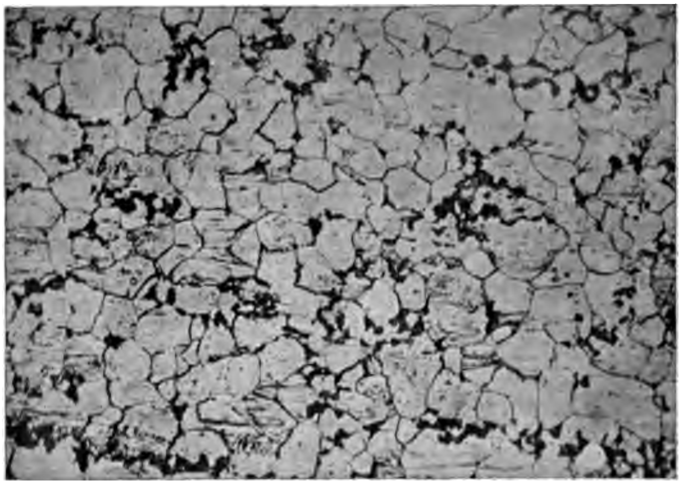

Fig. 18 Carbon 0.103 Per Cent, Undeformed

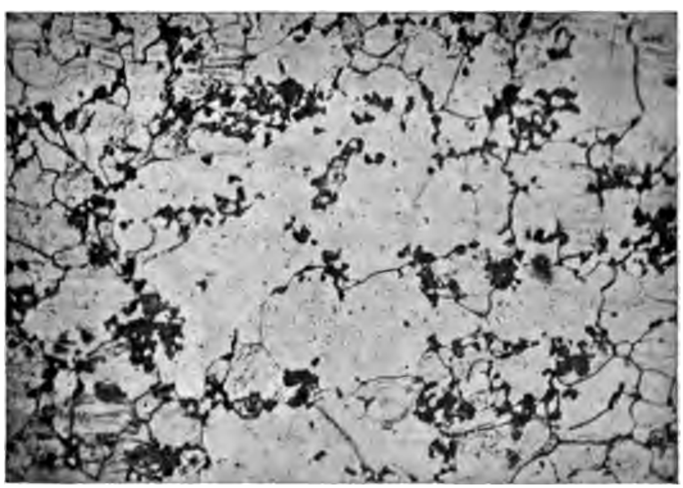

Fig. 19 Carbon 0.103 Per Cent, Deformed

FIGS. 18 AND 19 EFFECT OF CARBON CONTENT ON GRAIN GROWTH IN DEFORMED IRON WHEN HEATED BELOW THE CRITICAL TEMPERATURE; MAGNIFICATION 100; ETCHED WITH NITRIC ACID.

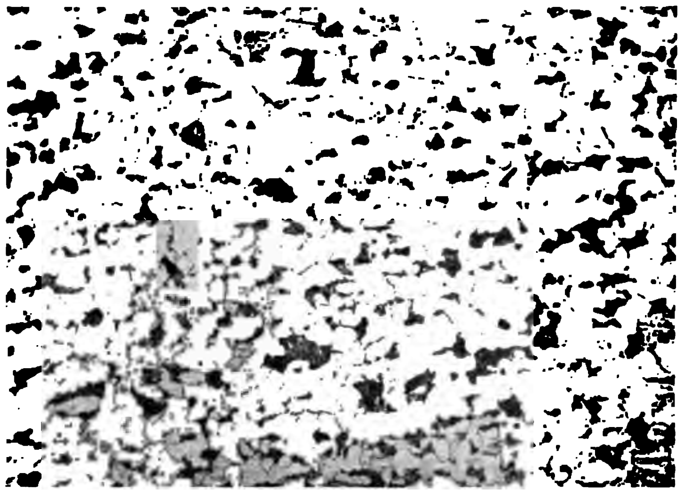

Fig. 20 Carbon 0.203 Per Cent, Undeformed

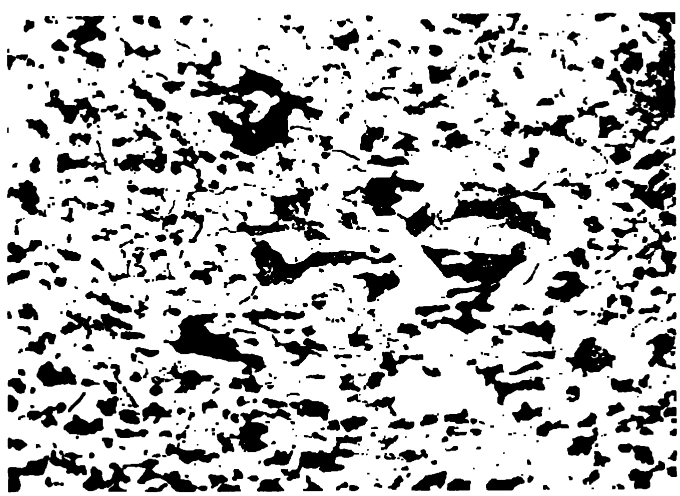

Fig. 21 Carbon 0.203 Per Cent, Deformed

Figs. 20 and 21 Effect of Carbon Content on Grain Growth in Deformed Iron when Heated below the Critical Temperature; Magnification 100; Etched with Nitric Acid

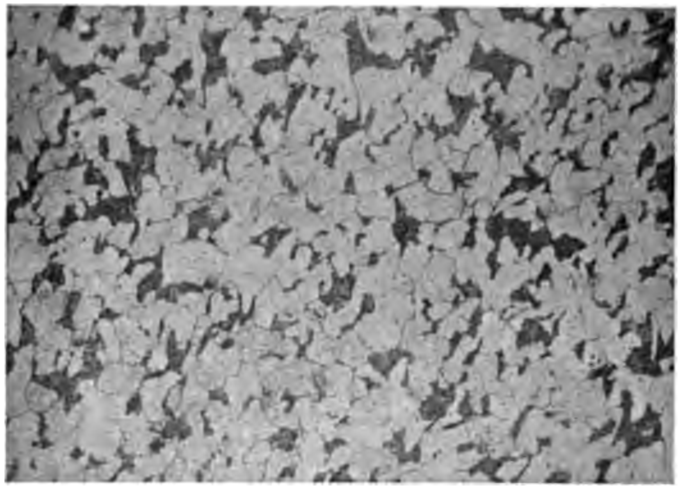

Fig. 22 Carbon 0.251 Per Cent, Undeformed

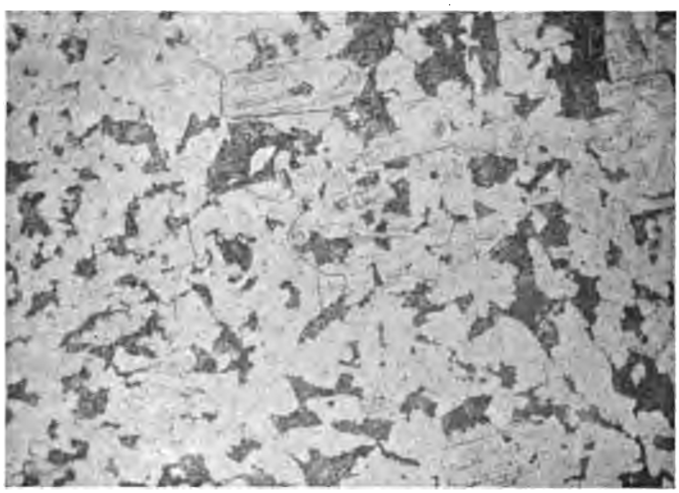

Fig. 23 Carbon 0.251 Per Cent, Deformed

FIG. 22 AND 23 EFFECT OF CARBON CONTENT ON GRAIN GROWTH IN DEFORMED IRON WHEN HEATED BELOW THE CRITICAL TEMPERATURE; MAGNIFICATION 100; ETCHED WITH NITRIC ACID

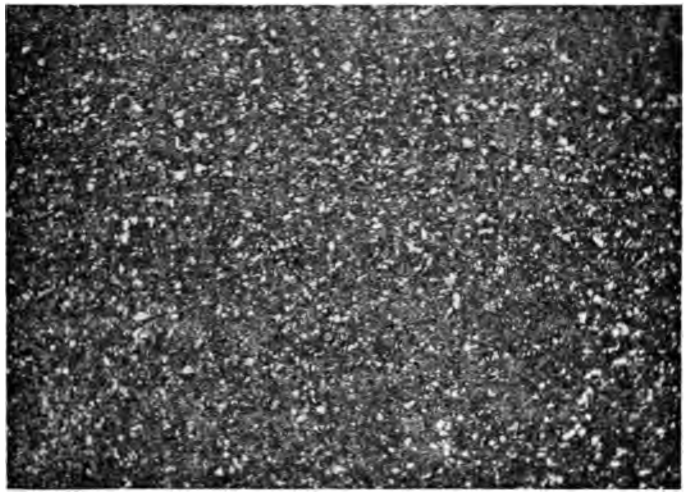

Fig. 24 Carbon 0.315 Per Cent, Undeformed

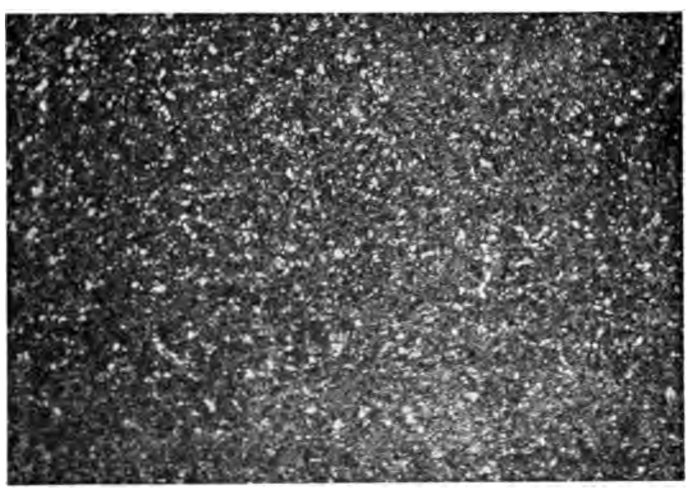

Fig. 25 Carbon 0.315 Per Cent, Deformed

Figs. 24 and 25 Effect of Carbon Content on Grain Growth in Deformed Iron when Heated below the Critical Temperature; Magnification 100; Etched with Nitric Acid

It has been prepared, however, to present certain facts, especially those relating to grain growth, to which tubes of the commonly accepted composition are so subject, and in view of these facts to question whether tubes with a carbon content varying between 0.30 and 0.35 per cent would not insure longer tube life and safer boiler operation than tubes with a carbon range between 0.08 to 0.18 per cent.

DISCUSSION

F. N. SPELLER.[1] The results of service tests given in Table 4 seem to indicate a distinct advantage in using medium carbon steel tubes, this being due in the writer's opinion more to the greater strength of the higher carbon steel at high temperatures. We have found by tests of the physical properties of steels of various carbon contents at high temperatures that the strength of these steels show about the same relative difference from normal temperature up to about 900 deg. cent. The higher carbon tubes are, therefore, less liable to bag in service, which is undoubtedly the most common cause of failure.

The results given would probably have been more strictly comparative had the two kinds of tubes been installed in the same row of each of the boilers side by side.

Referring to the theoretical discussion of the causes of tube failures in the first part of this paper, the author does not state that any of the samples examined came from tubes which failed in his test, nor in fact does he classify the tubes which failed in his test according to the causes stated.

The white area in Fig. 4 seems more likely to be due to decarbonization as the average phosphorus contents in B.O.H. tube steel is rarely over .020 per cent. Referring to microphotos (Figs. 8 and 9), in low carbon tube steel defects due to blow holes in the ingot are rarely found in the finished tubes, as most of these are welded up, especially in the case of lap welded tubes. Recrystallization does occur *with a certain degree* of cold strain and *within a narrow range of temperature;* however, the probability of such a combination occurring in service would be very remote. It is true that all cold drawn seamless tubes are, as the author says, "actually brought to final size when below the critical temperature" and he should have added "are

[1] National Tube Co., Pittsburgh, Pa.

given a final anneal." In spite of this and other distortion incidental to bending and setting tubes in the flue sheet, cases of failure due to grain growth are very rare in service, although, as Professor Sauveur, the author and others have pointed out, this condition can be developed under favorable conditions in the laboratory.

The results with this experience with medium carbon steel tubes is most suggestive and may develop into a useful change in tube specifications, but if so, the advantage would seem to be due to increase in strength at high temperatures rather than to any important improvement in quality of the steel.

There would probably be some difficulty in safe ending medium carbon tubes, but aside from this there seems to be no practical reason why such steel should not be used for seamless tubes after the advantages indicated by the author are proven out by further service tests.

Isaac Harter. The author, in various places in his paper, points out the presence of frequent ghost lines in ordinary steel boiler tubes, but the paper does not show whether such segregation areas have any tendency to become the seat of tube blistering in service, or whether, if a blister starts elsewhere, and reaches one of these segregated areas, there is any apparent difference at that point in the characteristics of the blister. It seems a fair point to make that if nonhomogeneity were a serious factor, charcoal-iron tubes would be unsatisfactory to a very marked degree as compared with steel. The paper does not bring out whether the higher carbon-steel tubes recommended were any more free from segregation than the ordinary tubes which were examined.

The author does not define specifically the form or forms of tube failure which he is considering. The possible forms of failure, however, would appear to be:

(*a*) Blisters or bagging
(*b*) Splitting made possible by brittleness
(*c*) Eating through by corrosion.

A tube blister begins with the localized heating of a part of the tube to a temperature sufficiently high either to allow very rapid external oxidation or bulge from the contained pressure or (and sooner or later) both.

Failure caused by brittleness when it occurs along the tube can be distinguished from blistering by the unreduced section at the split, and in this form is comparatively rare; though extreme brittleness in tube ends and splitting at that point is frequently observed.

The author mentions the following causes for tube failures:

(*a*) From brittleness from hydrogen absorption

(*b*) Blow-holes or other imperfections

(*c*) Recrystallization, to which may be added nitrogen.

It hardly seems likely that brittleness from hydrogen can play a part in splitting of tubes along the body of the tube, as cases where corrosive feedwater limits the life of the tube do not show failures by splitting, as would be expected if hydrogen were a cause. The cases where cracking has been traceable to the action of hydrogen have been those occurring in the joints of boilers under such circumstances that extreme concentration of the liquid in the joint is a necessary condition, and by extreme concentration is meant a concentration far beyond what it would be possible to operate with in a boiler as a whole, so that, again, on this account, hydrogen does not seem to be a likely factor. The splitting of tube ends projecting into drums or headers has never been investigated so far as the writer knows to a point where any positive explanation can be offered, but it would seem that this type of splitting cannot be accounted for by recrystallization, since the necessary temperatures are lacking; nor does hydrogen seem to be a likely explanation, in view of the reasons given just above.

The writer is of the impression that splitting of tube ends, and most cases of splitting in the body of the tube that are not accounted for by actual seams in the tube, are to be found in Bessemer steel tubes, and the brittleness has its origin in changes taking place in the nitrogen content of such material. This statement is made because splitting of tube ends was much more frequent ten or fifteen years ago, when Bessemer tubes were more common, than it is today, and because in one case which was examined for nitrogen, the nitrogen content was exceptionally large, even for Bessemer steel.

The author gives a formula connecting time and temperature for recrystallization in metal of the kind discussed, deformed below the critical point. From this formula it would appear that for growth to take place in five years, a temperature of 375 deg. cent., approximately, would be required; that if the temperature were maintained at 350 deg. cent., this period would increase to twenty-five years. If this is correct, then it would appear certain that no recrystallization can take place except in the furnace row of tubes, and that it would be of negligible amount in even the furnace row except within the region of the blister. In a blister it would undoubtedly take place, and so rapidly as to be complete, even before external oxidation could have developed sufficiently to cause a failure of the tube from that cause alone. It would appear, therefore, that though the higher carbon tubes which the author describes would not suffer this recrystallization in the blister, there is a question whether external oxidation is not so preponderating a factor that whether the blistered area was recrystallized or not, would be of comparatively little importance. It may, however, be a fact that the recrystallized material oxidizes more rapidly than it would if not recrystallized, and in that sense that the higher carbon may give a very real contribution to the life of the tube. It may also be that the tendency for segregation and other defects having their origin in the ingot would be less with the higher carbons.

The point of most importance brought out by the author is that service-test results showed a longer life for the high-carbon tubes. In this he has made a suggestion for an improvement in practice which is of great importance, and the writer believes that further trials should be undertaken upon a larger scale and under more definite comparison conditions. It is suggested that a boiler be tubed in the furnace row with alternate tubes of high and low carbon, and that the boiler be run under water conditions so controlled as to cause tube failures by scale, by scale and oil, and by corrosion, for, as the author remarks, tubes of higher carbon range are entirely practical from the point of view of manufacture and installation.

G. P. McNiff.[1] The author goes extensively into the metallographic research on boiler tube steel, but he does not attribute definitely, nor does the writer believe he can state from the facts presented, that the failures are due to imperfections in the metal.

It is our opinion that the microphotographs presented are representative of good commercial low carbon boiler tube material.

The writer's opinion along the metallographic investigation is thoroughly expressed in quoting from *Chemical and Metallurgical Engineering* of November 24, 1920, Vol. XXIII, No. 21, p. 1002, in an editorial, Good Rails are More Than Sulphur Prints. In quoting therefrom the writer is taking the liberty of substituting "tube" for the word "rail" in the last word of the second paragraph:

[1] Metallurgical Engineer, National Tube Co., Pittsburgh, Pa.

A metallographist, if asked to state the fundamental axiom of his science, might respond that metallographical uniformity is synonymous with metallurgical excellence. But he would be placing the cart before the horse, for it is not always true even that metallurgical excellence demands microscopic homogeneity; furthermore excellence certainly does not stop with uniformity, nor is metallurgy limited by the microscope. Following analogous sophisms, are not metallographists, chemists and testing engineers in danger of building up a picture of what good metal ought to look like, analyze like, and test like; and then when they find a piece of metal which approximates this masterpiece, say, "Behold, here is a really excellent tube"?

The author has brought out a feature in respect to the time temperature criterion of crystal growth. The crystalline growth of steel, especially .20 carbon and under, is brought out very admirably in Sauveur's The Metallography and Heat Treatment of Iron and Steel, pp. 265 to 273 inclusive. The time equations between the temperatures 550 deg. cent. and 675 deg. cent. brings out features which are novel and interesting. This crystalline growth in cold drawn seamless steel tubes .20 carbon and under, where the reductions in cold drawing are not over 22½ per cent, and final annealing operation not properly done, may be of some moment in practical usage. However, he does not show conclusively that the tube failures, as found in practice, of the low carbon tubes are due to the crystal growth as obtained in the laboratory.

Our conclusions from the data given are that the better results obtained in the use of hot rolled seamless .30 to .40 carbon grade over the standard .08 to .18 carbon boiler tube grade, are directly caused by the higher elastic limit and higher tensile strength of the .30 to .40 carbon grade over the lower carbon content tube, since these physical properties at high temperatures are proportional to their same values at normal temperatures.

D. S. Jacobus. The author should be commended for conducting a most important investigation. It is only through pioneer work of this sort that we are safeguarded from following too closely along set lines and are able to advance. Until someone was enterprising and bold enough to use steel plates for the shells of boilers it was thought that wrought iron was the only proper material, and it may be that we will have to depart from some of our present ideas as to the best material for certain uses. Steam pressures have increased to an extent that involves the employment of plates of such a thickness in some cases that it is questionable whether a higher tensile strength than that now used could not be employed to advantage. The tests indicate that there is a promising field for further experiments. Let us hope that more data of the quality given in the paper will be forthcoming.

E. B. Powell expressed his appreciation of the value of the paper and emphasized the deterioration of the tubes due to hydrogen mentioned by the author as a fruitful field for further investigation.

William H. Boehm. The author has presented some thoughts deserving of very serious consideration, particularly by the tube manufacturers, and there is no doubt that they will be willing to investigate the matter further. Probably not everybody appreciates the very considerable number of ruptures which occur in practice. Boiler insurance companies know it because they have so many small losses for which they must pay.

The Author. The discussions given by Messrs. Speller, Harter McNiff, Boehm, Powell, and Jacobus on this paper have been most appreciated for they all tend to develop the subject more fully and throw further light upon its numerous ramifications.

With reference to the remarks made by Mr. Speller, it is pleasurable to note that he agrees that the service tests given in Table 4 seem to indicate a distinct advantage to medium carbon steel.

Mr. Speller seems to believe that the white area in Fig. 4 is possibly due to decarburization rather than to phosphorus content, since the phosphorus content in basic open-hearth tubes is rarely over .02 per cent. The author believes, however, that this white banding cannot be attributed to decarburization. It is found within the metal and not on the surface. and there is no evidence that there is scale close to the band which. if present, might, it is granted, be a possible cause of inside decarburization. It is agreed that the average phosphorus content in basic open-hearth tubes is low, yet due to segregation that occurs near the pipe and which is present if the ingots are too closely cropped and segregation that occurs around blow holes, though these may later become welded in the rolling operations, there is strong reason to believe that there are portions of the metal sufficiently high in phosphorus to enable the production of ghost lines. Strong evidence in support of this statement is the fact that after an examination of several hundred specimens, ghost lines were found in at least fifty per cent of them.

Throughout the paper reference was made to the black centers, usually round and also usually surrounded by a white annular ring which is, in turn, enclosed by a black annular ring, as blow holes or other imperfections. The author does not know how to describe them otherwise. He is rather inclined to believe they are blow holes, though it is true that practically all of the blow holes in the metal are welded and when such is the case there would be no evidence or trace of them.

Radiating from many of these imperfections are irregular lines which have every appearance of being cracks. It is believed that when these imperfections form the nipple of a bag, failure will start fiom the center of the nipple, progress along some one or more of the cracks and continue until the bag opens up.

It is unquestionably true that the criteria for recrystallization are degree of strain, degree of temperature and time. Not every degree of strain will result in exaggerated grain growth, nor, so far as we know, will every degree of temperature result in the grain growth of critically strained metal. It is possible, however, to get in boiler tubes such strain, such temperature and such time that growth will occur. Nor is the critical temperature necessarily "a narrow range of temperature" for the author has obtained crystal growth in critically strained boiler tubes over a range of 300 deg. cent. or 540 deg. fahr.

It is unfortunate that it is so relatively difficult to prove or disprove the extent to which recrystallization has resulted in tube failure. The fact that failure needs only to start from a small surface, possibly not as large as a pin point, to produce disastrous results; and the fact that it is practically impossible after tube failure to find that particular pin point at which failure started, adds materially to the difficulties of points in contention. However, we do know that the metal is subject to mechanical deformation to such an extent in its handling, especially after it leaves the tube plant, that under proper temperature conditions growth will occur with all of the attendant undesirable properties. This condition is pointed out in the paper in Figs. 12, 13, 14, and 15. It is also not difficult to imagine that some small portion of the surface of some tubes is heated up to a recrystallization temperature when a piece of scale lodges on the water side of the tubes. If the scale lodges against metal that has been critically deformed, the possible result is grain growth with possible tube failure. Because of these conditions, therefore, the author believes that some cases of tube failure, hitherto unexplained, may be laid to recrystallization.

The remarks made by Mr. Harter have been read with much care. From several hundred sections of metal which have been examined it has been noted that there is a strong tendency for the blow holes or imperfections which have the same general characteristics as blow holes to locate in the ghost lines. This is not a universal tendency, though the larger number of imperfections of this type are found in the ghost lines. The author believes that were these imperfections absent there would be a decidedly smaller number of tube failures. Appreciating industrial conditions, he realizes the complete elimination of these imperfections is quite impossible; yet he believes that by cutting down their number to the lowest possible extent, there will be an increased life to tubes, especially those in the front row of boilers.

The author also feels that service tests would show that charcoal iron tubes would not have the same life under present operating boiler conditions as high-grade steel tubes. Though charcoal iron tubes are much freer from the common metalloids than steel, yet the effect of the slag would be deleterious to long life.

The writer is appreciative of the suggestion made by Mı. Harter that tube failures may result from one of three causes:

(*a*) Blisters or bagging

(*b*) Splitting made possible by brittleness

(*c*) Eating through by corrosion.

No attempt was made to discuss tube failure caused by corrosion, nor tube failure caused by splitting made possible by brittleness resulting from absorption by the metal of hydrogen or nitrogen or from the contained presence of excessive quantities of metalloids.

It is noted that Mr. Harter dwells at some length on brittleness in tubes made of Bessemer steel, and adds, thereby, materially to the contents of the paper. Along this same line some work which the author has recently had called to his attention would confirm what Mr. Harter states when he charges certain types of tube brittleness in Bessemer steel to nitrogen.

There is one slight correction to the remarks made by Mr. Harter in his eighth paragraph which the writer would like to make. It relates to his statement of the time required for cyrstal growth. Assuming the formula given in Par. 19 is applicable for temperatures below 550 deg. cent., substantially eight years would be required to pıoduce recrystallization at 375 deg. cent. in place of the five years given by Mr. Harter and twenty-four years in place of the twenty-five years. This time phase of the matter, however, is entirely hypothetical and problematical. It is believed that in most cases of tube failure resulting from recrystallization the growth developed in a very short interval of time, measured probably in minutes or seconds rather than in years.

The author feels that oxidation will occur more rapidly in a coarsely crystalline product than in a finely crystalline product. He has no proof on this matter, though it is well known that the evidences of a burnt structure are always intra-crystalline rather than intercrystalline and that the temperatures just preceding those of burning lead to the development of coarse crystals.

The author also believes that it is easier to make .30 to .40 per cent carbon steel freer from blow holes and segregations than steel with a carbon content between .08 to .18 per cent. He thus believes that steel of the first type is naturally freer from blow holes and imperfections.

He trusts that no impression has got abroad that the steel in boiler tubes is of an inferior quality. The conditions of manufacture are such that the steel must be of a fairly acceptable grade in order to withstand the severe mechanical operations to which it is subjected. He is not therefore complaining of the quality of the steel although he is pointing out that he believes that with a higher carbon content it will be possible to make steel even freer from imperfections than is the case with the present low carbon content.

The remarks made by Mr. McNiff have been read with much interest. His quotation submitted is a clever piece of English phraseology. The quotation was inserted to leave a feeling that metallographic uniformity is not synonymous with metallurgical excellence. Where chemical composition and heat treatment, if required, is of the proper type the author personally knows of no case where metallographic uniformity is not synonymous with metallurgical excellence.

The viewpoint with which this paper was written was not in great criticism of present boiler tube manufacturing practice. As found in Par. 32 the statement is made "it is on a very high plane with respect to quality." From the several hundred tubes which have been examined, however, much was found which needed improvement. The author is not interested in knowing where these tubes which he examined came from, but he trusts and hopes that some of the things which this investigation has disclosed will start investigation and work at plants making tubes to ascertain if the present conditions which this metallographic examination has disclosed represent the highest attainable metallurgical excellence.

No. 1768

DISTILLATION OF FUELS AS APPLIED TO COAL AND LIGNITE

By O. P. Hood, Washington, D. C.

Member of the Society

There is a widespread feeling that the present methods of using coal are not wise, but exaggerated ideas are held as to the possibilities and value of coal distillation. The necessary investment is so large that the popular hope for cheap solid liquid or gaseous fuels produced by distillation processes is not likely to be realized. It is more likely that fuels will be produced whose special qualities will command a good price.

Low temperature distillation of coal has hardly reached the commercial stage, the most advanced experiment being the carbo coal process. Coal distillation will play an increasing part in supplying industrial and city gas and smokeless fuel for domestic heating.

In the lignite areas of the United States where there are large supplies of low grade fuel, distillation methods can be applied to lignite, obtaining gas, tar, ammonium sulphate, and a carbonized residue of excellent heating value.

MUCH has been written on the subject of the distillation of coal, covering the field from the intricate chemical reactions involved to the economic aspects having a bearing on statecraft. The subject is by no means new, since men of perception have for many years pointed out the fact that coal is a raw material from which many values other than the heat of direct combustion can be obtained.

2 The growing appreciation of these values by the layman as a result of present writing is, however, new. There is a widespread feeling that we are not wise in our present methods of using coal. It is well to awaken general interest in this subject, and presenting the matter in exaggerated form to more quickly convey an impression is somewhat justifiable, but it is quite possible to overdo the matter. The layman is beginning to repeat some of the extravagances presented to him by the reporter and space writer who are exploiting the sensational possibilities of universal processing

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of coal. When coal is used raw for heat generation it should always be with a proper understanding that this material might, under more favorable circumstances, be used for higher purposes. Besides major products of coke, gas, and pitch, an alluring chain of oils, dyes, and medicinals might have been produced.

3 While all this is possible it does not warrant the conclusion that all coal should be made to yield the last-mentioned products. Because wheat can be made into cake the market for bread is not to disappear. There will always be much use for raw coal, for in this form the individual has the disposition of a maximum amount of energy with the least fixed liability. With raw coal in his own hands he is independent and can carry out his own will, while with the energy otherwise delivered he is dependent on a complex coöperative enterprise. With the energy of coal transferred to gas or to electricity where, as Mr. W. S. Murray says, we have maximum power and minimum bulk, the consumer must in some way assume his share of the upkeep of a vast distribution system not subject to his own disposition, and not readily reaching every locality. It must be recognized, therefore, that the distillation of coal is not for universal application, but that it will find its appropriate field of usefulness together with other and cruder methods.

4 The layman is likely to overlook the fact that processing of coal to obtain more than one or two products is necessarily a large enterprise and not to be entered into with small means or on a limited scale. Even on a large scale the tendency has been to look to a single product to justify the installation; e.g., in the production of metallurgical coke the gas has been wasted, and in the production of city gas the disposal of coke was a problem. While these conditions have changed, there remains a disinclination to enter into a business which depends upon marketing a chain of widely dissimilar products. Entrance into this business is therefore not easy, nor does it simply depend upon the possession of a source of raw material having favorable possibilities of higher distillates.

5 The investment required in large plants for coal processing is so great that it is no reproach to forego this form of ultimate economy so long as our savings are more needed for development in other directions. In other words, conservation of capital is as necessary as conservation of raw material. One may regret that the clearing of western timbered fields did not coincide with the present demand for black walnut lumber, but at that time the

cabinet wood was more worthily used as fence rails than as desk tops. The same principle may be applied to the present use of raw fuel.

6 There are, however, rapidly developing fields in which the processing of coal is extremely desirable and is practicable. These fields are made apparent by the greatly increased cost of coal, by a general realization that there is a limit to easily won high-grade coal, by the decline of natural-gas supply, the increased cost of transportation, and a growing appreciation of and dependence upon fluid fuels, both gas and oil. A study of each of these factors brings one at some time in contact with processes involving coal distillation.

7 There is a popular hope that if elements of higher value are recovered from coal by distillation, the remaining fixed carbon can be sold at a price lower than the raw fuel; in other words, that coke as a domestic fuel can be sold at a cheap price. Where in any process a chain of products is produced, the actual cost of production of each is largely a matter of bookkeeping and each is priced at what the market will bear. So far, the market has absorbed coke at a price which leaves little hope for cheap fuel from this source and it is not likely that processed coal can yield cheap fuel. It is more likely to produce a fuel whose special qualities of cleanliness and smokelessness will command a good price and supplement our declining store of anthracite coal. There is also the hope that by extensive distillation and other processes manufactured gas may reinforce our rapidly disappearing natural gas. There is little to justify the hope that relative prices for heat units in the form of gas can ever again be as low as they have been in the case of natural gas, for it is now recognized that even natural gas has been sold too cheaply to return the investment at the end of the producing period. The cost of transportation and distribution of gas is, in general, about equal to the cost of production, so that reduced manufacturing costs per thousand are not directly applicable to selling price. The reduction in heating value for town gas is greatly to be desired in order to make available cheaper methods of gas manufacture, but the more bulky heat carrier will increase the distribution charges, so that altogether very cheap heat in the form of gas is not probable. In this field of industrial and city gas, coal distillation is to play an increasing part as gas-oil supplies decline, but it cannot be on a price level comparable with former prices of natural gas.

PROCESSES OF DISTILLATION

8 The fact that motor and other oils can be obtained from coal distillates and that there is a market for all that can be produced, strengthens the position of distillation processes. There is a growing appreciation of the fact that the character of coal distillates is quite as much a factor of the process as of the coal used. The amount and quality of the distillates is determined by the heat experience of the primary complex constituents of the coal, so that variation in the engineering structure and mechanism of the apparatus, together with temperature treatment, gives opportunity for a variety of processes and results. These processes can, therefore, within limits, be fitted to the major products desired. A great deal of attention has been given to distillation resulting from temperatures not exceeding 1400 deg. fahr., which are lower than used in coke-oven or gas-retort distillation. Prof. S. W. Parr states that "in general, it is believed that all the products of decomposition have a higher intrinsic value as delivered under low-temperature conditions chiefly because excessive secondary decompositions are avoided." With the usual methods and lower temperatures there will be considerable difficulty in transmitting heat to coal at a sufficient rate to obtain reasonable capacities from a given investment, and in this particular low-temperature distillation is at a great disadvantage, which must be equalized by a greater refinement of its products. It is therefore hard to see how low-temperature distillations can be expected to yield a cheap fuel.

9 The most advanced experiment in this country using low-temperature carbonization and distillation is the carbocoal process, which produces a very fine fuel and a large yield of tar of special quality. The fuel, however, is a worthy competitor of anthracite rather than of soft coal and certainly will not be a cheap fuel. In this process nearly double the quantity of liquid products is produced, showing the adaptability of distillation processes.

10 Distillation of coal in by-product coke ovens is already the major process in the production of metallurgical coke, and in a few years probably only the peak load of productive years will be carried by beehive ovens. The gas produced is used in the adjacent industries which determine the location of the plant, and much of the tar is burned instead of fuel oil. An appreciable addition to the supply of motor fuels is made by the recovered benzol, and ammonium sulphate for fertilizer is recovered in quantity.

11 A return to distillation methods in the city gas industry is indicated now that gas oil is no longer cheap for the enrichment of water gas and an outlet for coke in domestic heating service becomes more assured. This is the most favorable field for the expansion of distillation processes. One of the most sensible fuel-conservation methods that can be fostered by the householder is to get the habit of buring coke. This should be obtainable in prepared sizes the same as anthracite, and in this form is finding favor in many markets.

12 There is great need for a high-class domestic fuel in the lignite areas of the United States where there are large supplies of low-grade fuel. The increased difficulties and cost of transportation of eastern coals into these districts makes the problem a specially urgent one. Experimental work has shown the possibility of applying distillation methods to lignite, obtaining gas, tar, ammonium sulphate and a carbonized residue of excellent heating value. In most regions where this need exists the solid fuel would be the main objective of the process, possibly wasting the others. The char is too fine for domestic use without briquetting, but when put in proper form it makes a fuel comparable with anthracite.

13 In the southern lignite area there is also a need of developing a distillation process which will make lignite available for the production of city gas. This seems quite possible from the indications of laboratory work, but in both of these fields there is no commercial development. It seems as though the laboratory and small-sized experimental plant had contributed about all that it could and the business awaits that longer step which must be taken to realize these results in full-sized apparatus. In fact, many fuel-distilling processes seem to have reached this interesting stage of expectancy.

No. 1769

STEAM FORMULAS

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The purpose of the following paper is twofold: first, to make a comparison of the experimental data and the various formulations involved in determining the various steam formulas given; and secondly, to present a new set of equations derived by the writer in the course of the investigation.

The paper is divided into four parts. Part I defines quantities and symbols, states exact values of fundamental constants, shows the field of the two characteristic equations, and develops the general form of those equations. Starting from the simplest statement the development runs to complete algebraic form and definition of terms. Part I also contains a general view of the body of physical data in its several departments, a classification of formulations as to kind, and a chronological list of all the formulations that are to enter into subsequent comparisons.

Part II is concerned with the properties of saturated steam, that is, with the boundaries of the region of water-and-steam mixture. These consist of the liquid line on one side and the dry-steam or saturation line on the other, merging at the critical temperature. Over the range up to about 1000 *lb. pressure the saturation curves are obtained by evaluating the general steam-gas or superheat equations. For the rest these boundaries are defined by empirical formulas which are set forth.*

Part III deals with superheated steam and is the most important section of the paper, giving the two general equations and several sets of constants. These are found in Tables 1 *and* 2. *The curve and data comparisons are of great interest in showing the high "state of the art" of steam formulation.*

Part IV gives some account of the methods used in fixing the constants of the general equations, which are listed in the first part of the paper. It also contains matters for debate among specialists.

I GENERAL VIEW OF SUBJECT

INTRODUCTORY

THIS paper presents a revised and thoroughly consistent formulation of the properties of steam. Begun in 1916, the investigation had for primary purpose no more than a critical comparison of the then existing formulations, listed in Par. 17. It soon appeared that there was room for another solution, which should more closely interpret all the data and have a little stronger probability of correct

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extension into the region above experiment. When nearly worked to a conclusion the matter was dropped on account of war conditions, to be resumed and completed in 1920.

2 The important quantities to be related, with their units of measurement in the foot-pound-fahrenheit and the meter-kilogram-centigrade systems, are as follows:

(*a*) Temperature t or absolute temperature T, degrees fahrenheit or centigrade.

$T = t + 459.6$ (fahr.); $T = t + 273.1$ (cent.) . . . . . .[1]

t_s = saturation temperature or boiling point under an existing pressure p

t_c = critical temperature of water = 706.3 deg. fahr. or 374.6 deg. cent.

(*b*) Absolute pressure p, pounds per square inch or kilograms pe- square centimeter.

These are practical or engineering units: in thermor dynamic equations, where pv should be a work quantity in foot-pounds or meter-kilograms, p would be in pounds per square foot (factor 144) or kilograms per square meter (factor 10,000).

(*c*) Specific volume v, cubic feet to the pound or cubic meters to the kilogram. Special symbols, for particular values or cases of volume v, are:

w = specific volume of water, at p and t_s

v_s = specific volume of dry saturated steam

$u = (v_s - w)$ = increase of volume during vaporization.

(*d*) Total heat or heat content h, British thermal units per pound or calories per kilogram: these are "mean" heat units. For particular values or cases of h,

q = heat of liquid, required to raise the unit weight of water from freezing point to t_s, under constant pressure p

r = heat of vaporization or latent heat, at p and t_s

$h_s = (q + r)$ = heat content of dry saturated steam.

Note that h is taken to be the sum of internal energy plus the total outer work Apv of a constant-pressure expansion from zero volume to v; it therefore includes the work of the feed pump or Apw_f.

3 The secondary cardinal properties, density, internal energy, and entropy, self-determinate but derivable from those just named

by mere routine computation, will not be considered in this discussion.

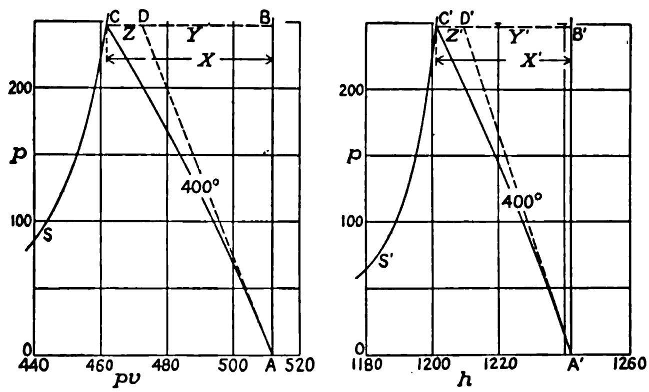
4 The exact values here used for several fundamental constants are:

(*a*) For the mechanical equivalent of heat,

$$J = 778 \text{ ft-lb. per B.t.u., or } 426.84 \text{ m-kg. per cal.} \quad . \; . \quad [2]$$

(*b*) With the practical pressure units as prescribed above, the effective value of the heat equivalent of work, in general the reciprocal of J, becomes

$$A = \frac{144}{778} = 0.18509, \text{ or } A = \frac{10,000}{426.84} = 23.428 \quad . \; . \; . \quad [3]$$



FIGS. 1 AND 2 ISOTHERMAL CURVES OF pv AND OF h ON p, TO ILLUSTRATE THE GENERAL EQUATIONS

for use in converting pv quantities into heat units in thermodynamic formulas.

(*c*) With those same units the ideal gas constant for water in the equation $pv = BT$ is here taken exactly as

$$B = 0.5952, \quad \text{or} \quad B = 0.0047023 \; . \; . \; . \; . \; . \; . \quad [4]$$

A systematic collection of symbols and ratios of conversion will be found in the Appendix, while slightly variant values of J and B appear in the tabulation of formula constants, Par. 36.

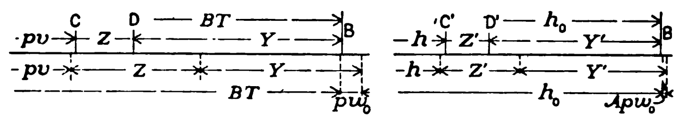
5 In order to visualize the problem of steam formulation, turn first to Figs. 8 and 9 and glance at the boundaries of the saturated state, as laid down in the main curves AAK and BBK, merging at critical point K. Note that the base (vertical) is t_s, and that in the volume diagram we do not plot w and v_s directly, but rather the

products pw and pv_s; this is far more convenient, because the latter quantities have only a small range of variation. In Fig. 10 the two lines of dry saturation are replotted on a pressure base, showing the form which they really take when serving as limits for the two general equations of superheated steam.

6 These two characteristic equations will relate outer work pv and heat content h, respectively, to temperature T and pressure p as independent variables. For an ideal or perfect gas the two quantities would be very simple functions of T alone, according to the equations

$$pv = BT \qquad h = cT \quad . \; . \; . \; . \; . \; . \; . \quad [5]$$

where B is the gas constant and c is a constant specific heat. These are represented by the vertical lines AB, A′B′ in Figs. 1 and 2, which are ideal isothermals. The behavior of any actual gaseous substance



FIGS. 3 AND 4 DETAILS OF AN ORDINATE FROM FIGS. 1 AND 2, CARRYING ALSO DIMENSIONS FROM THE GOODENOUGH EQUATIONS

is characterized by a contraction of both pv and h along the isothermal with rise of pressure, as represented by the lines AC, A′C′ and expressed by writing the general equations in the form

$$pv = BT - X \qquad h = h_o - X' \quad . \; . \; . \; . \quad [6]$$

Quantity h_o, the heat content at zero pressure, is made definite in Par. 39.

7 Contractive terms X and X' must satisfy the physical concept and fact that departures from the behavior and dimensions of the ideal gas are greater as pressure and density are higher and as the gas approaches its limiting state of saturation. They will, therefore, be functions of both pressure and temperature, increasing with rise of p and with fall of T. Experiments upon both volume and heat content have shown that the isotherms in diagrams of either pv or h on p are curved; and this fact is met mathematically in reasonable and flexible fashion by splitting X and X' each into two terms, Y and Z, Y' and Z'. Of these, Y and Y' contain p as a factor, Z and Z' some higher power p^s; then $Y = yp$ lies between the straight lines AB and AD, Fig. 1, while $Z = zp^s$ reaches from line AD to curve AC.

8 The complete equations now take the form

$$pv = BT - Y - Z = BT - yp - zp^s \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

$$h = h_\sigma - Y' - Z' = h_o - y'p - z'p^s \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

and are further illustrated by dimensions along a single ordinate in Figs. 3 and 4. These are in effect large-scale details from Figs. 1 and 2, showing the ordinate from ideal gas over to saturation at 400 deg. fahr. The dimensions above the lines come from the new equations of the paper, those below from the Goodenough equations of 1914. The latter are given at this point in order to show the introduction of another term into the general formulas. Basing their argument upon reasoned conditions, especially for the high densities at and near the critical state of the substance, most writers have given the p-v-T equation the form,

$$p(v - b) = BT - X, \quad \text{or} \quad pv = BT + bp - X \quad . \quad . \quad [9]$$

The quantity b is supposed to be the combined volume of the molecules, taken to agree with the volume of the liquid at its greatest density, here called w_0. A discussion of the use of this "covolume" term will be found in Par. 61.

9 It must be understood that Equations [7] and [8] are not limited mathematically by the saturation lines, but that this limit is imposed independently by substituting coincident saturation values of t and p.

THEORETICAL RELATIONS

10 The X and X' portions of the general steam equations [6] or [7] and [8] are very definitely tied together by a necessary theoretical relation, most conveniently taken in the form

$$\left(\frac{dh}{dp}\right)_t = -A\left[T\left(\frac{dv}{dt}\right)_p - v\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

In words, $(dh/dp)_t$ is the rate of change of h with p along the isotherm, or the slope of curve A'C' from the vertical in Fig. 2; and $(dv/dt)_p$ is the rate of change of v with t, or the rate of expansion, under constant pressure. The effect of applying this relation, fully worked out in the Appendix, is as follows:

11 The y and y', z and z' factors in Equations [7] and [8] are inverse functions of T, conveniently and satisfactorily taken as

$$y = \frac{E}{T^m}; \qquad y' = \frac{E'}{T^m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

$$z = \frac{F}{T^n}; \qquad z' = \frac{F'}{T^n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

E and E', F and F' being constant coefficients. Through Equation [10] are then derived the ratios

$$\frac{y'}{y} = \frac{E'}{E} = k_1 = (m+1)A \quad . \; . \; . \; . \; . \; . \; . \; . \quad [13]$$

$$\frac{z'}{z} = \frac{F'}{F} = k_2 = \frac{(n+1)}{s}A \quad . \; . \; . \; . \; . \; . \; . \quad [14]$$

factor A being the constant expressed in Equation [3]. Less simple forms were used in the writer's equations of 1913, namely,

$$y = \frac{E}{(T-e)^m}; \qquad z = \frac{F}{(T-f)^n} \quad . \; . \; . \; . \quad [15]$$

the effect is to add an intricate second term to the value of k_1 and of k_2 in Equations [13] and [14].

12 It is apparent that to make Equations [7] and [8] definite is a question of fixing the five constants E, F, m, n, and r; but with three of them involved as exponents the obvious method of simultaneous equations is barred out. The only available procedure is the laborious one of successive trial solutions. The data are neither precise nor extensive enough for close guidance, still less so for precise definition; and a highly important and useful determinant, in the close adjustment of constants, is furnished by Clapeyron's law. This relation, which holds at and along the saturation line, is commonly expressed as

$$\frac{r}{u} = AT\left(\frac{dp}{dt}\right)_s \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [16]$$

but it is more convenient to use the ratio

$$c = \frac{r}{pu} = \frac{AT}{p}\left(\frac{dp}{dt}\right)_s; \quad \text{or} \quad c = \frac{r}{Apu} = \frac{T}{p}\left(\frac{dp}{dt}\right)_s \quad . \; . \; . \quad [17]$$

The first form is good with our English units, where r and pu are so nearly alike in magnitude that c ranges from 3.66 to 1.38. With metric-centigrade units it is better to use r/Apu, ranging from 20 to 7.5; moreover this latter form is absolute or independent of scale of measurement, being taken between quantities that are expressed in the same unit.

13 A useful means of applying the indirect data obtained by throttling experiments is supplied by the relation

$$\left(\frac{dh}{dp}\right)_t = -\left(\frac{dh}{dt}\right)_p\left(\frac{dt}{dp}\right)_h = -c_pM \quad . \; . \; . \; . \; . \; . \quad [18]$$

which was fully employed in the writer's paper of 1913. (See also Par. 52.) It gives an initial indication of the size and manner of variation of function y' in Equation [11] and furnishes a rigorous and searching final test of the correct form of the heat equation.

DATA AND PROCEDURE

14 Referring to Figs. 8 to 10 in order to see the ranges named, the situation in regard to physical data may be summed up as follows:

(*a*) For liquid water there is dependable knowledge of volume and heat content up to 600 deg. fahr. or 1500 lb. pressure.

(*b*) The pressure-temperature relation of saturated steam has been accurately determined up to its limit at the critical temperature of over 700 deg. fahr., or to a pressure of 3200 lb.

(*c*) For dry saturated and superheated steam good volume and heat data exist up to 400 deg. fahr. saturation temperature. On Figs. 8 and 9 this range looks fairly high, but in Fig. 10 the corresponding pressure limit of about 250 lb. is seen to be relatively very low; however, an extension of the steam formulation far into the region of high pressures is not nearly so extreme and unguided a case of extrapolation as it may appear.

15 In the matter of formulation and tabulation the following classification may be made:

(*a*) The older steam tables, including all based wholly or in part upon the Regnault data, presented almost independent interpretations of the several lines of experiment. The data were not accurate enough to give really determinative effect to the theoretical interrelations. Those tables were restricted to the range of experiment, or to the limit of pressure in standard steam engineering practice, stopping at about 250 lb. abs.

(*b*) Using better data in a method of combined graphical and numerical layout, tables have been made that are of good reliability up to 1000 lb. pressure or to 550 deg. saturation temperature. This description covers the Marks and Davis Tables and one prepared by the writer for a textbook in 1909.

(*c*) With full use of theoretical relations in a method of definite mathematical formulation, consistent results can be obtained and the several groups of data be made to react upon and check each other. By such a synthesis of information the extension of the steam table to high pressures has its element of guesswork reduced to a minimum.

16 In the definition of a physical variable of the type whose law must be fixed empirically there are three important considerations, namely, absolute accuracy, consistency of expression or smoothness of variation, and correctness of trend. The last is especially important where passing from a known region into one of extrapolation. The volume and heat data for steam, while essentially correct, are not very precise, showing irregularities of the order of 0.25 per cent on either side of a fair mean. Over the range of experiment all the better formulations lie within the band of 0.5 per cent width thus located, but they show annoying discrepancies and differ pretty strongly in their trend and course into the region of high pressures.

17 Listed in chronological order, the formulations of steam properties to which more or less attention is now to be given are as follows:

(R) Tables based on the Regnault data, here represented in a few general comparisons by curves from the Peabody tables of 1907, the latest published under the older data.

(M) The equations proposed by Callendar in 1900 and adopted by Mollier for his tables issued in 1906; these were republished in English by Smith and Warren in 1912.

(C) With a slight adjustment of constants, Callendar himself embodied those same equations in a set of tables published in 1915. This formulation is consistent under Equation [10] but it does not conform to the new data that have appeared from 1905 to 1915, nor does any attempt seem to have been made to satisfy Clapeyron's relation.

(L) Linde in 1905 proposed a p-v-T equation to represent the Knoblauch-Linde-Klebe volume experiments at Munich, but on the side of heat content he did not go beyond a suggestion as to specific heat. The writer has developed and tried out the corresponding h-p equation, with results of no final value. The Linde formula, metric-centigrade units, is

$$pv = 0.00471T - (p + 0.02p^2)\left[0.031\left(\frac{373}{T}\right)^3 - 0.0052\right] \quad . \; . \; [19]$$

In our general form Equation [7], this makes $y = (E/T^m - b)$; and as an expedient for getting a desired manner of variation over a narrow range of experiment, constant b is taken so large that the whole term Y will pass through zero and become negative at a not very high temperature.

(MD) The Marks and Davis tables, published in 1909.

(H1) The writer's table of 1910 in The Steam Fngine and Turbine. For saturated steam this agrees with Marks and Davis, but is more smoothly worked out for high pressures, with full recognition of the importance of the isotherms of superheated steam.

(H2) The writer's equations of 1912, submitted to The American Society of Mechanical Engineers in 1913 and printed in Vol. 35 of Transactions, have not been put into published tables, as they involve the complicating mathematical expedient partly shown by Equation [15], and appear here only in a few comparisons.

(G) The GOODENOUGH equations (1914) and tables (1915) have a prominent place in the discussion. It should be noted that these are revised from the form presented to The American Society of Mechanical Engineers in 1912.

(H) The writer's latest equations, offered in this paper.

18 In the development of a set of steam formulas the order of procedure is as follows:

(*a*) Establish the pressure-temperature relation for saturated steam and get the Clapeyron ratio c.

(*b*) Fix pw and q in final form up to, say, 500 deg., tentatively up to the critical temperature.

(*c*) From specific-heat experiments and throttling data get a working expression for c_{po}, with resultant manner of variation of h_o.

(*d*) Use throttling data, as far as possible, to help fix the constants of the general equations.

(*e*) By trial and approximation determine these two general equations, taking Clapeyron's law as the close determinant, but checking continually against the direct experimental data and incidentally fixing the absolute value of h_o.

(*f*) Come back to saturation and make definite the final form of the curves shown in Figs. 8 and 9.

19 With emphasis upon results rather than their derivation, the presentation and comparisons now to be made will follow the different order usual with steam tables, first considering saturated steam in full, then taking up superheated steam and discussing the several phases of that subject.

II SATURATED STEAM

PRESSURE-TEMPERATURE RELATION

20 For the relation between pressure and temperature the equation of Goodenough may be accepted as final. In Figs. 5 and 6

it is compared with the vital data and with two other equations. The experiments plotted are those of

(SH) Scheel and Heuse, 1910...........0 to 50 deg. cent.
(HH) Holborn and Henning, 1908.....50 to 200 deg. cent.
(HB) Holborn and Baumann, 1910.....200 to 375 deg. cent.

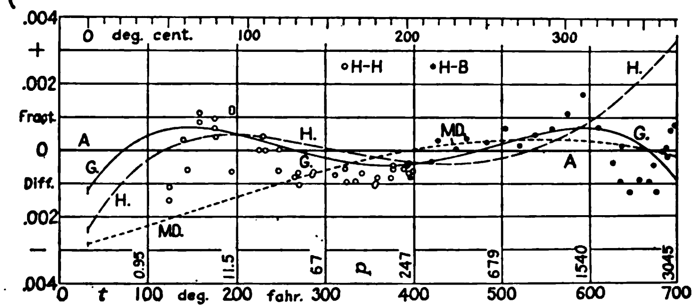


FIG. 5 FULL-RANGE PRESSURE-TEMPERATURE COMPARISON

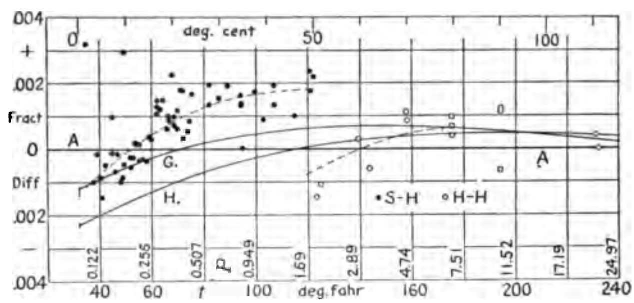


FIG. 6 LOW-RANGE COMPARISONS

All were made at the Reichsanstalt and published in the years indicated. The equations compared are those of Marks (MD) and the writer's (H) modification of the same. These are of the form

$$\log p = A - \frac{B}{T} - CT + DT^2 \quad . \; . \; . \; . \; . \; . \quad [20]$$

and for p in lb. per sq. in. and T in deg. fahr. the two sets of constants are:

	(MD)	(H)
A	10.515354	10.606400
$\log B$	3.6878597	3.6897500
$\log C$	7.6075579 − 10	7.6205462 − 10
$\log D$	4.1439399 − 10	4.1601803 − 10
	$T = t + 459.64$	

21 Goodenough puts one more term into the main expression and adds a small supplementary or corrective term, getting the general forms:

$$\log p = A - \frac{B}{T} - C \log T - DT + ET^2 - \Delta \quad . \; . \; . \; . \quad [21]$$

$$\frac{dp}{dt} = 2.3026p\left(\frac{B}{T^2} - \frac{C'}{T} - D + 2ET - \frac{d\Delta}{dt}\right) \quad . \; . \; . \; . \quad [22]$$

$$C' = C + 2.3026.$$

The constants of this Goodenough equation are:

Constant	English-Fahrenheit	Metric-Centigrade
p	lb. per sq. in.	kg. per sq. cm.
A	10.568808	9.411852
$\log B$	3.6881209	3.4328484
C	0.0155	0.0155
$\log C'$	7.82811 − 10	7.82811 − 10
$\log D$	7.6088020 − 10	7.8640745 − 10
$\log E$	4.1463000 − 10	4.6568450 − 10
T	$t + 459.6$	$t + 273.11$
Δ	$0.0^42(10 - 10\theta^2 + \theta^4)$	$0.0^42(10 - 10\theta^2 + \theta^4)$
$\frac{d\Delta}{dt}$	$0.0^42(-0.2\theta + 0.04\theta^3)$	$0.0^436(-0.2\theta + 0.04\theta^3)$
θ	$\frac{t - 370}{100}$	$\frac{t - 187.78}{55.556}$

The values of p and c given in Tables 4 and 5 are calculated by means of these formulas, the latter with heat equivalent A numerically as in Equation [**3**].

22 The plotted diagrams, Figs. 5 and 6, show proportional or fractional differences in p to large scale on base t, the vertical unit representing 0.1 per cent or one part in one thousand. The values of p written at the bottom of each ordinate are intended to give an idea of the absolute values thus shown, without the trouble of turning to the table. The base or standard, appearing as straight line AA, is the Goodenough equation without its term Δ. Below 300 deg. fahr. the (MD) curve runs low. The (H) equation was made to remedy this fault, but was not intended to be used above 550 deg. or about 1000 lb. pressure. The (G) curve traces throughout a very fair mean of an experimental band that, except for a few erratic points, has a total width of about 0.25 per cent. The only considerable discrep-

ancy shows up in Fig. 6, where dotted curves are sketched in to show the trend of the two sets of observations; at 50 deg. cent. there is a distinct gap of 0.25 per cent, which is very fairly divided by the (G) curve.

23 In similar fashion, but with a base line representing the complete (G) equation and with less magnification of difference percentages, the relative standing of the Regnault and the Callendar equations is shown by Fig. 7. The second of these is not a formulation of direct experiment, but is derived through a rather abstruse theoretical relation from heat and entropy quantities. It is good up to 200 deg. cent., but beyond that is of no value.

PROPERTIES OF THE LIQUID

24 These are represented by the liquid lines AAK in Figs. 8 and 9, which show pw and q, respectively. For neither has a single formula been found to cover the whole range up to critical point K,

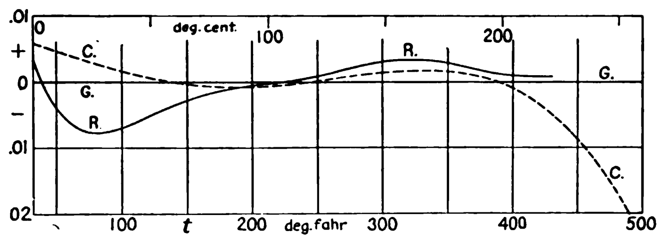


FIG. 7 COMPARISONS OF REGNAULT AND CALLENDAR PRESSURES WITH THE GOODENOUGH FORMULA

but for both the method here employed is to fix a middle-range equal tion, make such adjustment as may be needed toward the lower limit at freezing point, and adopt a distinct high-range equation for the accelerated curvature over to K. In both diagrams AAA′ is the curve of the lower equation, detail (a) shows adjustment at the bottom, KAA″ represents the high-range equation, and the merging of the two is analyzed at (b). Of the Equations [**23**] to [**26**] in this section all but [**24**] are original with the paper, in form as well as constants.

25 For water volume the first equation makes w itself a function of t, leaving pw to be obtained by multiplication: the formula, with English-fahrenheit (EF) and metric-centigrade (MC) constants, is

$$\text{(EF)}\quad w = 0.015896 + 0.0^{5}3084(t-32) + 0.0^{8}374(t-32)^2 + 0.0^{10}2334(t-32)^3 \quad \textbf{[23a]}$$

$$\text{(MC)}\quad w = 0.0^{3}9924 + 0.0^{6}3465t + 0.0^{9}7565t^2 + 0.0^{11}8498t^3 \quad \textbf{[23b]}$$

Over its effective range this equation conforms to the accepted tabular column of w, as given in the Marks and Davis tables; but below

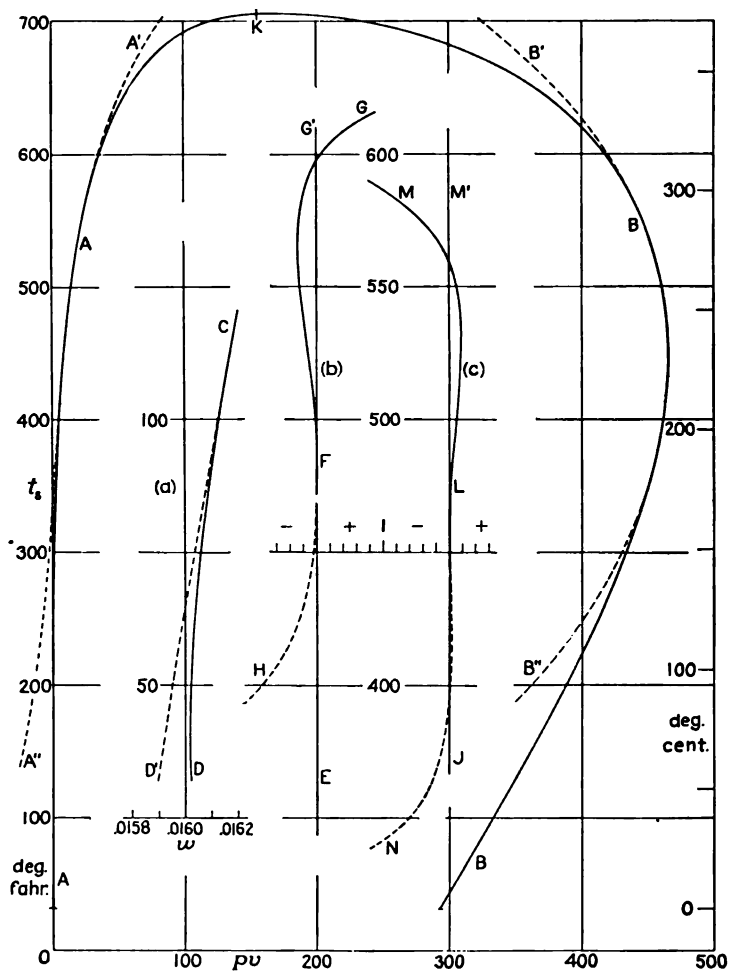


FIG. 8 PRESSURE-VOLUME DIAGRAM FOR SATURATED STEAM

Main plot, pw in AAK, pv_s in BBK; secondary diagrams, here and in Fig. 9, illustrate details of formulation.

110 deg. fahr. it diverges as shown by dotted curve CD′, Fig. 8(a), from the tabular curve CD.

26 The corresponding equation for heat of liquid q is based on Dieterici's formula for specifc heat $c_w = dq/dt$, with slightly modified constants, namely,

(EF) $c_w = 0.9990 - 0.0^4 8556(t - 32) + 0.0^6 7315(t - 32)^2$ . . [24a]
(MC) $c_w = 0.9990 - 0.0^3 154t + 0.0^5 237t^2$ . . . . [24b]

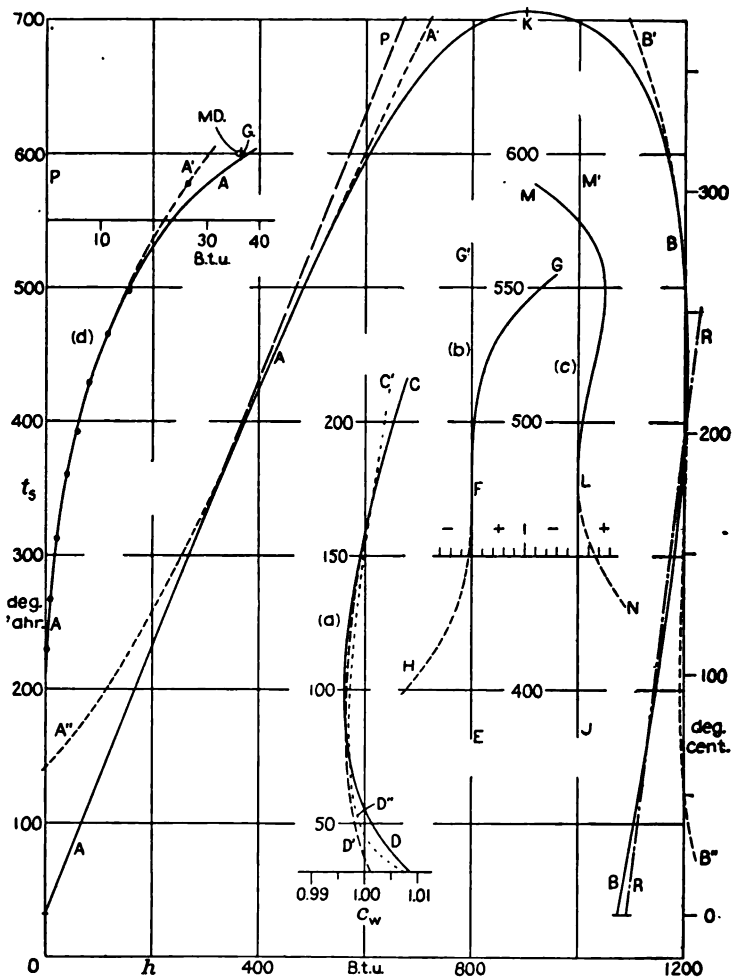


FIG. 9 HEAT-CONTENT DIAGRAM FOR SATURATED STEAM

Heat of liquid q in AAK; total heat h in BBK

This is shown by the dash line CD′ in Fig. 9, but below 212 deg. it is replaced by curve CD, carefully adjusted so as to make q equal to 180 at 212 deg. For comparison the Callendar and Barnes curve is dotted in as C′D″. Up to 180 deg. q as tabulated comes from curve CD, then it is given by the equation

$$(\text{EF})\quad q = 0.144 + 0.999(t - 32) - 0.0^{4}4278(t - 32)^{2} + 0.0^{6}2438(t - 32)^{3} \quad \ldots\ldots\ldots\ldots \quad [\mathbf{25a}]$$

$$(\text{MC})\quad q = 0.080 + 0.999t - 0.0^{4}77t^{2} + 0.0^{6}79t^{3} \quad \ldots\ldots\ldots \quad [\mathbf{25b}]$$

27 For the high-range curves of pw and q, as also of pv_s and h_s, a common general form is used, namely,

$$x = A + B(t - 32) + C(t - 32)^{0.5} + D(t_c - t)^{0.5} \quad . \ . \quad [\mathbf{26}]$$

Defining all four curves, since it will be a simple matter to refer back when we presently take up the saturation lines, and grouping them according to physical relation, the constants for English-fahrenheit units, with $t_c = 706.3$, are:

		A	B	C	D
(a)	pw	+ 313.16	− 0.73734	+ [1.11581]	− [1.21677]
(b)	pv_s	− 979.48	+ 0.02113	+ [1.63748]	+ [1.53924]
(c)	q	+ 222.25	− 1.2471	+ [1.76704]	− [1.46650]
(d)	h	+ 316.63	+ 1.7037	− [1.34090]	+ [1.58874]

The special notation introduced under C and D consists in representing a number by its logarithm within brackets. In order to convert these pv formulas to metric-centigrade units, divide A by [2.35762], B by [2.10235], C and D by [2.22998]. To convert q and h_s, divide A by 1.8, keep B the same, divide C and D by $\sqrt{1.8}$ or [0.12764].

28 The merging of these high curves with the lower ones is given a very severe test by the magnified plots of differences at (*b*) and (*c*) in Figs. 8 and 9, where the multiplying factor is 50 for horizontal but only 2 for vertical dimensions of the main diagram. The lower curve is rectified as base, appearing in lines EG′ and JM′, while the higher curves take the form GFH, MLN. Although the quantities involved are small, the matching is by no means ideal; and it appears that a better effect could be got by using some such form as $(t' - t)^{0.5}$ instead of $(t - 32)^{0.5}$ in the third term of Equation [26], with 752 deg. fahr. or 400 deg. cent. for a suggested value of t'.

29 In diagram (*d*) of Fig. 9 the q curve is compared with Dieterici's experiments, by plotting the excess of heat units over degrees, or the quantity $[q - (t - 32)]$ — this difference lying between curve AA and straight line AP of the main figure. The dotted

curve of Equation [25a] agrees excellently with all of the data points, but the full-line curve of Equation [26c] swings rather sharply away from the highest one. For a comparative showing two single points, from the (MD) and the (G) tables, respectively, are laid down on the 600-deg. line.

30 The main purpose in fixing these high-range curves is not to get results that in themselves are of much practical utility, but rather to test the upper trend of the curves rising from the region of experiment. This matter is of greater interest on the side of the saturation lines, where the limit of experiment is comparatively low.

THE SATURATION LINES

31 With regard to the curves of pv_s and h_s, marked BBK in Figs. 8 to 10, the immediate question is one of fact, namely, where do the curves lie and how do the several formulations compare? This is answered in Figs. 8 to 12, leaving the further question as to how the curves are determined to be discussed in the last section of the paper: see Par. 46.

32 All of the formulations listed in Par. 17 are finally brought into the comparison, according to the following program:

(*a*) In Figs. 8 and 9 the two (H) curves are laid down alone, except for the introduction of Regnault's straight line of h_s. As carried up to B′ they show the result of a full-range evaluation of Equations [27d] and [28d] with saturation values of t and p. The high-range curves BK have just been defined in Par. 27.

(*b*) Fig. 10 is primarily a redrawing of the (H) curves on a different base, with enlarged scale of h_s: a new element, introduced for later reference, is the showing of the Clapeyron check by the dotted curve AA, along h_s above 1800 lb. The three important sets of published steam tables are represented by plots in this high range; and to them are added the only data available, which are the Battelli high pv isotherms. These are not of determinative accuracy, but the pv_s curve conforms well enough to them.

(*c*) Returning to t_s as base, a large-scale direct plot of all the formulations is made in Fig. 11, running up to 600 deg. and giving a very good general view of conditions.

(*d*) Finally, magnified differences are shown in Fig. 12, over the data range up to 400 deg. and with the (H) curves rectified as reference lines. Experimental determinations of h_s are plotted; those of pv_s are taken to be sufficiently represented by the (L) curve.

33 The prominent results of this comparison may be briefly summarized, leaving a detailed study of the diagrams to those especially interested. It will be better to reverse the chronological order of Par. 17, running in a general way from agreement toward divergence.

(G) A notable fact is the close agreement between the (G) and (H) curves: further comment on this point will be found in Par. 57.

(H2) The writer's formulas of 1913 agree closely with the latest equations in the low range, but above 400 deg. they run high in both volume and heat content.

(H1) These curves of 1909 lie with those of Marks and Davis, except from 450 to 550 deg. in pv_s.

(MD) The path of the pv_s point sags to the left around 500 deg., then recovers and runs fairly with (G) and (H) until it turns (in Fig. 10) toward a K point at the formerly accepted critical temperature of 689 deg. In h_s the tabular column is not carried beyond 600 deg.: note that around 300 deg. it lies inside of the (G) and (H) formulations, then above 400 deg. runs consistently larger.

(L) As indicated in the last paragraph, the chief use of the Linde curve (pv_s only) is to serve as the interpretation of a series of experiments. The computed extension beyond 400 deg. is merely a matter of interest in its trend.

(C) Up to 500 deg. the (C) curves represent Callendar's tables, then there is a little break because the extensions are computed with p as related to t by Equation [26] — compare Fig. 7. It is evident that these curves completely fail to meet the requirement of correct trend and course toward a probable critical point.

(R) The Regnault curves are put in merely to show the relative location of the older tabulations.

34 Most of the plotted data points for h_s in Fig. 12 really represent determinations of latent heat r, to which heat of liquid q has been added. The older experiments were made by water calorimetry; the more recent ones involve electrical measurement of energy supplied to produce vaporization. The Dieterici determination at 32 deg. fahr. was made by a special method — that of the Bunsen ice calorimeter — and serves as an extreme illustration of an uncertainty as to exact ratio of energy units that generally exists to some degree: the horizontal heavy line with stopped ends, about 2.5 B.t.u. long, shows the range of possible interpretation of this result. The numerous Regnault observations are represented by group averages made up by Smith.

35 It is generally accepted that in interpreting his experiments and proposing a straight-line formula for h_s on t, Regnault was misled by the two high points between 350 and 400 deg. Leaving these out, straight line R′R′ is in this figure a very fair mean of the other observations. Similarly, if we disregard the one Henning point

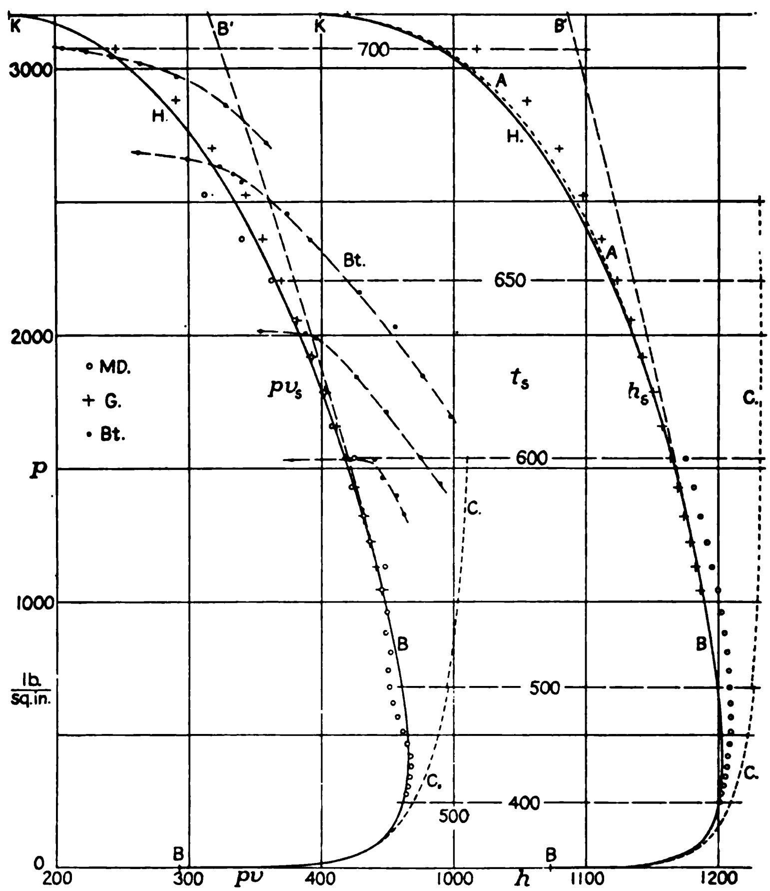


FIG. 10 DIAGRAM OF pv_s AND h_s ON PRESSURE BASE, WITH HIGH-RANGE COMPARISONS

above 350 deg., line H′H′ is an excellent mean of the more recent experiments. There is good reason for locating the absolute saturation line to the right of this observational line, in the extreme physical difficulty of producing perfectly dry steam. Light is thrown upon this matter by the two Smith points S_1 and S_2 near 212 deg.:

under conditions otherwise the same, the first was obtained with rapid ebullition, the other with slow vaporization—and there is a difference of nearly 8.5 B.t.u. between them. It seems probable that as pressure rises and density increases this difficulty in attaining absolutely dry steam (without superheat) will become greater,

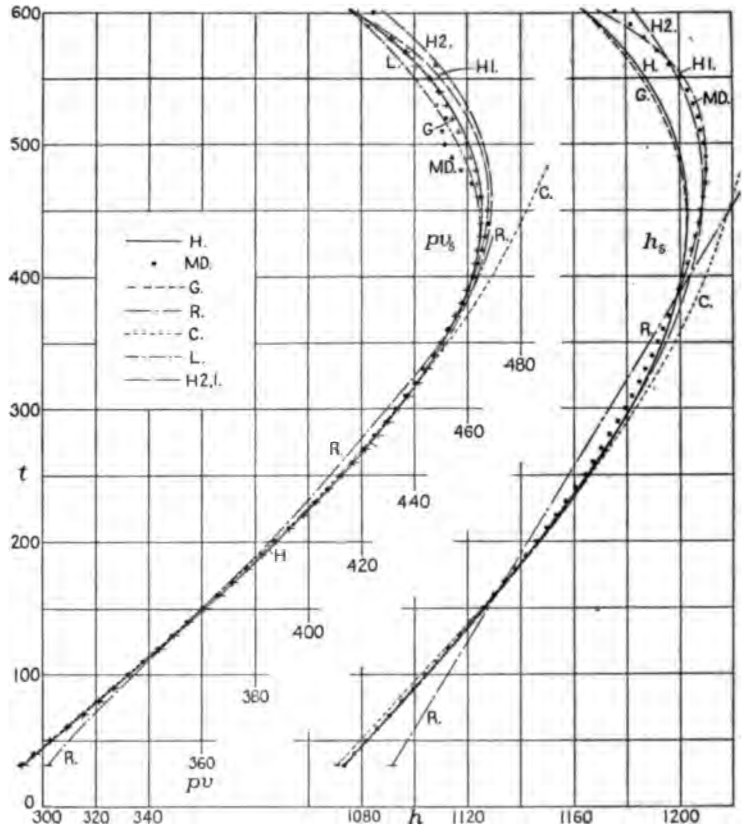


FIG. 11 COMPARATIVE DIRECT PLOTS OF THE SATURATION LINES ON TEMPERATURE BASE

and that toward the critical state the saturation line is not nearly so sharp a locus of change of state as the diagrams and formulas imply.

III SUPERHEATED STEAM

THE GENERAL EQUATIONS

36 Repeated from Equations [7] to [12], with the covolume term included so as to make a complete general form, the two characteristic equations are

$$pv = BT + pw_o - \frac{E}{T^m}p - \frac{F}{T^n}p^s \quad . \; . \; . \; . \; . \; [27]$$

$$h = h_o + Apw_o - \frac{E'}{T^m}p - \frac{F'}{T^n}p^s \; . \; . \; . \; . \; . \; . \; [28]$$

The constants belonging to the important complete sets of general equations, all reduced to this form and including the ratios expressed

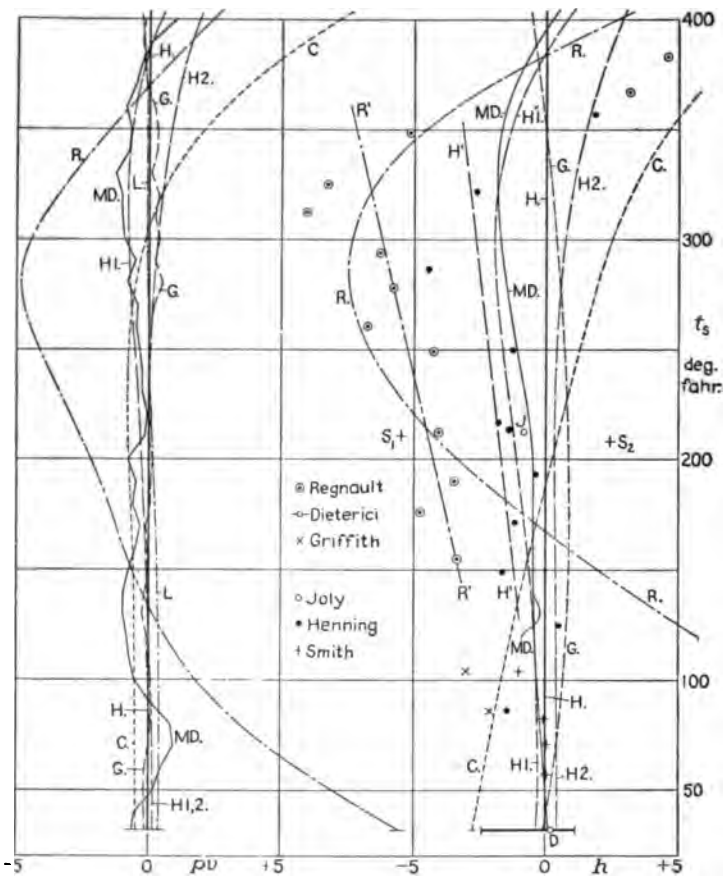


FIG. 12 PLOTS OF MAGNIFIED DIFFERENCES, WITH THE (H) CURVES AS REFERENCE LINES

by Equations [13] and [14], are collected in Tables 1 and 2, lines 1 to 15, with distinguishing letters at bottoms of columns; note that the Callendar-Mollier equations do not contain the term in p^s, so that their isotherms are straight lines, and that the covolume term is omitted from the Heck equations.

TABLE 1 CONSTANTS FOR THE GENERAL FORMULAS, ENGLISH-FAHRENHEIT UNITS

No.	Constant	Mollier (M) 1906	Callendar (C) 1915	Goodenough (G) 1914	Heck (H) 1920
1	$T_0 = T - t$	459.4	459.58	459.6	459.6
2	J = ft-lb. per B.t.u.	778.28	777.93	777.64	778
3	$A = 144/J$	0.18502	0.18510	0.18517	0.18509
4	$B = pv/T$	0.5949	0.5948	0.59485	0.5952
5	w_0 = cu. ft. per lb.	0.01602	0.01602	0.01602 [1]	Not used
6	Aw_0 = B.t.u. per lb.	0.00296	0.00296	0.00296	
7	Exponent m	10/3	10/3	4	3.5
8	Exponent n	....	....	4	7.2
9	Exponent s	....	....	1.5	2
10	Ratio k_1	0.8018	0.8025	0.9259	0.8329
11	Ratio k_2	....	....	0.6172	0.7589
12	log E	9.05112	9.01826	10.82500	9.47041
13	log F	....	....	9.53500	17.37749
14	log E'	8.95516	8.95261	10.79155	9.39100
15	log F'	....	....	9.32546	17.25765
16	At 32 deg. fahr. h_s =	1070.4	1069.70	1073.06	1073.22
17	At 212 deg. fahr. h_s =	1150.4	1150.74	1151.72	1151.00
18	At 212 deg. fahr. q =	180.8	180.00	180.00	180.00
19	At 212 deg. fahr. r =	969.4	970.74	971.72	971.00
	Equations [27], [28]	(a)	(b)	(c)	(d)

[1] See Par. 59.

TABLE 2 CONSTANTS FOR THE GENERAL FORMULAS, METRIC-CENTIGRADE UNITS

No.	Constant	Mollier (M) 1906	Callendar (C) 1915	Goodenough (G) 1914	Heck (H) 1920
1	$T_0 = T - t$	273	273.1	273.11	273.11
2	J = m-kg. per cal.	427	426.7	426.65	426.84
3	$A = 10{,}000/J$	23.419	23.436	23.438	23.428
4	$B = pv/T$	0.004700	0.004699	0.0046995	0.0047023
5	w_0 = cu. m. per kg.	0.001	0.001	0.001 [1]	Not used
6	Aw_0 = cal. per kg.	0.0234	0.0234	0.0234	
7	Exponent m	10/3	10/3	4	3.5
8	Exponent n	....	....	4	7.2
9	Exponent s	....	....	1.5	2
10	Ratio k_1	101.48	101.55	117.19	107.14
11	Ratio k_2	....	....	78.13	96.05
12	log E	6.99560	6.99273	8.59929	7.37233
13	log F	....	....	7.88579	15.48790
14	log E'	9.00199	8.99943	10.66819	9.39527
15	log F'	....	....	9.77860	17.47041
16	At 0 deg. cent., h_s =	594.66	594.27	596.12	596.23
17	At 100 deg. cent., h_s =	639.6	639.30	639.84	639.44
18	At 100 deg. cent., q =	100.5	100.00	100.00	100.00
19	At 100 deg. cent., r =	539.1	539.30	539.84	539.44
	Equations [27], [28]	(a)	(b)	(c)	(d)

[1] See Par. 59.

37 The last four lines of Tables 1 and 2 constitute a supplement, exhibiting the results of evaluating for saturation heat content at 32 or 0 deg. and 212 or 100 deg. — these exact numerical results being of interest in connection with the general comparisons of Figs. 11 and 12. As a further and more detailed illustration, the three sets of formulas are evaluated by parts for saturated steam at 392 or 200 deg. in Table 3. Present interest centers on the results in lines 5, 6 and 11, but the relative size of terms Y and Z, Y' and Z' will be a matter of importance in the dis-

TABLE 3 COMPARATIVE EVALUATION OF THREE PAIRS OF EQUATIONS, FOR SATURATED STEAM

No.	Constant	English-Fahrenheit t = 392 deg., p = 225.31 lb.			Metric-Centigrade t = 200 deg., p = 15.840 kg.		
		(C)	(G)	(H)	(C)	(G)	(H)
1	BT	506.51	506.57	506.89	2.2231	2.2234	2.2248
2	pw_o	3.60	3.60	....	0.0158	0.0158	....
3	Y	43.00	28.62	36.93	0.1887	0.1256	0.1621
4	Z	....	22.04	9.67	....	0.0967	0.0424
5	pv	467.11	459.51	460.28	2.0502	2.0169	2.0203
6	v	2.0732	2.0395	2.0429	0.12642	0.12733	0.12754
7	h_o	1241.57	1239.05	1238.36	689.76	688.36	687.98
8	Apw_o	0.67	0.67	....	0.37	0.37	....
9	Y'	34.51	26.51	30.75	19.17	14.72	17.09
10	Z'	....	13.60	7.34	....	7.56	4.07
11	h	1207.73	1199.61	1200.27	670.96	666.45	666.82
12	y	0.1909	0.1271	0.1639	0.01192	0.00793	0.01023
13	y'	0.1532	0.1177	0.1365	1.2104	0.9297	1.0787
14	c_{po}	0.4772	0.4598	0.4624	0.4772	0.4598	0.4624
15	M_o	0.2310	0.2559	0.2952	2.536	2.022	2.333

cussion of form of equations. The last four lines, which lead to a comparison with the throttling data, are also for later reference.

VOLUME AND HEAT DATA

38 In Fig. 13 isotherms from the three pv formulas are compared with the Knoblauch-Linde-Klebe experiments on steam volume — see (L), Par 17. Each row of dots, rising slowly toward the right, represents a series of observations at constant volume. The real base (temperature) does not appear in this diagram, but knowing the temperature associated with each point it is a simple matter to interpolate points (marked by crosses) that correspond with the isotherms drawn. Larger dots on the ordinate lines are

points from the Linde formula, Equation [**19**]. What has been said in Par. 16 as to degree of precision of data has here a good illustration, for the differences between the (G) and (H) curves lie well within the range of experimental uncertainty. It is clearly shown, however, that the (M) or (C) straight isotherms are not correct. A single row of Battelli observations is included, showing his determination of an isotherm at 361.3 deg. fahr., with a decided deficiency in slant toward the saturation line. The Ramsay and Young experiments, tried out by a similar plot, rank low in absolute accuracy but in trend agree well with the latest formulations and are decidedly against the straight isotherm.

39 At the limiting state of zero pressure specific heat c_{po} and heat content h_o are expressed as follows:

MOLLIER

$$c_{po} = 0.477 \quad \ldots\ldots \quad [\mathbf{29}]$$
$$(FF)\ h_o = 1055.20 + 0.477t \quad \ldots\ldots \quad [\mathbf{30a}]$$
$$(MC)\ h_o = 594.70 + 0.477t \quad \ldots\ldots \quad [\mathbf{30b}]$$

CALLENDAR

$$c_{po} = 0.47719 \quad \ldots\ldots \quad [\mathbf{31}]$$
$$(EF)\ h_o = 1054.41 + 0.47719t \quad \ldots\ldots \quad [\mathbf{32a}]$$
$$(MC)\ h_o = 594.32 + 0.47719t \quad \ldots\ldots \quad [\mathbf{32b}]$$

GOODENOUGH

$$(EF)\ c_{po} = 0.320 + 0.0^3126T + 23{,}583/T^2 \quad \ldots\ldots \quad [\mathbf{33a}]$$
$$(MC)\ c_{po} = 0.320 + 0.0^32268T + 7278/T^2 \quad \ldots\ldots \quad [\mathbf{33b}]$$
$$(EF)\ h_o = 1095.61 + 0.320t + 0.0^463T^2 - 23{,}583/T \quad \ldots \quad [\mathbf{34a}]$$
$$(MC)\ h_o = 614.36 + 0.320t + 0.0^31134T^2 - 7278/T \quad \ldots \quad [\mathbf{34b}]$$

HECK

$$(FF)\ c_{po} = 0.3526 + 0.0^3128(t - 32) + 54.92/T \quad \ldots\ldots \quad [\mathbf{35a}]$$
$$(MC)\ c_{po} = 0.3526 + 0.0^32304t + 30.16/T \quad \ldots\ldots \quad [\mathbf{35b}]$$
$$(EF)\ h_o = 725.57 + 0.3526t + 0.0^464(t - 32)^2 + 125 \log T \quad . \quad [\mathbf{36a}]$$
$$(MC)\ h_o = 427.09 + 0.3526t + 0.0^31152t^2 + 69.444 \log T \quad \ldots \quad [\mathbf{36b}]$$

Curves from Equations [**33a**] and [**35a**] are plotted in Figs. 14 and 15.

40 For the specific heat c_p at any pressure we readily get from Equation [**28**] the expression,

$$\left(\frac{dh}{dt}\right)_p = c_p = c_{po} + \frac{mE'}{T^{m+1}}p + \frac{nF'}{T^{n+1}}p^s \quad \ldots \quad [\mathbf{37}]$$

(G) and (H) evaluations of this are shown in Fig. 14, along the saturation line (c_{ps}) and under atmospheric pressure (c_{pa}). With the last are compared Regnault's four determinations, on which the old constant value $c_p = 0.48$ was based: the small high circles show

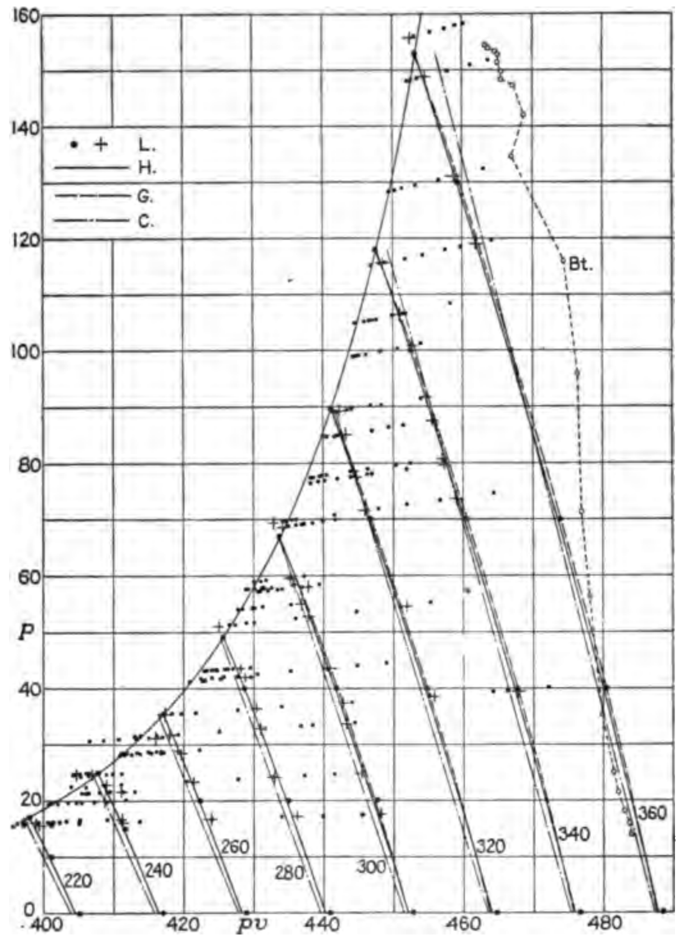


FIG. 13 COMPARISON OF (M) (G) AND (H) ISOTHERMS WITH THE MUNICH VOLUME EXPERIMENTS

Regnault's original results, the larger low circles Davis' recomputed values, while each horizontal line through a circle indicates the temperature range of its experiment. The one low point was considered less reliable by Regnault, but here helps to bring the average into close accord with the (G) and (H) curves.

41 Just as at all other pressures, the curve of c_{po} should rise toward the saturation line; but it is difficult to get a formula that will give this effect without making it too strong. So great a rise as the (G) curve shows in Fig. 14, with consequent marked initial drop of the c_{ps} curve, cannot be correct.

42 In order to compare the several formulations with respect to zero-pressure conditions, curves are drawn for a wide range of temperature in Fig. 15. First c_{po} is plotted directly, including the constant (C) value, then h_o is shown by its difference from the assumed reference line

$$h' = 1050 + 0.48t \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad [38]$$

which appears as base AA. Beside results from Equations [32a], [34a] and [36a], a Marks and Davis curve is obtained by a slight

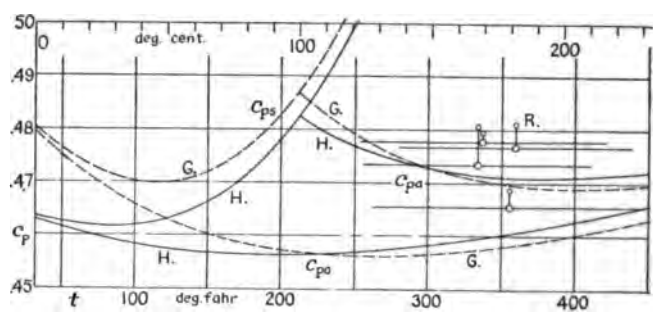


FIG. 14 SPECIFIC HEAT AT ZERO PRESSURE, ALONG SATURATION AND AT ATMOSPHERIC PRESSURE

adjustment of the h values in their tabular line for 1 lb. pressure. The lowest point thus found is at 101.7 deg.; the curve then runs up to about the same initial point (at 32 deg.) as the others.

43 A comparison of very great interest is made in Figs. 16 to 18, where separated, individual diagrams are drawn for each of the twelve pressures (from 0.5 to 20 kg. per sq. cm.) in the Munich experiments on specific heat. These were made and published in three groups, namely,

Knoblauch and Jakob, 1907. . 2, 4, 6, 8 kg., to 400 deg. cent.
Knoblauch and Mollier, 1911 . . . . . 2–8 kg., to 600 deg. cent.
Knoblauch and Winkhaus, 1915 . 0.5–20 kg., to 400 deg. cent.

44 In Figs. 16 and 17 there is first a direct plot of the experimental results, with a horizontal line to show the range of tempera-

ture in each measurement of heat added; in Fig. 18 this range line is omitted. Final corrected values are used, as given by the authors, and by a series of dots at each pressure the tabular line of c_p (from the latest publication and up to 380 deg. cent.) is laid down — thus showing the experimenters' interpretations of their own observations. With these data are compared (G) and (H) curves computed by means of Equation [37]. The close agreement at low pressures, up to 2 kg., strongly confirms the c_{po} curves in Fig. 15. At higher pressures the (K) curves seem to run a little low for their data points, which are just about as well represented by the (H) curves; with

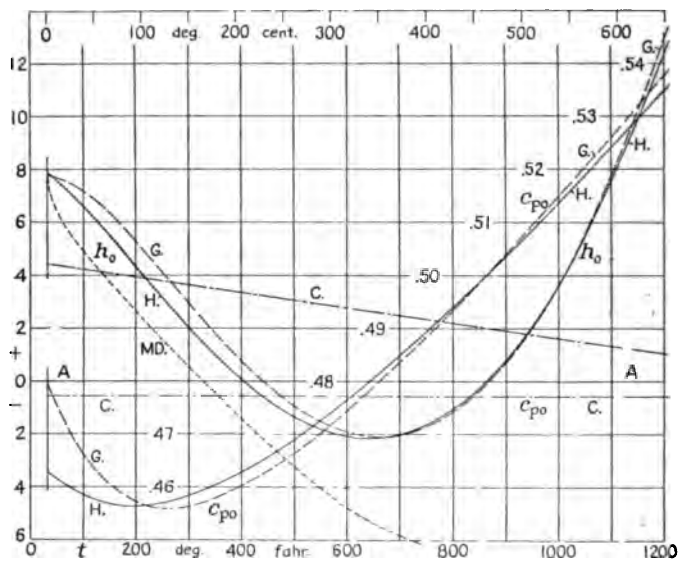


FIG. 15 WIDE-RANGE CURVES OF c_{po} AND h_o

respect to the latter, however, there is a slight degree of consistent discrepancy, which may be explained as follows:

(a) In these experiments the steam was carefully preheated to an initial temperature, then further heated by measured electrical energy. Near saturation it was very difficult to get rid of the last vestige of molecular aggregation, so that the observed increment of heat was likely to be just a little too large.

(b) The current of steam suffered some drop of pressure in its flow through the superheater; correction had to be made in order to get the effective range of temperature, smaller than that directly observed. For this the Mollier heat equation was used, and its deficient slant of the isotherms tends to make the t range yet a little too big, hence c_p a trifle low.

45 A very useful showing of comparative results as to heat content is made in Fig. 19, by plotting isotherms for 400, 500, 600, and 800 deg. fahr., up to 600 lb. pressure. In addition to lines from the three heat equations of Table 1, points interpolated in the Marks and Davis tables are plotted, showing a peculiar variation of curvature. This is due in part to the use of early Knoblauch c_p values, extrapolated (without any external check) far above their limit of 8 kg. pressure. The writer's contemporaneous (H1) curves are also drawn in, to show the effect of proper attention to the shape of the isotherm, even with those same data. At high temperatures the (MD) and (H1) values of h run low, because of dependence upon data (Holborn and Henning) that have been superseded by the experiments shown in Fig. 18.

IV FORM OF THE STEAM EQUATIONS

THE CLAPEYRON CHECK

46 Thus far we have been concerned with the first question in Par. 31, or with showing and comparing formulations and data. The operation of making definite the general forms of Equations [7] and [8] or [27] and [28], or of fixing constants as suggested in Par. 12, is now to be briefly described, with some discussion of the differences that have appeared in the comparisons just completed.

47 The working of Clapeyron's law as a determinant is visualized by Fig. 20, which is in effect a collection of ordinates like BC and B′C′ of Figs. 1 and 2, or is restricted to saturation conditions (at the physical limit of each isothermal). The general idea is to get from volume quality pu by Clapeyron ratio c a value r_v, then $h_v = q + r_v$, to be compared with h_s from the heat formula. The detailed scheme of Fig. 20, which is drawn for the (H) equations, [27d] and [28d], involves the application of factor c to all parts of the pv expression: thus we have—

$$\begin{aligned} h_v &= q + r_v = q + cpu = q + c(pv_s - pw) \\ &= (q + cBT - cpw) - cY - cZ = h' - cY - cZ \quad . \quad . \quad [39] \end{aligned}$$

to be plotted in comparison with

$$h_s = h_o - Y' - Z' = h_o - k_1Y - k_2Z \quad . \quad . \quad . \quad . \quad . \quad [40]$$

Dimensions along the ordinate at 1300 lb. pressure relate the diagram to these expressions; note that dotted curves with odd numbers

TABLE 4 PROPERTIES OF SATURATED STEAM AT EVERY 10 DEG. FAHR

Quantities defined in Par. 2, except c in Par. 12 and h_e in Pars. 6 and 39

t	p	log c	w	v_s	pw	pv_s	q	h_s	h_e
32	0.08870	0.56418	0.01602	3297.6	0.0014	292.50	0.00	1073.22	1073.30
40	.12173	.55550	1602	2441.6	.0019	297.23	8.05	1076.89	1077.00
50	.17802	.54483	1602	1702.9	.0027	303.14	18.09	1081.47	1081 62
60	0.25611	0.53431	0.01603	1206.6	0.0041	309.03	28.09	1086.03	1086.23
70	.36282	.52395	1605	867.9	.0058	314.90	38.07	1090.57	1090.83
80	.50657	.51374	1607	633.2	.0081	320.76	48.04	1095.09	1095.43
90	.69767	.50368	1610	468.1	.0112	326.59	58.01	1099.58	1100.02
100	.94861	.49377	1613	350.4	.0155	332.39	67.97	1104.04	1104.60
110	1.2740	0.48400	0.01616	265.44	0.0206	338.17	77.93	1108.47	1109.18
120	1.6915	.47437	1620	203.31	.0274	343.90	87.90	1112.87	1113.76
130	2.2214	.46489	1625	157.38	.0362	349.60	97.87	1117.22	1118.33
140	2.8873	.45554	1629	123.02	.0471	355.25	107.85	1121.55	1122.91
150	3.7163	.44632	1634	97.10	.0607	360.84	117.83	1125.82	1127.48
160	4.7391	0.43724	0.01639	77.310	0.078	366.38	127.83	1130.04	1132.04
170	5.9904	.42828	1645	62.045	.099	371.85	137.83	1134.21	1136.61
180	7.5090	.41946	1651	50.240	.124	377.25	147.85	1138.31	1141.17
190	9.3381	.41076	1657	40.969	.155	382.58	157.88	1142.36	1145.74
200	11.5253	.40220	1663	34.433	.192	387.82	167.92	1146.33	1150.30
210	14.1229	0.39376	0.01669	27.825	0.236	392.96	177.99	1150.23	1154.87
212	14.6966	.39209	1671	26.808	.244	393.98	180.00	1151.00	1155.78
220	17.1881	.38545	1676	23.156	.288	398.01	188.06	1154.05	1159.43
230	20.7828	.37726	1683	19.389	.350	402.95	198.16	1157.78	1164.00
240	24.9739	.36920	1691	16.328	.423	407.77	208.28	1161.40	1168.56
250	29.8343	.36126	1699	13.825	.507	412.47	218.42	1164.94	1173.13
260	35.439	0.35345	0.01707	11.768	0.605	417.03	228.58	1168.37	1177.70
270	41.869	.34577	1716	10.064	.719	421.45	238.77	1171.68	1182.27
280	49.215	.33821	1725	8.650	.850	425.72	248.98	1174.86	1186.85
290	57.567	.33078	1734	7.297	1.000	429.83	259.23	1177.95	1191.43
300	67.021	.32347	1744	6.472	1.169	433.77	269.50	1180.88	1196.01
310	77.681	0.31630	0.01754	5.6328	1.362	437.54	279.80	1183.68	1200.60
320	89.651	.30925	1765	4.9204	1.581	441.11	290.13	1186.33	1205.19
330	103.045	.30232	1776	4.3135	1.830	444.48	300.50	1188.81	1209.78
340	117.977	.29553	1788	3.7943	2.110	447.64	310.90	1191.14	1214.38
350	134.571	.28886	1800	3.3482	2.423	450.58	321.34	1193.28	1218.98
360	152.948	0.28232	0.01813	2.9637	2.773	453.30	331.82	1195.25	1223.58
370	173.243	.27592	1827	2.6308	3.165	455.77	342.33	1197.04	1228.19
380	195.588	.26964	1841	2.3416	3.600	457.98	352.89	1198.63	1232.81
390	220.123	.26349	1855	2.0894	4.083	459.93	363.49	1200.01	1237.43
400	246.992	.25748	1870	1.8689	4.619	461.59	374.13	1201.19	1242.06
410	276.34	0.25160	0.01886	1.6754	5.21	462.98	384.83	1202.14	1246.69
420	308.33	.24585	1902	1.4845	5.86	464.08	395.57	1202.87	1251.33
430	343.11	.24024	1919	1.3549	6.58	464.87	406.35	1203.35	1255.97
440	380.85	.23476	1936	1.2218	7.37	465.32	417.18	1203.60	1260.62
450	421.71	.22942	1954	1.1037	8.24	465.44	428.06	1203.56	1265.27

TABLE 4 — *Continued*

t	p	log c	w	v_s	pw	pv_s	q	h_s	h_v
460	465.87	0.22421	0.01973	0.9986	9.19	465.23	439.00	1203.30	1269.93
470	513.51	.21913	1992	.9049	10.23	464.67	449.99	1202.76	1274.60
480	564.79	.21419	.2011	.8212	11.36	463.75	461.03	1201.93	1279.27
490	619.93	.20939	2030	.7460	12.59	462.47	472.14	1200.82	1283.95
500	679.10	.20473	2050	.6785	13.92	460.80	483.33	1199.47	1288.64
510	742.52	0.20020	0.02069	0.6178	15.36	458.73	494.63	1197.84	1293.34
520	810.38	.19580	2088	.5630	16.92	456.24	506.05	1195.93	1298.04
530	882.90	.19155	2108	.5134	18.61	453.32	517.61	1193.69	1302.75
540	960.30	.18743	2129	.4695	20.44	449.92	529.36	1191.08	1307.46
550	1042.80	.18344	2151	.4277	22.43	446.03	541.33	1188.06	1312.18
560	1130.65	0.17960	0.02177	0.3906	24.61	441.61	553.54	1184.59	1316.91
570	1224.08	.17589	2205	.3567	27.00	436.60	566.05	1180.62	1321.65
580	1323.34	.17231	2238	.3257	29.61	430.96	578.91	1176.09	1326.40
590	1428.70	.16887	2275	.2971	32.50	424.63	592.17	1170.91	1331.15
600	1540.42	.16556	2316	.2711	35.68	417.53	605.90	1164.99	1335.91
610	1658.8	0.16239	0.02365	0.2469	39.22	409.55	620.19	1158.23	1340.68
620	1784.1	.15935	2420	.2245	43.17	400.58	635.16	1150.48	1345.46
630	1916.6	.15644	2485	.2037	47.62	390.45	650.94	1141.55	1350.25
640	2056.7	.15366	2561	.1843	52.67	378.93	667.73	1131.21	1355.04
650	2204.6	.15101	2652	.1659	58.46	365.74	685.77	1119.08	1359.84
660	2360.8	0.14849	0.02762	0.1484	65.21	350.40	705.48	1104.51	1364.65
670	2525.4	.14610	2901	.1315	73.27	332.21	727.45	1087.18	1369.47
680	2699.1	.14383	.3088	.1149	83.25	309.89	752.76	1065.18	1374.30
690	2882.0	.14168	3345	.0975	96.40	280.95	783.67	1035.65	1379.14
700	3074.6	.13965	3796	.0760	116.7	236.5	827.3	989.3	1383.98
706.3	3201.0	....	0.04843	0.0484	155.	155.	900.	900.	....

represent Equation [39], full-line curves with even numbers are from Equation [40]. Final curves 5 of h_v and 6 of h_s, which ought to agree exactly, have really the small difference shown magnified along DD and not exceeding 0.5 B.t.u. except at the top of the diagram.

48 With the converted pv quantities thus transferred to the h-p plane, it is a characteristic fact that h' is considerably larger than h_o, while cX or $(cY + cZ)$ is decidedly larger than X' or $(k_1Y + k_2Z)$. Conceiving two state points as moving to the left on line BC, one from point B to generate cX, the other from B′ (at a slower but definitely related rate) to generate X', there is evidently one place where the two points will coincide and make $h_v = h_s$. When this result has been attained within narrow limits and throughout a wide range of pressure, then in so far as this determinant is effective the solution is correct.

49 In the high range above 450 deg. saturation temperature, which is about 70 per cent of the total height of Fig. 20, the differ-

TABLE 5 PROPERTIES OF SATURATED STEAM AT EVERY 10 DEG. CENT.
Metric-Centigrade Units

t	p	c	w	v_s	pw	pv_s	q	h_s	h_v
0	0.00624	19.806	0.001000	205.87	0.00001	1.2838	0.00	596.23	596.28
10	.01252	18.943	1000	106.31	01	1.3305	10.05	600.82	600.90
20	.02382	18.140	1001	57.82	02	1.3770	20.04	605.37	605.51
30	.04322	17.390	1004	32.93	04	1.4232	30.01	609.88	610.10
40	.07514	16.690	1008	19.55	08	1.4691	39.97	614.33	614.68
50	.12568	16.036	1012	12.05	13	1.5144	49.94	618.74	619.26
60	0.2030	15.423	0.001017	7.681	0.00021	1.5592	59.92	623.08	623.84
70	0.3176	14.848	1023	5.048	033	1.6032	69.91	627.33	628.41
80	0.4827	14.308	1029	3.411	050	1.6463	79.91	631.49	632.97
90	0.7148	13.802	1036	2.362	074	1.6884	89.94	635.53	637.53
100	1.0334	13.326	1043	1.666	108	1.7292	100.00	639.44	642.10
110	1.4612	12.879	0.001051	1.2104	0.00154	1.7686	110.09	643.21	646.67
120	2.0252	12.458	1060	0.8917	215	1.8060	120.22	646.80	651.23
130	2.7555	12.062	1039	.6685	294	1.8421	130.39	650.21	655.80
140	3.6862	11.691	1079	.5089	398	1.8758	140.60	653.39	660.38
150	4.8549	11.341	1090	.3929	530	1.9072	150.86	656.36	664.96
160	6.303	11.012	0.001102	0.30717	0.00695	1.9361	161.18	659.07	669.55
170	8.076	10.703	1115	.24297	0897	1.9611	171.57	661.49	674.14
180	10.221	10.412	1129	.19420	1154	1.9850	182.01	663.60	678.74
190	12.791	10.139	1144	.15671	1463	2.0044	192.53	665.39	683.35
200	15.840	9.883	1160	.12754	1837	2.0202	203.13	666.81	687.97
210	19.429	9.643	0.001178	0.10459	0.02288	2.0321	213.80	667.85	692.60
220	23.618	9.418	1196	.08636	2825	2.0397	224.55	668.49	697.24
230	28.473	9.208	1216	.07175	3462	2.0429	235.39	668.68	701.90
240	34.064	9.011	1236	.05856	4209	2.0412	246.33	668.40	706.56
250	40.462	8.827	1258	.05028	5089	2.0345	257.38	667.63	711.23
260	47.747	8.656	0.001280	0.04236	0.0611	2.0225	268.55	666.37	715.91
270	55.995	8.497	1302	.03581	.0729	2.0049	279.87	664.63	720.61
280	65.298	8.350	1325	.03034	.0865	1.9811	291.46	662.32	725.32
290	75.742	8.213	1350	.02571	.1023	1.9471	303.43	659.30	730.04
300	87.424	8.087	1381	.02187	.1207	1.9115	315.88	655.42	734.78
310	100.45	7.971	0.001422	0.018555	0.1428	1.8635	328.98	650.51	739.53
320	114.93	7.864	1470	.015731	.1689	1.8079	344.33	644.26	744.29
330	131.04	7.766	1534	.013243	.2010	1.7353	358.08	636.28	749.07
340	148.69	7.677	1620	.011060	.2408	1.6446	374.88	625.83	753.87
350	168.24	7.597	1740	.009071	.2928	1.5261	393.26	611.92	758.68
360	189.76	7.524	0.001923	0.007168	0.3650	1.3602	418.20	591.77	763.50
370	213.41	7.459	2318	.005107	.4947	1.0899	453.70	556.15	768.34
374.6	225.06	......	3023	.003023	.6803	0.6803	500.	500.	......

ence $(h_v - h_s)$ is very sensitive to the kind of adjustment or matching of curves that is so fully shown in Figs. 8 and 9. Here the full-

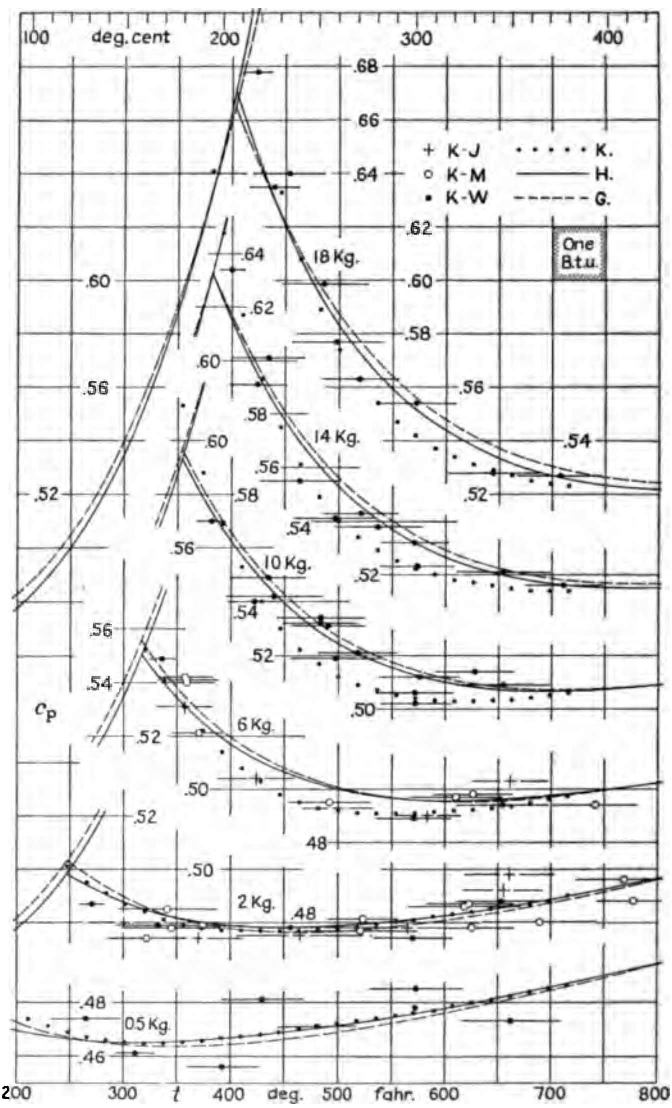


FIG. 16 SPECIFIC-HEAT COMPARISONS

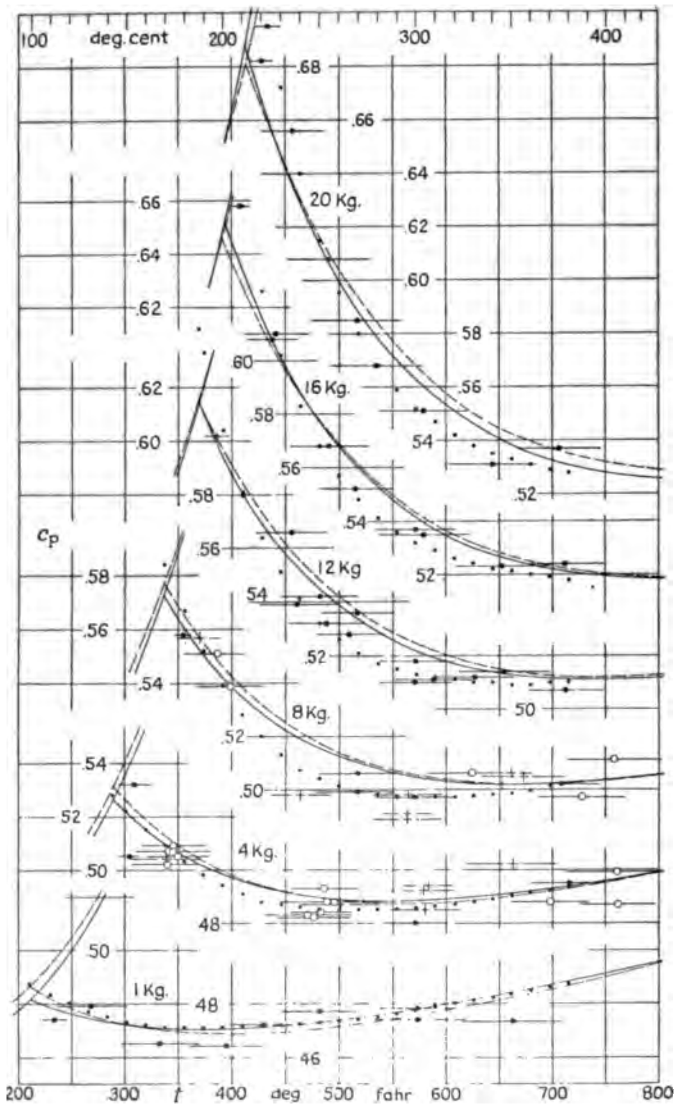


Fig. 17 Continuation of Fig. 16

line curve along DD represents the effect of Equations [26a] and [26c], while the dotted curve results from earlier formulas of the writer for pw and q — most of the change being in q. The discrepancy beginning above 550 deg. is what appears in Fig. 10, where the dotted curve shows h_v against h_s in full line.

50 On the interesting question as to how nearly Clapeyron's law is in itself a full determinant, consider the exhibit in Fig. 21. Using temperature instead of pressure as base, saturation curves of pv (at AA) and of h (at BB) are drawn for three solutions which check to the degree shown by the curves at DD — this with the pw and q values corresponding to the dotted DD curve in Fig. 20

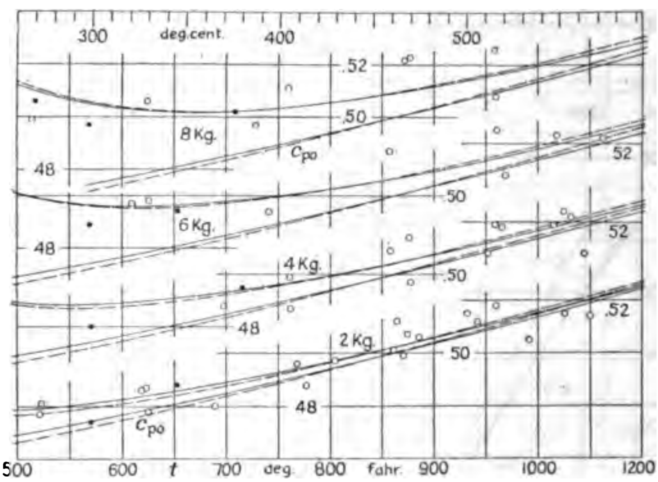


FIG. 18 HIGH-TEMPERATURE EXTENSION OF THE c_p COMPARISON

With the final (H) solution (a) are compared the two fairly good trials (b) and (c) that ran respectively smallest and largest over the range above 400 deg. Of these (b) contains the covolume terms pw_o and Apw_o — see Pars. 8 and 36 — while (c) is like (a) in omitting them. Since cpw_o is much larger than Apw_o, point B in Fig. 20 is shifted to the right much more than B′, requiring larger values of X and X' and thus tending to shift final curve 5–6 toward the left. Note that trials (b) and (c) both run large below 400 deg., especially in h_s.

51 It is evident that this check is by no means a complete determinant, and that formulas which satisfy Clapeyron's law are not therefore necessarily correct. But when a solution conforms

well to all the experimental data — which (*b*) and (*c*) of Fig. 21 do not — and also keeps ($h_p - h_s$) very small, there is strong probability of its close approximation to absolute correctness.

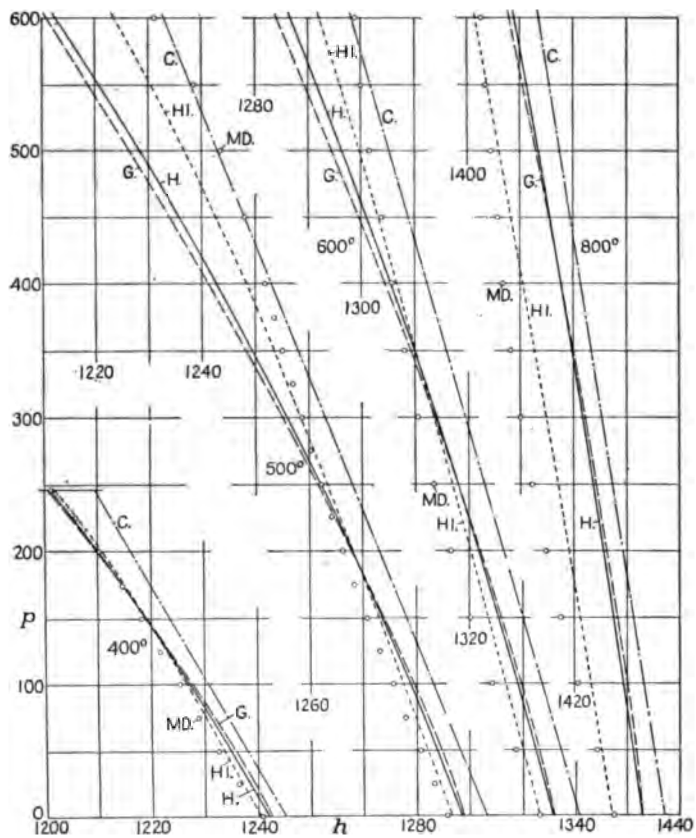


Fig. 19 Comparison of Heat-Content Isotherms

THROTTLING RELATIONS

52 By a throttling operation is meant one in which pressure and temperature vary while heat content remains unchanged. The relative rate of variation $(dt/dp)_h$, the slope of an experimental curve of t on p, is called the Joule-Thomson "coefficient" M. According to Equation [18] this coefficient is the quotient of the (negative) slope of the heat isotherm N or $(dh/dp)_t$, by the specific heat

c_p or $(dh/dt)_p$. From Equation [28], with Equation [8], the general expression for heat slope is

$$N = Aw_o - y' - sz'p^{s-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [41]$$

so that it can readily be computed, then used with c_p from Equation [37].

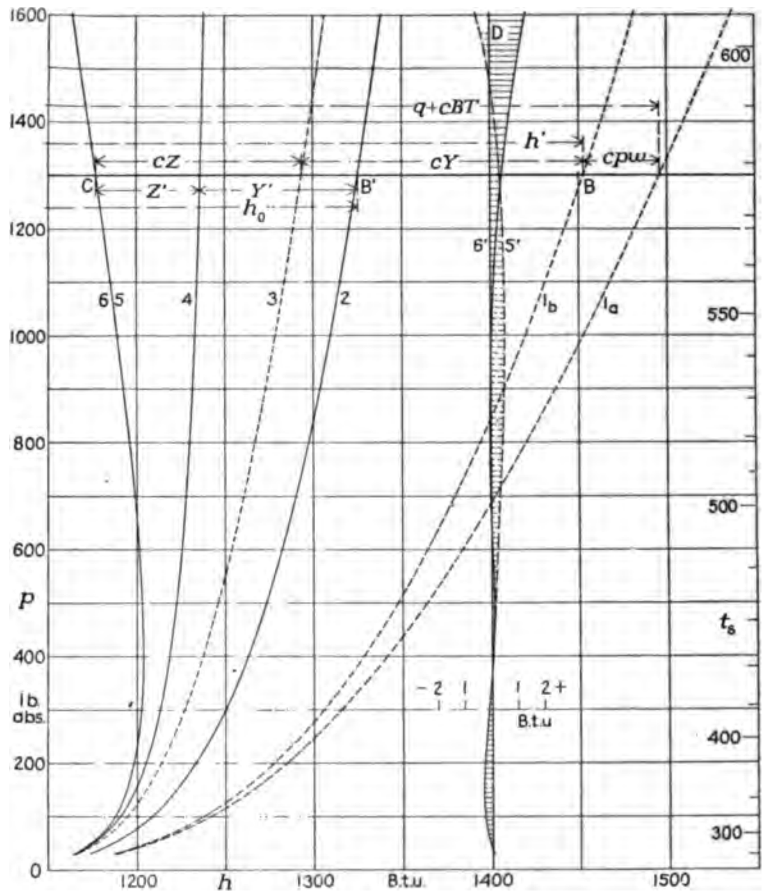


FIG. 20 OPERATION OF CLAPEYRON'S LAW

53 A considerable body of experimental information, from other gases as well as steam, indicates that M is nearly or quite independent of p and h, being a function of T alone. While not fully established and probably not holding rigorously near saturation or at very high densities, this hypothesis is one to which an acceptable formula for heat content ought to conform at least fairly well.

54 Special interest attaches to the particular, simple case of zero pressure, where the last term of Equation [41] disappears, so that we have (with or without the covolume term)

$$N_o = -(y' - Aw_o) \quad \text{or} \quad N_o = -y' \quad . \quad . \quad . \quad . \quad [42]$$

and

$$M_o = -N_o \div c_{po}$$

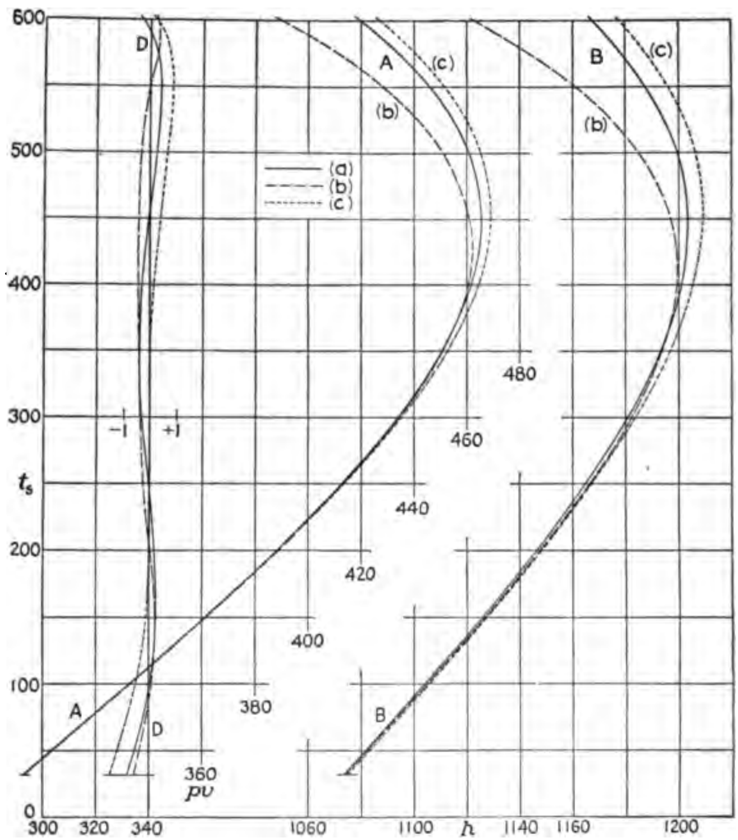


FIG. 21 COMPARISON OF TRIAL SOLUTIONS

as exemplified at the end of Table 3. Curve group (a) of Fig. 22 shows M_o as thus derived from the (H), (G) and (C) equations; and with them is drawn the empirical curve (D) of Davis for M, replotted from Fig. 6 of his paper on The Law of Corresponding States, 1908. It is from throttling data as thus expressed and used that a preliminary indication for function y' is obtained.

55 How the several heat formulas make M vary with p is shown

by group (*b*) of Fig. 22, where curves are drawn for pressures in the geometrical progression 25, 50, 100, 200, 400, and 800 lb., with a line SS of M_o at saturation. The scheme is to take M_o from equation (H) for reference quantity, appearing as base AA, then express all other values by their ratio to this M_o. For the (H) formula saturation curve SS at first rises in reasonable fashion, then above 300 deg. begins to drop and at 500 deg. falls below M_o — this

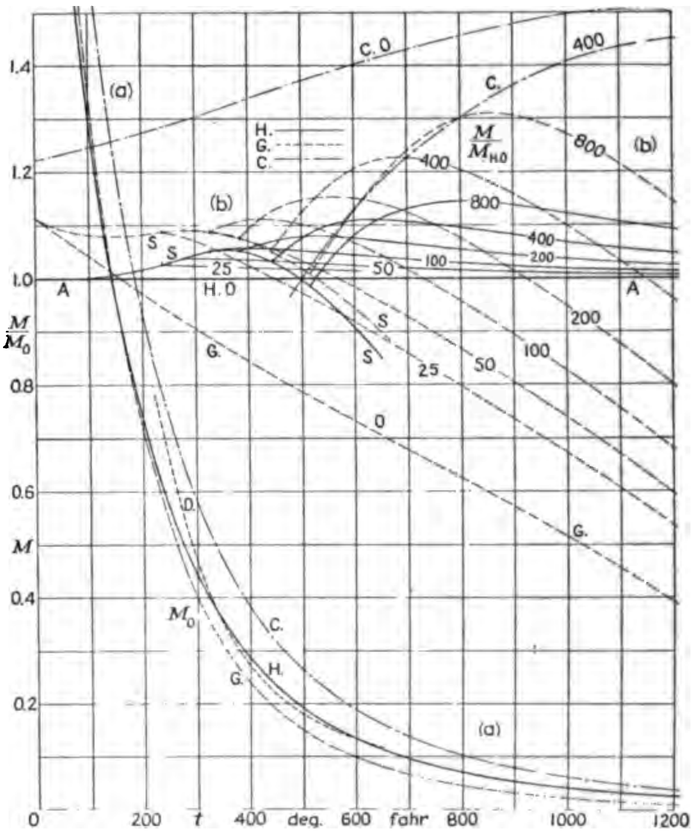


FIG. 22 COMPARATIVE CURVES OF THROTTLING COEFFICIENT M

indicating that c_p rises somewhat too rapidly near saturation at high pressures. The constant-pressure curves run above M_o in moderate degree, to a maximum of 11 per cent for 400 lb. and over 14 per cent for 800 lb.

56 With the (G) formula M_o drops too rapidly as temperature rises, because the value 4 of exponent m in term Y' of heat equation [8] or [28] is too large. The variation of M with p is of the same

general character as in the (H) system, but very much larger: at 1200 deg. $M_{800} = 2.82M_o$, or there is a relative rise of 182 per cent. The (C) equation keeps N constant while c_p rises with p, consequently the M curves (of which only that at 400 lb. is drawn) necessarily lie below M_o; this is not rational, for since M represents a departure from ideal gas behavior, it ought rather to increase with density.

FORM OF THE EQUATIONS

57 In the overall comparison of results, at saturation in Figs. 11 and 12, for superheat in Figs. 13 and 19, very good agreement has been shown between the (G) and (H) formulas. In the mind of the reader will arise the question, Why not consider the investigation an endorsement of Goodenough and let it go at that? Certainly the (G) tables, from the viewpoint of the writer, are superior to any others that have preceded them, as appears especially in Fig. 19; but the closer analysis reveals several points in which they are seriously at fault. The first of these, the excessive rise of c_{po} and c_{ps} toward freezing point in Fig. 14, is a minor matter; but others are of real importance.

58 Figs. 3 and 4, with Table 3, show that the (G) value of curvature term Z or Z' in Equations [7] and [8] is relatively much greater than the (H) value. Goodenough gives insufficient slant to the straight lines AD, A′D′ of Figs. 1 and 2, then compensates by making the curvature term too large. Results appear in the excessive low-pressure curvature of isotherms, Figs. 13 and 19, and in the very bad variation of M, Fig. 22. By using a much larger exponent n with moderately larger s — incidentally getting a better value of ratio k_2 — the writer has secured isotherms that for moderate pressures straighten out rapidly with rise of temperature, but curve over strongly toward saturation at high pressures.

59 In evaluating his equations Goodenough has followed the peculiar practice — undeniably an error — of using in the covolume terms not constant w_o but variable w; so that the example in Table 3 does not conform exactly to his method. To show results at saturation, the following values of h_s have been computed from the (G) equations and compared with the table:

t	500	520	540	560	580	600
With Apw	1197.85	1194.07	1189.03	1182.65	1174.92	1165.73
With Apw_o	1197.28	1193.31	1188.04	1181.35	1173.22	1163.56
(G) Table	1197.9	1194.1	1189.0	1182.7	1174.6	1164.2

Evidently there begins at 580 deg. a curvature over toward critical point K of the kind shown in Fig. 9. Now w is primarily a function of t; but when used after this manner in the region of superheat it must be taken as dependent upon p. In further illustration, computation of the isothermal at 600 deg. gives for h,

p	50	100	200	400	800	1540.4
With Apw	1333.20	1329.83	1322.20	1304.47	1261.94	1165.78
With Apw_0	1333.19	1329.80	1322.11	1304.21	1261.21	1163.56
(G) Table	1333 2	1329.8	1322.2	1304.5	1261.9	1164.2

60 The difference between pw and pw_0 is negligible below 200 lb., but for high pressures it has a serious effect, especially upon the Clapeyron check. Thus at 600 deg. $(pw - pw_0) = 11.43$ for saturation; with factor $c = 1.464$ this would change r_v or h_s by 16.75 B.t.u., while the direct value of $A(pw - pw_0)$ is only 2.17; and there would thus be a discrepancy of 14.58 B.t.u. Goodenough's tabular values show an excellent Clapeyron check throughout, but the correct use of the covolume term will disturb this badly, requiring a considerable decrease in coefficients E and F in order to restore the balance — on trial a decrease of 7 per cent in F alone gave a fair check up to 550 deg.

61 As regards the use or omission of the covolume term, the argument may be stated as follows:

(*a*) This term is not essential to a rational theory of the subject. In Par. 8 its introduction is credited to reasoning upon conditions near the critical state. From Figs. 8 to 10 can be drawn the conclusion that equations of our general form will not determine the saturation curve more than halfway up to the critical *pressure*. Over this effective range there is no reason why the influence of molecular volume cannot be conceived as a part of whatever physical condition is represented by contractive terms X and X'.

(*b*) The practical effect of using the covolume term is to make smaller the values of pv_s and h_s that will satisfy Clapeyron's relation at high pressures — see Par. 50. Thus in Fig. 21 a solution as good as (*a*) would swing to about the same distance from (*b*) that (*a*) now is from (*c*), with a strong tendency toward the extra swell below 400 deg. that is contrary to the data in Fig. 12.

(*c*) The close agreement of the (H) with the (G) high saturation lines is fortuitous in two respects: first, the location of the (G) curves is obtained by an incorrect use of thermodynamic principle; second, the writer did not aim at this agreement, but came to it as an entirely independent conclusion. Such predilection as he entertained was for the h_s curve of his 1912 formula (H2), running with Marks and Davis to a maximum of 1210 B.t.u. around 450 deg. and established by full use of the throttling data and method. It has not been found possible to reconcile that indication with other information of equal or greater weight; but there does not seem to be good reason for going yet farther from it in order to meet a theoretical requirement of doubtful validity.

DATA YET NEEDED

62 Reviewing statement (*c*) of Par. 14 in connection with the various comparisons that have been made, it is clear that for positive confirmation of any scheme of formulation there is need of some dependable and accurate steam data in the region of very high pressures. To isolate energy quantities for precise measurement under such pressure conditions would be almost impossible, so that there is little hope of good heat data. On the side of volume the physical quantities are much simpler and easier to handle; and under the scheme of related formulas there is diminished need for data in both fields. A comparatively small amount of first-class physical experimentation, following preferably the constant-volume plan of Linde and using methods such as Battelli employed thirty years ago — but without bothering about saturation — would be sufficient to resolve any present uncertainty. Pending such information, and with the confident expectation that it can call for but little readjustment, the writer offers his new formulas as the best that can now be done toward establishing a standard steam table.

APPENDIX

63 The following list contains all the symbols that have particular meaning, except a few of strictly local use, with reference to the place where they are more fully defined.

		Par.
A =	heat equivalent of work	4
B =	gas constant in p-v-T equation	4
c =	specific heat, used only with distinguishing subscripts.	
c =	Clapeyron ratio r/pu or r/Apu	12
E, F =	coefficients in general equations	11
h =	heat content or total heat	2
J =	mechanical equivalent of heat	4

		Par.
k	= ratio in general equations	11
M	= Joule-Thomson coefficient, throttling relation	52
N	= slope of heat-pressure isotherm	52
m, n	= exponents of T in general equations	11
p	= pressure, absolute	2
q	= heat of liquid	2
r	= heat or vaporization or latent heat	2
s	= exponent of p in general equations	7
T, t	= temperature, absolute and thermometric	2
u	= volume increase during vaporization	2
v	= specific volume, in general	2
w	= specific volume of water	2
X	= total contractive term, general equations	7
Y	= contractive term with p	7
Z	= contractive term with p^s	7
y, z	= temperature functions	11

Generally used subscripts, with their meanings, are as follows: c, at critical temperature or pressure; h, for constant heat content; o, at zero pressure; p, at constant pressure; po, at constant zero pressure; s, at saturation; t, with temperature constant; v, derived from volume by Clapeyron's ratio.

64 Useful ratios of equivalence of units are:

Quantity	English-Fahrenheit	Metric-Centigrade	Logarithm
Surface	1 sq. in.	6.452 sq. cm.	0.80967
	1550 sq. in.	1 sq. m.	3.19033
Volume	35.314 cu. ft.	1 cu. m.	1.54795
Weight	2.2046 lb.	1 kg.	0.34333
Pressure	1 lb. per sq. ft.	4.8824 kg. per sq. m.	0.68864
	1 lb. per sq. in.	703.07 kg. per sq. m.	2.84700
	14.223 lb. per sq. in.	1 kg. per sq. cm.	1.15300
Atmosphere	14.697 lb. per sq. in.	1.0334 kg. per sq. cm.	
Work	7.233 ft-lb.	1 m-kg.	0.85931
Temperature	1.8 deg. fahr.	1 deg. cent.	0.25527
Heat	3.9683 B.t.u.	1 cal.	0.59860

65 Between numerical values of important compound quantities in the two systems the relations hold,

	(EF) (MC)	Log	
Specific volume	$v = 16.019\ v$	1.20462	[43]
Pressure × volume	$pv = 227.84\ pv$	2.35762	[44]
Gas constant	$B = 126.58\ B$	2.10235	[45]
Mechanical equivalent	$J = 1.8227\ J$	0.26071	[46]
Throttling coefficient	$M = 0.1266\ M$	$\bar{1}.10227$	[47]

66 In order to establish Equation [10], Par. 10, take the total differential of h as a function of t and p, in the form

$$dh = c_p dt - A\left[T\left(\frac{dv}{dt}\right)_p - v\right]dp \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [48]$$

and by imposing the condition t = constant get the partial derivative,

$$\left(\frac{dh}{dp}\right)_t = -A\left[T\left(\frac{dv}{dt}\right)_p - v\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [49]$$

From the pv equation, using the more general form of Equation [27],

$$v = \frac{BT}{p} + w_o - \frac{E}{T^m} - \frac{F}{T^n}\,p^{s-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [50]$$

Differentiating with respect to t, holding p constant,

$$\left(\frac{dv}{dt}\right)_p = \frac{B}{p} + \frac{mE}{T^{m+1}} + \frac{nF}{T^{n+1}}\,p^{s-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [51]$$

Substituting Equations [50] and [51] in Equation [49],

$$\left(\frac{dh}{dp}\right)_t = \left[\frac{BT}{p} + \frac{mE}{T^m} + \frac{nF}{T^n}\,p^{s-1} - \frac{BT}{p} - w_o + \frac{E}{T^m} + \frac{F}{T^n}\,p^{s-1}\right]$$

$$= Aw_o - A(m+1)\,\frac{E}{T^m} - A(n+1)\,\frac{F}{T^n}\,p^{s-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [52]$$

Integrating with respect to p along the isothermal between the limits 0 and p,

$$h = h_o + Apw_o - A(m+1)\frac{E}{T^m}\,p - A\frac{(n+1)}{s}\,\frac{F}{T^n}\,p^s \quad . \quad . \quad . \quad . \quad [53]$$

This fixes ratios k_1 and k_2 as given in Equations [13] and [14].

67 Results got by evaluating the (H) Formulas, Equations [27d] and [28d] for saturation, with the high-range equations [26a–d] as shown by Figs. 8 and 9, are given in Tables 4 and 5, at ten-degree intervals on both temperature scales. All units are as defined in Par. 2; but note that c is r/pu in Table 4, r/Apu in Table 5 — compare Par. 12. These are intended for easy comparison with other formulations, would serve as bases for service tables, and are expressed to the full degree of accuracy of five-place logarithms; but the second table is not quite sure in the last figure of its fuller numbers, because it is obtained by interpolation and conversion from the first, not by direct, independent computation

DISCUSSION

D. S. Jacobus. There is a need of a steam table which will give correct values for the higher pressures, and it would seem that values based on the author's formulas are as good as can be secured with the present available data. It would be a good plan for the author to outline the data which will be of the greatest use in arriving at exact values for the high pressure range in order that future investigators may be guided thereby and not waste effort on experiments not as directly applicable as others.

Edgar Buckingham. The author should be commended for having gone through the tremendous labor of making the comparisons of data and formulas as represented in his paper and for having himself contributed new formulas. This, however, is not the time to adopt a new table. Any one of the present steam tables is good enough for most purposes if used consistently and to get any real improvement in accuracy, new experimental data are absolutely indispensible. With the present information, the existing steam tables can not be improved upon, their discrepancies being due to the necessity of guessing at values which can be known only from experiments not yet performed. Any further reworking of the present data alone would be a waste of time.

William D. Ennis. The differences in existing steam tables are occasionally of practical importance and it is a matter of prime importance that correct values for the higher range region should be agreed upon and generally accepted. This situation suggests that we have before us really two problems.

The first is, What new experimental work must be done, if any, to eliminate existing differences? This decided, the second may be attacked, namely, the obtaining of new experimental data and the preparation of uniform tables based thereon.

Charles E. Lucke. While it is likely that the results submitted for the high pressure ranges will prove helpful, they cannot be regarded as conclusive, nor can the rest of the data be regarded as much more than an interesting mathematical manipulation of old facts. Nothing has been presented that would justify the abandonment of existing tables now in practical use for any of the problems of daily practical work, though attention has again been called to the desirability of a complete review of the entire subject by adequate experimental equipment in the hands of the most competent physicist available. New facts would be immensely useful; new formulations of old facts are just as likely to be harmful and befog the issue as to be helpful.

Charles W. Berry. The analysis of the physical data and the comparison of the equations for steam made by the author serve but to emphasize more strongly the need of more data in the regions already investigated and the extension of experimental investigation into the higher pressure regions both for saturated and superheated steam.

With the steam tables of Peabody and of Marks and Davis giving reasonably accurate data in the lower temperature ranges and with the tables of Goodenough agreeing fairly well with the new equations proposed by the author in the upper regions, it seems unnecessary at the present time to calculate new tables unless these are based upon more acurate and extensive data than we now possess.

The most profitable thing to do would be for physicists to undertake such investigations and for the engineering societies to give financial aid either to such private investigators or to the Bureau of Standards.

It seems desirable that more attention should be given to developing more rational equations of state than to spend any more time endeavoring to adjust coefficients of terms in empirical equations in an endeavor to wind snakelike through clusters of conflicting data.

H. B. EVANS.[1] Accurate knowledge of the behavior of steam under varying conditions is at present confined to relatively small limits. Present practice has actually gone beyond these limits. An experimental campaign sufficiently comprehensive to provide an equally reliable extension which will cover completely all present and prospective demands seems beyond the powers of any single private or college organization.

Our only practicable means for further extension now lie in the standard method of incorporating from time to time with our previous knowledge reliable new data. Each new step should bring us either to a more reliable set of constants or to a formulation of empiric terms which have a greater probability of theoretic correctness. The present paper seems to do both and seems to be a step in advance.

Recent developments in molecular and atomic theory seem to promise eventually a theory from which steam and similar formulas may be deduced with all mathematical correctness. There seems to be no reason to hope for it, however, in the near future, and in the meantime empiric methods must continue.

Our present knowledge is not yet sufficient to differentiate

[1] Univ. of Penn., Philadelphia, Pa.

between possible empiric terms with satisfactory accuracy, but the present paper embodies a very definite amount of progress and brings into usable form the results of the most recent data.

L. C. LOEWENSTEIN. The author has performed a real service in placing together for comparison the available data upon which steam formulas can be based. His work also shows the possible discrepancies in existing steam tables. Without further experimental data no one can say that the values given by the author which he derives from curves drawn in a rational manner to conform to existing data are any less accurate or any better than those given in the more recent steam tables published.

The need for further experiments is of importance because we have almost reached a stage where our research lags behind the demands of practice. Our power stations and our Navy are going to temperatures and pressures which require more data throughout the upper range so that designers of steam prime movers can more closely check the results obtained from their designs.

SANFORD A. MOSS. The author has given much new and valuable information as to the fundamental requirements which steam formulas must satisfy and has also well shown the degree with which this satisfaction is accomplished by the various recognized steam tables and formulas. However, there arises a question as to whether or not the author's purely empirical formulas satisfy the conditions he sets down with enough greater accuracy to warrant their acceptance at this time in preference to previous values. The experimental data which he has collected show such divergence as to make it questionable as to whether or not his equations are any nearer the temporarily unknown truth than any of the other equations.

Steam formulas should satisfy some or all of three conditions.

1 They should agree with complete and exact experimental data. We lack such at present so that the author obviously cannot fulfill it.

2 There should be some rationality. The author's general forms are shown to be rational, but the details are wholly arranged to satisfy empirically the very divergent experimental data. These experiments are so divergent that mere agreement of formulas with them, without complate rationality, does not carry conviction.

3 The formulas should be mathematically rigid. There are two conditions to satisfy, as the author points out, the thermodynamic relation between total heat and volume, and Clapeyron's equation. The author gives rigid adherence to the first test, but with Callendar, Goodenough and all of the others, has an independent saturation equation which gives satisfaction of the Clapeyron equation merely as a coincidence.

It can be shown[1] that the grandfather of all gas formulas is, an expression for what Gibbs calls "thermodynamic potential." This is given by $H/T - \phi$ where ϕ is entropy. If an expression can be found for this quantity which can be handled mathematically for both liquid and gas, the entire set of values, including the saturation equation, follows by simple mathematics. This cannot be found from the author's values in a form which inspires confidence. The existing experimental and theoretical data do not warrant us in accepting any one authority over the other at the present time.

F. G. Keyes. Knoblauch-Linde-Klebe found that at constant volume the pressure increased linearly with the absolute temperature, and nevertheless, they set up an equation of state for the steam which made the pressure a non-linear function of the temperature when the volume of a unit mass of steam is kept constant. The author does the same thing. The writer has the results that Mailey obtained on steam up to 400 deg. cent. and 7000 lb. pressure, and the isometrics are certainly straight lines.

There is evidently great need of more experimental data in the superheated region both for the p, v, t relation and for the specific heats, although the writer feels confident that with a rational equation of state, the specific heats made can be taken care of since the specific heat of steam at one atmosphere is accurately known.

It seems that it is futile at this time to attempt to improve the steam tables without a very considerable amount of additional data, the writer agrees with the comment of Sanford A. Moss that the equations lack rationality, and the refinements that have been attempted in calculations are not warranted in view of the fact that there are no particularly new data available at present.

Lionel S. Marks. The primary need at the present time is for more complete data as a basis for new formulas. The unsatis-

[1] *Physical Review*, May 1915.

factoriness of the heat content data is shown by Fig. 12, especially by the two Smith points S_1 and S_2 at 212 deg. Smith obtained these divergent results by evaporating water at different rates, the lower heat content being obtained when the rate of evaporation was more rapid, with the result that wet steam was being generated; the higher value was obtained at a slower rate at which the steam was drier than dry steam, according to the author's figure.

It is possible that "dry and saturated" steam is not obtainable experimentally, and the divergence of 8 B.t.u. represented by these points may indicate a zone of "dry and saturated" steam; in order to make the tables absolutely definite, it may be necessary to make a new definition of what is meant by dry steam.

Turning to the superheated region represented in Figs. 16 and 17, the new formulas of the author do not seem to represent the experiments any better than do the Knoblauch points.

Besides need for more data at high pressures, there are discrepancies at the low-pressure end as shown by the break in the pressure-temperature experiments data at 50 deg. cent. in Fig. 6, with resulting doubt as to the exact data for the Clapeyron ratio below 110 deg. fahr.

HARVEY N. DAVIS. There are two empirical tests which the author has not mentioned, namely, the "straight diameter," and Thiesen's plot of log r (latent heat) against log $(t_c - t)$, both of which are nearly straight for most substances.

The writer questions the shape of the curve for CC_{p0} at the low-temperature end, Fig. 14, on the ground that for air, where fairly accurate data are available, there is no minimum. Also, the pressure variation of the Joule-Thomson coefficient, M, Fig. 22, while much better from the (H) than from the (G) formulation, is still probably wrong in sign as well as in magnitude by analogy with the definitely known trend for air, making the "rationality" of either formulation somewhat doubtful.

G. A. GOODENOUGH. Throughout his paper the author has used Goodenough's table of steam properties as a background on which to project his results; hence in this discussion the writer will confine himself principally to the comparison between the (H) and (G) formulations.

Assuming that a table is constructed from the (H) equations, to what extent will the tabular values differ from those of the (G) tables? The writer has compared the two sets of figures, one set taken from the table in the paper, the other from his published tables, and he finds that, for the most part, the differences are less than one part in one thousand; only in a few instances do they reach two in one thousand. The average difference seems to be about six or seven parts in ten thousand. Granting that the (H) formulation is more accurate than the (G) — and it ought to be, as it is so much more complicated — it is a fair question whether it is worth while to construct new tables that will differ so slightly from existing tables. The author himself seems to have this in mind when he says, in Par. 57, "In the mind of the reader will arise the question, Why not consider the investigation an endorsement of Goodenough and let it go at that?" That the reader may not complacently take such a view, the author hastens to point out certain alleged atrocities in the (G) formulation. The writer will attempt to show that these are not as serious as they are represented.

The first matter is the use of the variable w instead of constant w_0 for the covolume term; see Par. 59. That such use is an error, the writer is unable to comprehend. The problem is to get a purely empirical equation between p, v, and t of superheated steam; an equation that satisfies the experimental data. If, under constant pressure, water is vaporized and the vapor is then superheated, the increase of volume during the two operations is $v - w$. If it suits the writer's purpose to take the difference $v - w$ instead of the final volume v there is nothing in the laws of thermodynamics to prevent him from so doing; and he cannot see that he is thereby guilty of "incorrect thermodynamic reasoning." The writer's reason for adopting this device was merely to reduce the labor of computation. Taking the variable volume w instead of the constant w_0, the difference $v_s - w$, which is a term of the Clapeyron relation, is easily computed from the characteristic equation.

The criticism expressed in Par. 60 has, therefore, no point whatever. If the writer had used the constant w_0 as prescribed by the author, he would certainly have reached incorrect results; but using the variable w as he did, the satisfaction of the Clapeyron relation was complete.

The most serious of the faults pointed out by the author is, therefore, not a fault at all. The course pursued in the (G) formulation is quite defensible and it leads to results that are fully confirmed by the (H) analysis.

The other fault is that discussed in Pars. 55 and 56, with the accompanying Fig. 22. The author points out the considerable variation of the values of M, the Joule-Thomson coefficient, deduced from the (G) equations. The criticism might have some weight if (1) we were particularly interested in accurate values of this coefficient and (2) if we had sufficient knowledge of this coefficient to give a sure basis of comparison. The value of M is the slope of the curve, h = constant, when drawn on the p–t plane. Referring now to Fig. 19, it is seen that the (H) and (G) values of the heat content h of superheated steam show remarkable agreement. To show how close the values are, the following figures for temperatures of 500 and 800 deg. fahr. are presented:

AT A TEMPERATURE OF 500 DEG. FAHR.

Pressure	50	100	150	200	250	300	400	500	600
(H)	1283.8	1278.8	1273.7	1268.0	1262.2	1256.0	1242.6	1228.2	1212.7
(G)	1284.7	1279.6	1274.0	1268.1	1261.8	1255.2	1241.3	1226.5	1210.8
Difference per 10,000	+7	+6	+2.4	+0.8	−3.2	−6.4	−10.4	−14	−16

AT A TEMPERATURE OF 800 DEG. FAHR.

(G)	1431.1	1429.4	1427.5	1425.5	1423.6	1421.8	1417.7	1413.4	1409
(H)	1431.6	1429.9	1428.1	1426.2	1424.3	1422.2	1417.7	1413.0	1408
Difference per 10,000	+3.6	+3.6	+4.3	+4.9	+4.9	+2.8	0	−2.8	−7

In Fig. 19, the curves are lines of constant temperature with p and h as coördinates. Evidently, from the close agreement of the h values, curves of constant h with p and t as coördinates would show the same order of coincidence. We have here a case that occurs frequently. Two curves represent certain data with practically equal accuracy, but nevertheless the difference of curvature is such that the values of the first derivatives of the two curves are quite discordant. Thus the difference between the curves is for the most part less than one in one thousand, while the difference between the derivatives, as shown by Fig. 22, may reach 20 or 30 per cent.

As a matter of fact, we have no accurate knowledge of the Joule-Thomson coefficient for steam. Values were computed by Dr. Davis from the available throttling experiments, but these experiments showed discordant results and were evidently vitiated by gross errors; and errors in the throttling curve are enormously magnified in the values of M deduced from them. The coefficient

M may be independent of the pressure as is assumed by the author, but for all we know at present, it may show a considerable variation with pressure. It is, therefore, quite unsafe to make any dogmatic assertion regarding the relative accuracy of two sets of curves shown in Fig. 22. Even granting the author's contention that the (G) values of M are bad, the "serious fault" lies in the M values, which are never tabulated. Certainly the h values, which are practically coincident with those of the (H) formulation, cannot be severely criticized.

The second "serious fault" of the (G) formulation turns out, to be as harmless as the first.

An over-all comparison of the two formulations leads to the following conclusions:

1 For the range covered, 32 deg. to about 560 deg. fahr., the two formulations give almost identical values for all the properties of saturated and superheated steam that are usually tabulated. Above 560 deg. the values run fairly close to about 620 deg.

2 For extrapolation from the highest temperature for which experiments are available to 620 deg. the formulations are equally good.

3 For the range 620 deg. to the critical temperature the results differ considerably, as different methods of extrapolation were employed. The (H) guess is as good as the (G) guess, but no better.

4 As regards simplicity of the equations employed, the (G) formulation has a distinct advantage.

5 In the matter of accuracy there is probably little choice. In some respects the (H) formulation seems to have a slight advantage; in other places the advantage lies with the (G) formulation. Thus there is little doubt that the (G) value of the latent heat at 212 deg. is nearer the truth than the (H) value.

The one outstanding point is that there are two entirely different formulations built up by different methods but each representing quite accurately the experimental data. They lead to practically identical values of the various properties. The evident conclusion is that we are converging rapidly to final results.

H. L. CALLENDAR.[1] It is an ungrateful task to criticize such ambitious efforts as those of the author of the present paper, but his remarks on the writer's work are so misleading that he has no option but to reply. Fortunately the recent publication of the writer's book,[2] which was written before the war, will enable him to curtail his explanations by giving references to the pages discussing the points at issue.

The author states: "Callendar's equations, when published in 1900, were a fairly good guess . . . but do not conform to the new data that have appeared since . . . nor does any attempt seem to have been made to satisfy Clapeyron's relation." He does not appear to realize that the writer's equations were founded on very laborious experiments extending over a period of ten years, and including,

1 A determination of the variation of the specific heat of water from 0 to 100 deg. cent. and an elaborate comparison of the platinum thermometer with the gas thermometer (pp. 11–16)
2 A measurement of the specific heat of steam at atmospheric pressure (p. 45) by the continuous electric method, since commonly employed
3 Observations of the Joule-Thomson cooling effect (p. 54) by a differential method, which has not yet been surpassed
4 An investigation of the adiabatic equation for dry steam (p. 52) and of the phenomena of supersaturation, with a compensated platinum thermometer.

These experimental data, when taken in conjunction with the laws of thermodynamics, sufficed to determine all the required properties of steam, including the saturation pressures, without any guess work. The deduction of the equation of saturation pressure from the entropy (p. 137) which the author criticises as "a rather abstruse theoretical relation between heat and entropy quantities," is precisely equivalent to the integration of Clapeyron's equation (p. 140) and is the only satisfactory method of securing exact consistency between the various properties in the state of saturation, besides affording a severe test of the validity of the theory (p. 142).

[1] Professor, Imperial College of Science and Technology, South Kensington, London, S. W., England.

[2] Properties of Steam. Page numbers appearing in parentheses refer to this book.

The values of the total heat of saturated steam thus obtained in 1900, while disagreeing materially with Regnault's formula, gave good agreement with the observations of Dieterici at 0 deg. cent., of Griffiths at 30 and 40 deg. cent., and of Joly at 100 deg. cent., when reduced to a common unit (pp. 34, 36); but they differed by no less than 16 B.t.u. at 390 deg. fahr., and by 27 at 500 deg. fahr., from Thiesen's formula for the latent heat as originally applied by Thiesen himself in 1902 to the case of steam (p. 112) with 365 deg. cent. for the critical temperature. This very serious discrepancy was removed in 1904 by the discovery of Traube and Teichner (p. 39) that the true value of the critical temperature was 374 deg. cent., which brought Thiesen's formula into agreement with the writer's results by the Joule-Thomson method within 1 B.t.u. all the way from 32 deg. to 500 deg. fahr. To make sure of this remarkable agreement, the writer immediately verified the value of the critical temperature with a platinum thermometer, and made arrangements with Professor Dalby to extend the observations of the cooling effect up to the same limit, which was accomplished, in 1906, with the assistance of the late Professor Ashcroft, and verified the equations previously published.

There can be no doubt at the present time that the formula of Thiesen for the latent heat, which has been verified for a great variety of substances in the critical region, such as CO_2 (p. 158) affords the best available indication of the "correct trend and course of the values of the total heat toward the critical point," a requirement which, according to the author, the writer's equations "completely fail to meet." The author is judiciously silent about the Thiesen formula, but the fact that his tables and equations give values of H_s about 27 B.t.u. lower than the writer's at 500 deg. fahr., does not necessarily prove that the latter's are incorrect at this point, although the writer makes no claim to represent the critical state, which cannot be done without intolerable complexity, and is not required by the steam engineer.

The formula which the writer adopted in 1902 as representing the total heat of the liquid, h, under saturation pressure, p, namely

$$h = st + avT(dp/dt) \dots\dots\dots\dots\dots\dots [1]$$

was chosen primarily because it agreed with the results of the experiments by the continuous electric method, and because it introduced the greatest possible simplification into all the relations for saturated steam. Thus the general equation of saturation pressure, deduced from Claperyon's relation, reduces immediately to the very simple form,

$$\int L'dt/T^2 = \int a(V/T)dP, \text{ (at constant } T) \ldots\ldots\ldots [2]$$

in which L' is the ideal value of the latent heat (p. 138), and the integrals are taken between any pair of corresponding limits of p and t, on the saturation curve. If P is given as a function of V and T, as in Van der Waals' equation, it suffices to replace the integral on the right by its equivalent $(aPV/T) - \int a(P/T)dV$, taken between limits as before. An example of this case is given for CO_2 (p. 185), where it is shown to give far better values than can be obtained from Maxwell's theorem. This agreement verifies the suitability of Formula [1] for use up to the critical point. The formula is also confirmed in the case of water by Dieterici's measurements in 1905 of the intrinsic energy, when corrected (p. 28) to total heat by adding *apv*, which the author and others neglect. The single Formula [1], owing to its extreme simplicity, is much to be preferred to the three arbitrary and ill-fitting formulas employed by the author, and gives a more natural approach to the critical point.

The variation of the specific heat of steam with pressure, predicted by the experiments on the cooling effect, was roughly confirmed by the experiments of Knoblauch and Jakob in 1906 extending to 8 atmospheres. But their extrapolation to higher pressures was clearly impossible, and was conclusively disproved by the experiments of Prof. Carl Thomas in 1907, by greatly improved methods, extending to 34 atmospheres. The author omits to mention these experiments, which confirmed the variation required by the cooling effect (p. 106) as closely as could be desired up to 500 lb. pressure. The increase with temperature of the specific heat at zero pressure, amounting to 14 per cent between 100 and 400 deg. cent. according to Knoblauch and Jakob, was reduced by Knoblauch and Mollier in 1911 to 4 per cent. Goodenough, from the same observations, deduces a *diminution* of 1 per cent over the same range, and the author an increase of 2 per cent, each with a minimum value about 0.456 (which is probably too low) somewhere between 150 and 300 deg. cent. This variation is evidently much too small and uncertain to be worth considering in any equations for steam engine work. The latest experiments of Knoblauch and Winkhaus in 1915, extending to 20 atmospheres, entirely fail to support the extrapolations of Davis and Jakob (p. 113), though still affected by the old systematic errors which the equations of the author and Goodenough were specially constructed to represent.

The author and Goodenough lay great stress on the curvature of the isothermals on the PV–P diagram representing the Munich observations of 1905 on the volume of steam, which they endeavor to reproduce by equations of the same type as that employed by Linde. It is well known that an equation of this type cannot represent the approach to the critical state satisfactorily. Linde's equation leads to a maximum value for the total heat of saturated steam in the neighborhood of 400 deg. fahr., and gives a value no less than 47 B.t.u. lower than Thiesen's formula at 500 deg. fahr., both of which results are quite impossible. It is easy to show theoretically that the error of the rectilinear isothermal should not exceed 1 in 1000 at 180 deg. cent. all the way from zero to saturation pressure, and could not have been detected at all in the Munich experiments, because they did not give the isothermal directly, but only the lines of constant volume (p. 92) which the observers assumed to be straight. The excessive curvature represented by Linde's equation, which leads to such impossible results when extrapolated, must be attributed to experimental errors, e.g. surface condensation (p. 93), which are inevitable in such difficult experiments. Apart from these errors, which are readily discounted, the Munich experiments on the volume verified the values predicted by the differential throttling method and the adiabatic equation as closely as could be desired.

It is important for practical purposes that the values of the saturation pressure p should be exactly consistent with the equation assumed for V, which is readily secured by the method indicated in Formula [2]. But if this condition is satisfied, an error of 10 lb. in p at 500 deg. fahr. makes only a negligible error in the value of H at saturation, and less than one in one thousand in H or PV for superheated steam. Owing to the phenomenon of supersaturation and the continuity of the adiabatic on the H-log$P$ diagram, the exact position of the saturation line makes very little difference in the adiabatic heat drop, provided that the values of H are right. There is no object in employing a complicated empirical formula for the saturation pressure, such as that of Goodenough, which the author considers "should be accepted as final." The observations on which this formula is founded show obvious discontinuities at 50 and 100 deg. cent., where the method of experiment was changed. There is a similar peculiarity at 200 deg. cent., and the results at higher temperatures are probably vitiated by the presence of gas. Holborn and Baumann were working in the dark by a static method with apparatus designed for 5000 lb. per sq. in. and unsuitable for 250 lb.

Any systematic errors in p are many times magnified in the values of dp/dt, and are further exaggerated by extrapolating an impossible type of formula for V. It is not surprising that the author's values for the latent heat and total heat, deduced by such a tortuous process, should disagree materially with the Thiesen formula and with the variation of the total heat as directly deduced from the cooling effect. The accuracy obtainable depends chiefly on the measurement of temperature, to which sufficient attention has seldom been given, especially when mercury thermometers were employed.

It is most essential for practical purposes that the equations should be as simple as possible and exactly consistent with the laws of thermodynamics. This cannot be secured satisfactorily if empirical formulas are mixed up with rational equations. To be of any use, the tables must agree precisely with the expressions employed for the adiabatic heat drop and for the discharge through a nozzle. These points were carefully considered in the light of practical tests before the writer's equations were published, but they did not seem to appeal to the pure theorist. With such limitations it is impossible to include the critical state, but ample accuracy can be secured for the ordinary range of steam engine practice. The tables and equations are required primarily as a standard of reference which ought not to be subject to capricious alteration in unimportant details without cogent reason. The explanation of the fact that the writer's equations have been republished in their original form, which the author finds so hard to understand, is simply that they have been so closely verified in all essential respects by subsequent work that no alteration was required. It is not, after all, so very remarkable that the original experimental data should have been so nearly correct, if one considers the amount of time and labor expended on the work, and the facility acquired in the construction and use of so many varieties of platinum thermometer, on which the accuracy of the results mainly depend.

H. M. Martin.[1] Before proceeding to any criticism of the author's own work, it is desirable at the very outset to correct one most serious error under which he labors. In Par. 17 he suggests that Callendar's steam tables do not satisfy Clapeyron's equation. As a matter of actual fact, the saturation volume as deduced from Callendar's equation of state is in absolutely perfect accord with his values for the latent heat and for $(dp/dt)_{sat}$. The misapprehension cited seems to indicate that engineers on your side of the Atlantic

[1] 26 Addiscombe Rd., Croyden, London, W.C.2, England.

have been as slow as our own to realize the true scope and bearing of Callendar's work. This view is confirmed by the circumstance that although Callendar had pointed out in 1900 that equations purporting to represent properties of a vapor must be mutually consistent, this principle was totally ignored in the elaborate and very convenient tables compiled some years later by Peabody and by Marks and Davis. All these authors made use of empirical German formulas, based on inexact German experiments. The formulas in question are mutually incompatible and if we deduce the specific heat at constant pressure from the formula for the total heat of superheated steam, we get results differing as much as 20 per cent from the value deduced from the empirical equation of state. Moreover, the work done in the adiabatic expansion of steam as deduced from the pressure and volume differs from that deduced from the total heat and entropy.

The writer discovered this in 1912 when attempting to reduce by means of these tables some exceptionally careful and complete tests on a steam turbine. The defect in question led to the waste of work extending over several weeks, and since this experience he has entirely abandoned the use of these tables in spite of their very great convenience, in which regard they excel all others.

The author's object is by no means clear. Apparently it has been to devise some sort of equation which shall represent the average of all the experimental data between 0 deg. cent. and the critical temperature. Experiments, however, should be weighed as well as tabulated, otherwise there is a danger of repeating the blunder of Encke who, by a most laborious application of the method of least squares to all available data arrived at a very erroneous value for the solar parallax. Experiments on steam at high pressures and temperatures are extremely difficult, and whilst these difficulties could no doubt be overcome by experimental geniuses such as, say, Professor Millikan or Professor Michelson, there is every reason to believe that the observations of the erudite mediocrities at Munich and Berlin suffer from systematic errors, and the consistency of their results merely means that they are consistently wrong.

In reducing observations which may be inaccurate it is desirable to bear in mind the actual mechanism at work, a procedure which will often yield valuable hints as to what form the equation should take.

Probably no single concept in physics is today more securely based than the kinetic theory of gases and vapors. As the writer has observed elsewhere,[1] we ought to distinguish between the perfect gas and the ideal gas. The perfect gas may be regarded either as a continuum or as composed of Boscovich points having mass but devoid of volume, whilst the ideal gas differs from actual gases in that the collisions between its molecules are perfectly elastic in character. Taking the case in which the molecules are regarded as volumeless masses, Joule, by very elementary reasoning, showed that we could obtain the equation

$$PV = \frac{2}{3}\,\frac{v^2}{2g}$$

where P denotes the pressure in lb. per sq. ft. exerted on the walls of the containing vessel, V the specific volume of the gas in cu. ft. per lb., whilst v^2 denotes the mean squared velocity of the molecules. Moreover, since this is proportional to the absolute temperature, we get for the perfect or Boscovich gas the equation

$$PV = RT$$

Let us now suppose that the containing vessel holds just one pound of our gas. Then, if we diminish its capacity by b, the pressure will be altered and its new value will be given by the relation

$$P(V - b) = RT$$

Now obviously it does not affect the argument in the least if the volume b is distributed equally amongst the molecules so that each now has an individual volume of its own. This then is the equation of state for an ideal gas as distinguished from the perfect gas, but it should be noted that whilst the covolume b is proportional to the aggregate volume of the molecules, its actual volume may be greater than this; being dependent on how closely the molecules can be packed. From the above consideration we get a first hint as to the form of the equation of state appropriate for steam. The author's equation does not contain this term b, which a consideration of the mechanism at work shows to be sensibly constant, quite unaffected by changes of temperature and pressure; and little if at all by co-aggregation of the molecules. It is this covolume b which is responsible for the fact that hydrogen is heated on expansion through a porous plug. In short, the work done in delivering the compressed gas to the plug is P_1V_1 ft-lb. per lb., whilst that done in

[1] *Engineering*, Jan. 21, 1921, p. 63.

leaving the plug and, say, pushing aside the atmosphere, is P_2V_2. If there were no change of temperature we should have

$$P_2(V_2 - b) = P_1(V_1 - b)$$

or

$$P_1V_1 = P_2V_2 + b(P_1 - P_2)$$

so that the work done by the gas in leaving the plug is less than that required to deliver the compressed gas to the plug.

In general, however, a gas is cooled by expansion through a plug; and here again a consideration of the mechanism at work affords a hint as to how the gas equation must be modified to include this additional fact. The kinetic theory shows us that the number of molecules in one cubic centimeter of any gas, at a given temperature and pressure, is the same whether the gas be a vapor such as steam or "plu-perfect" like hydrogen. Professor Millikan has shown by experiments on the electron that at standard temperature and pressure the number of molecules of H_2 in a cubic centimeter is 2.705×10^{19} and also he finds that the weight of this molecule is $1.662 \times 2 \times 10^{-27}$ grammes. Therefore, the weight of one molecule of H_2O is $\frac{1.008 \times 2 + 16}{2}(1.662 \times 2 \times 10^{-27}) = 2.994 \times 10^{-26}$ grammes. At 100 deg. cent. and one atmosphere pressure there will be $\frac{273.1}{373.1} \times 2.705 \times 10^{19} = 1.980 \times 10^{19}$ molecules in one cubic centimeter. Hence the weight of one cubic centimeter of steam at the boiling piont should be $1.980 \times 10^{19} \times 2.994 \times 10^{-26} = 5.928 \times 10^{-7}$ grammes. The actual experimental figure is 5.99×10^{-7} grammes, which shows conclusively that the average weight of the molecules is more than if they were all represented by H_2O.

Obviously what has happened is that, in at any rate some of the collisions between molecules, the partners hold together for an appreciable time, during which they count as but a single molecule from the standpoint of the kinetic theory. It is evident that the more closely the molecules are packed, that is to say the higher the pressure, the more collisions there will be and the greater the opportunity for forming partnerships. So long, however, as it was believed that the dissolution as well as the formation of partnerships was directly due to collisions, no satisfactory explanation was forthcoming of the cooling effect generally observed when a gas is expanded through a porous plug. By reducing the pressure, the number of encounters capable of dissolving an existing partnership would be reduced in the same proportion as those capable of forming new ones, and the proportion of co-aggregated molecules would be unaltered in a porous plug experiment. This difficulty disappears if we accept Professor Perrin's view that collisions play little or no part in the dissociation of chemical compounds, and extend his theory to the co-aggregated molecules of a vapor, which are held together by exactly similar forces to those concerned in the formation of chemical compounds. The effective agent in all cases of dissociation whether it be the tearing of an electron out of an atom by X-rays or the dissolution of the bonds uniting the co-agregated molecules of a vapor is radiation. Now the nature of the radiation traversing a gas is fixed almost wholly by its temperature, hence, although the rate of formation of partnerships is reduced by the expansion of a vapor through a porous plug the mechanism that dissolves the partnerships remains practically unaffected. The loops of electric force, of which, according to modern ideas, radiation is composed, are unaltered in number and character and whenever a co-aggregated molecule encounters an appropriate loop, suitably oriented, the partners are torn assunder. Of course after an expansion of a vapor each loop has, on an average, to traverse a greater distance before it succeeds in breaking a partnership than it would have if the pressure had not been decreased, but the net result is that the proportion of co-aggregate molecules is reduced, and as the dissociation involves an absorption of energy, most gases and all vapors are cooled by throttling. In the case of hydrogen at ordinary temperatures and pressures, however, the co-aggregation is small, and the dissociation produced on throttling it absorbs less energy than the difference between the work required to deliver the gas to the plug, and the work done by the gas in escaping from the plug. Hence, in this case, there is a resultant heating effect.

The above consideration leads us to conclude that a rational form for the equation of state of a vapor is

$$V - b = \frac{RT}{P} - c \qquad [1]$$

where c represents the volume lost by the pairing of a certain proportion of the molecules.

In selecting his formula for c, the author seems to have been governed solely by mathematical considerations. He has no less than five independent constants, not one of which has any physical interpretation, and it would be very surprising if he could not, with this multitude of constants, fit very closely any consistent experiments on the properties of steam, whilst maintaining the necessary compatibility of his equation of state with his equation for the total heat. It turns out, however, that using but three constants, every one of which has a physical significance, Callendar has obtained an equation of state which, over the range for which reliable data are available, represents the experimental facts at least as well as the author's, whilst incomparably superior for purposes of theoretical discussion or practical application.

Any attempt to deduce from the author's formulas an equation to represent the adiabatic expansion of superheated or supersaturated steam results in an almost inconceivably complicated expression, utterly useless for all practical purposes, and the complication is enhanced if an expansion be effected with less than unit efficiency, which is always the case in steam turbine practice. Of course charts may be constructed to represent the author's equations, but whatever their utility in ordinary routine work charts are almost useless for theoretical purposes, and also when it is necessary to deal with small ratios of expansion.

Coming now to the total heat equations, it is well known that the internal energy of an ideal gas is represented by the equation

$$E = \frac{1}{\gamma - 1} P(V - b)$$

where γ is the index in adiabatic expansion. Of course in no actual gas does this equation hold to the bitter end. At very low temperatures the molecules of hydrogen appear to lose their power of acquiring spins, whilst at high temperatures, the molecules of most gases appear to acquire additional degrees of freedom, and their internal energy is then due not merely to velocities and spins, but is in part represented by internal disturbances of some kind, probably incipient ionizations. Hence no one formula of reasonable simplicity can be framed to represent either the total heat or the internal energy of a gas through the whole of the experimental range.

Fortunately, the properties of steam under extreme conditions have no engineering interest. The nature of our materials of construction limits us to a temperature of 350 deg. cent., or thereabouts, as a maximum; whilst it is barely practicable to attain temperatures of less than about 18 deg. cent. in our condensers. In important work again pressures are hardly likely to go much above 750 lb., and even this figure will probably be rare owing to boiler troubles.

Hence in framing equations for the total heat of steam, we are justified in taking as a basis for development the known form of the expression for an ideal gas. So far as the internal energy of the steam is represented by the spins and the velocities of its molecules it must be capable of expression in a form similar to that of the ideal gas.

Hence we may write for the internal energy of steam

$$E = \frac{10}{3} P(V - b) + B \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where B represents energy not due to velocities or spins.

From physical considerations we see that B must for the most part represent work done in separating molecules from each other so as to form the vapor to which should perhaps be added plus or minus terms depending on co-aggregation phenomena. Hence at any rate a large fraction of B must be a physical constant.

Comparison with experiment shows that within the range over which reliable data are available B is sensibly constant, an observation which accounts for the fact that in adiabatic expansion superheated steam follows the law

$$P(V - b)^{1.3} = \text{constant} \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

or

$$PT^{\frac{-13}{3}} = \text{constant} \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

This latter equation has been carefully verified by precise experiments with delicate platinum thermometers.

Calendar has shown that Equations [2], [3] and [4] will be consistent with each other and with the equation

$$V - b = \frac{RT}{P} - c$$

if c be chosen so that we get

$$V - b = \frac{RT}{P} - \frac{1}{P}\psi\left(\frac{P}{T^{\frac{13}{3}}}\right)$$

where ψ is any function whatever of $\frac{P}{T^{\frac{13}{3}}}$. Obviously by suitably choosing ψ we could represent any consistent experiments on steam whilst still satisfying the law of adiabatic expansion. Callendar finds, however, that within the practical range c can be considered a function of T only and represented by a single term so that we get for our equation of state

$$(V - b) = \frac{RT}{P} - 0.4213\left(\frac{373.1}{T}\right)^{\frac{10}{3}} \qquad [5]$$

From this equation and from Equation [2], the equation for the internal energy E, we can derive all other equations needed to represent the properties of steam. Amongst others we can calculate the saturation pressure. The values thus found are intermediate between the closely concordant values of Regnault and Holborn and Henning from 0 deg. up to 200 deg. cent. above which the experiments are unreliable, the apparatus used by the latter observers in particular being subject to systematic errors which become sensible at high temperatures, so that the recorded pressures are a little too high, or rather the temperatures are a little too low.

It is very remarkable that two such simple expressions as Equations [2] and [5] should represent within the practical working range the properties of steam with such extraordinary accuracy. There is in the writer's opinion not the slightest probability of the author's purely empirical, complicated, and not more accurate formulas being adopted here where Callendar's tables have been accepted as the standard by the British Electrical and Allied Manufactures Association and by the Nozzle Committee of the Institution of Mechanical Engineers.

In concluding, the writer may call attention to another remarkable discovery due to Callendar, viz., the physical explanation of the rise in the specific heat of a liquid with increasing temperatures. It is well known that if salt be dissolved in a liquid its molecules can give rise to an osmotic pressure, the value of which is exactly the same as if these molecules had, to themselves, the whole of the space occupied by their solvent. In the case of a vapor the surface film of the liquid is pervious to the vapor molecules, so that the "gas" pressure on both sides of the surface film is the same, or in other words, a liquid in equilibrium with its vapor dissolves its own volume of this vapor, and the latent heat represented by this volume is responsible for the apparent rise of the specific heat of a liquid with increase of temperature. In the case of steam this appears to be accurately true for temperatures over 40 deg. cent. Below this limit the specific heat of water is anomalous, as Rowland showed many years ago. The accepted explanation is that up to 40 deg. cent. water contains dissolved in it molecules of H_6O_3, which however are completely dissociated at temperatures above 40 deg. cent. This discovery of Callendar's leads to a very simple formula for the entropy of the liquid. It may be observed incidentally that it is wrong to deduce this entropy from the specific heats as generally tabulated since these have commonly been obtained with water heated at atmospheric pressure instead of merely under the pressure due to its own vapor. The error of course is very small.

To sum up, the writer has little use for purely empirical formulas, and not very much more even for equations arrived at merely by mathematical reasoning. Such convey no picture of what is actually going on.

THE AUTHOR. The first purpose of the paper was, by full graphical comparisons of data and formulations, to submit a report on the state of knowledge of the subject. A principal thing shown is the present insufficiency of experimental information and inconsistency of steam tables. In thus making a concrete and definite showing of a condition with which all workers with properties of steam were more or less familiar, the paper has helped to forward the movement begun by the Conference in Boston on June 23. Numerous comments and suggestions looking toward a better experimental knowledge are met by the report of that conference, published in MECHANICAL ENGINEERING for August 1921.

Regarding the second purpose of the paper, an addition to the knowledge of the subject in the new formulation proposed by the author, there is a good deal to be said here. Discussion of this phase falls naturally into three divisions, concerned with,

a The form and validity of characteristic equations based on present data

b Comparison with the divergent Callendar equations on present data

c Comparison with the very similar Goodenough equations.

In establishing characteristic equations for steam the fundamental idea is that of orderly and definite departure of actual from ideal or "perfect" gas. The perfect gas is a rational entity: assuming that its molecules are indefinitely small compared with the space in which they move and that they are perfectly elastic and nonattractive, the equation $pv = BT$ can be reasoned out; and assuming further that the kinetic energy of the molecules is all due to linear or translational velocity, specific heat will be constant or the heat equation will be $h = b + c_pT$.

At zero pressure any influence of molecular bulk and attraction vanishes, so that the actual gas agrees with its ideal prototype. With rise of pressure and density the real substance contracts in both pv and h along the isotherm, as illustrated by Figs. 1 and 2. Beside this, all gases have been found to show a rise of specific heat with temperature, apparently due to an increasing absorption of energy into some action within the molecule.

It is imaginable that we might have such complete and definite quantitative knowledge of the laws of molecular action as to be able to build up synthetically expressions for the departures from perfect-gas behavior. Then the equations of state would be completely rational, in the sense of rigorously logical mathematical development; they would automatically satisfy all the theoretical interrelations; and when evaluated they would conform to every line of experimental observation. In order, however, to furnish a means of independent approach to the problem, the quantitative laws of secondary molecular behavior would have to be found in some other way than by inference from the very experiments to be formulated; unless independently derived they would have only confirmatory and explanatory value.

Limitations of space, together with adherence to the purpose of presenting only matter of quantitative effect or relationship, barred from the paper description of hypotheses as to the form of the molecular actions which cause differences between actual and ideal steam. Mr. Martin's contribution of those ideas makes the presentation of the subject more complete.

As to the direct interpretation of experiment, it is the author's understanding that by a "purely empirical" curve or formula is meant one established essentially as follows: Plot points of observtion, sketch in a fair curve, and select or find by trial a suitable form or equation for the point series; then choose good guide points and fix the constants of the equation by the method of indeterminate coefficients or some equivalent method. In the paper Formulas [**23**] for water volume and [**24**] for variation of heat of liquid are wholly of this type, as is also the very important pressure-temperature Equation [**21**]. To be thus empirical is no reflection upon the character and standing of a formula, provided that the data are adequate in range and quality.

The method used by Goodenough and by the author builds a theoretical structure upon the simpler and more fully known empirical relations, checking it by the quantities more difficult and less complete of determination. There is reason, if not rationality, in the choice of Equation [10] as the governing relationship, because it does thus make the simpler physical quantities fundamental. Selection is offered from a number of theoretical relations, nowhere better presented than in the article to which Dr. Moss refers; but theory does not indicate one as more effective than another, so that it is correct to let convenience control choice.

SCHEME OF FORMULATION

Characteristic equations developed with the help of relation [10] depend directly upon the empirical precision of only two comparatively simple quantities, namely,

a Gas constant B, Eqs. [4] to [7] and [27]; this is not an isolated constant for steam, but is tied in with the values for other gases by the molecular weight of H_2O. Line 4 of Tables 1 and 2 shows a variation of 7 in 10,000, but Linde's higher value, Eq. [19], more than doubles the apparent range of possible variation

b Specific heat at zero pressure c_{po} and its integral h_o, with the constant of integration fixed by reaction from the Clapeyron check; this is very closely determined by the data plotted in Figs. 16 to 18, over the range from 200 to 1200 deg. fahr.

The further data that actually enter into the determination of the equations, although not directly involved like B and c_{po}, are those needed for applying Clapeyron's law, namely,

c The pressure-temperature relation for saturated steam, as expressed in the Clapeyron ratio c; note that this ratio involves the mechanical equivalent J

d The specific volume w and heat content q of water under saturation conditions.

Of these the first is fully and accurately known and formulated. The other two are subject to very small uncertainty up to at least 550 deg. fahr. or 1000 lb. pressure. With primary data thus ranging to 1000 lb. and beyond, the low limit of direct measurements of volume and heat content is not so serious a matter as might appear.

The function of those direct data, which are not actually used in the mathematical operation of closely fixing the constants of the equations, is to show or suggest trends and to serve as guides or for final tests by comparison. Thus the (H) isotherms and specific-heat curves in Figs. 13 and 16 to 18 were not even tried out until after the general equations had fully and definitely fixed; but their close conformity to the experimental points is a strong indication that the physical basis and rational construction of the equations are correct. The saturation data had much more influence during the process, since the operation of applying the Clapeyron check naturally furnished curves to be compared with them; unfortunately their lack of precision works against close determinative value.

The process outlined in the last four paragraphs is to a large degree rational, if that term is broadened to mean something more than mere mathematical synthesis. To call it "purely empirical" or to speak of "adjusting coefficients . . . in an endeavor to wind snakelike through clusters of conflicting data" is decidedly superficial, to say the least. As a matter of fact, considering the difficulty of high precision in some lines of experiment, the data are remarkably consistent. The author has by no means attempted an indiscriminate averaging of all the data, as suggested by Mr. Martin; it was in the exercise of just discrimination that Thomas' superheat experiments were left out of the comparisons.

REASONED CONSIDERATIONS

The general form in Equations [27] and [28], with relations [11] to [14], is not at all an arbitrary or a personal assumption, but has been evolved through study of the subject by a number of investigators: while not claiming exclusive physical sanction it does meet and can express every physical indication, with adequate flexibility and completeness and with maximum convenience. Its application began with Callendar, whose use of the straight isotherm in pv and h and p is a first approximation, fairly good over a limited range. Goodenough tied toegther the two parts of the contractive term, by using only one exponent on T. The complete form here presented is advantageously freed from that restriction.

Following the ideas set forth in Par. 7 of the paper, a general consideration governing formulation for steam is that the lower-pressure range of use and experiment is far more important than the high-pressure region toward the critical state. Equations that begin at the zero-pressure limit of conformity to ideal steam will be much more serviceable than (possibly) more general forms that might work downward from the critical pressure. Our type of service equation is capable of complete consistency and ample accuracy up to about 1500 lb. pressure; the fact that farther up its curves will require graphical adjustment does not at all detract from its practical merit.

Taken individually, the contractive terms and factors are subject to certain physical indications which help to fix values or constants, even though they cannot be completely separated out for an independent measure of each influence. Thus for the straight-line or slope term Y or Y' the considerations which led Callendar to use exponent $m = 3.33$ are very little modified by the introduction of the second term in p^s. The author's trial values ranged from 3.33 to 3.50, with the latter finally giving best results in the whole combination. In the curvature term Z or Z' exponent s on pressure p is the primary variable, with n on T depending upon it. In the concept that departure from the perfect gas should be progressively greater in kind and in amount at high density and toward saturation there is good reason why s ought not to be less than 2. With 1.5, for instance, the second derivative of the isotherm, which measures rate of change of slope and determines curvature, will contain the divisor $p^{0.5}$. As pressure approaches zero the radius of curvature will go with it; and the excess curvature thus introduced at low pressure is what causes the extreme variation of throttling coefficient M as deduced from the Goodenough equation. The trial values ranged from 1.5 to 2.4, but the best result was got with simple 2. With larger s a bigger exponent n must be used on T in order to meet the Clapeyron check; and this meets the reasonable requirement and physical indication (the latter from other gases) that the isotherms ought to straighten up pretty rapidly with rise of temperature.

SUNDRY QUESTIONS

The question of real physical definiteness of the state of dry satu ration was raised during the discussion. Certainly the steam table or formula conceives the saturation point R, in either form of Fig. 23, as the sharp intersection of the constant-pressure isotherm of vaporization with the dropping-pressure isotherm of superheated steam. Physically there seems to be some of the vagueness of definition that would make an isothermal operation follow the rounded corner

DE. Reversal of this tendency, in compression, would be a manifestation of the phenomenon of supersaturation.

The time element in an action such as is depicted in Fig. 23 would help to account for the divergence between the Smith points S_1 and S_2 on the total heat diagram, Fig. 12. Too much weight must not, however, be given to the discordant high point S_2, for with very slow evaporation the correction for radiation will be abnormally large, with greater likelihood of error. All that direct experiment has done is to indicate a range from 1149 to 1152 B.t.u.

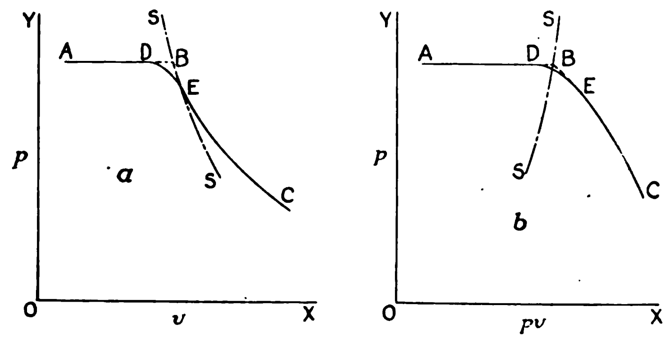


FIG. 23 ILLUSTRATION OF THE SATURATION POST

for saturation total heat at 212 deg. fahr. Clapeyron ratio c (tested and validated by application over a long range), with gas constant B and including mechanical equivalent J, is what really fixes this total heat in a complete formulation.

One misapprehension to be corrected is found in Dr. Moss' statement that Goodenough and the author have applied Clapyron's law to an independent saturation equation. What each has really done is to evaluate the general equations with coincident saturation values of t and p, then check between those results. The special saturation formulas exhibited in Figs. 8 to 10 are used only for the upper half of the total pressure range, above about 1500 lb.

Someone has asked for the classic throttling data: Fig. 24 shows these, with curves from the three sets of equations (M instead of C for Mollier-Callendar). Their divergence is such that almost anybody can get some consolation from the diagram. The Callendar curves differ decidedly from the others. It is interesting to note how the big differences in M, Fig. 22, integrate into relative small differences in the t-p curves over a fairly wide range.

THE CALLENDAR EQUATIONS

The foundation of Callendar's formulation is the assumption (on empirical grounds) that in the adiabatic expansion of superheated steam the relation p/T^{m+1} = const. is rigorously true, where $m + 1$ is the $k/(k + 1)$ or $c_p/(c_p - c_v)$ of the adiabatic of a perfect gas. The value of $m + 1$ was found to be 13/3, and since $c_p - c_v = AB$ and $AB = 0.11012$, the value of c_{po} at zero pressure, taken to be

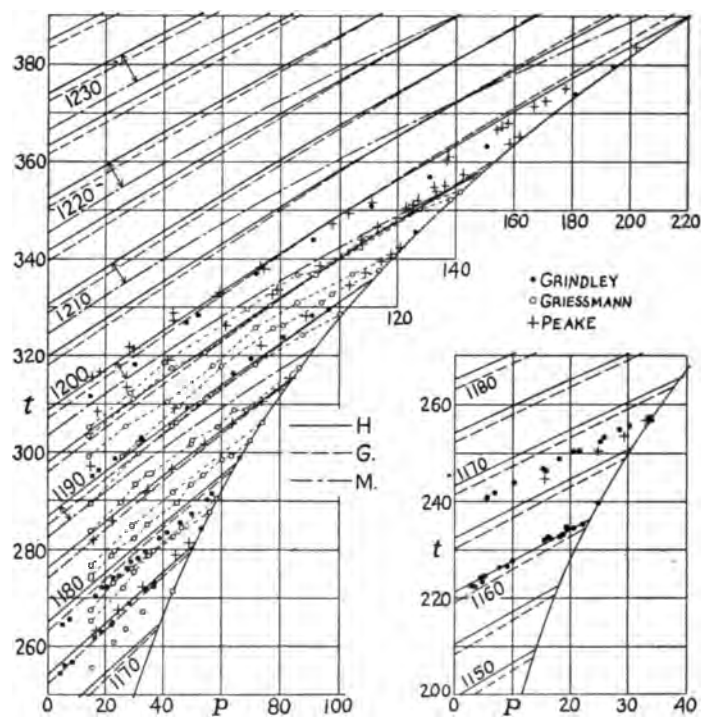


FIG. 24 PLOT OF CLASSIC THROTTLING DATA

constant, is computed as 0.47712. From constant m is reasoned the form of Eq. [27] without its last term. The heat equation is not reduced to the terms of [28], but is of equivalent effect and gives identical results. Observations of the cooling effect M in throttling fixed what is here the constant E and checked exponent m.

Naturally this equation is simpler and easier to handle than the more elaborate form: but in constant c_{po} and in straight-line isotherms it does not fit the facts. Over the range of common steam practice it is a fair approximation; but what we are after now is a form that fits all the known facts, has correct trend where it leaves the region of experiment, and can be extended with high probability of correctness to high pressure ranges.

A statement in the paper, "nor does any attempt seem to have been made to satisfy Clapeyron's relation," must be modified to read ". . . to make an independent Clapeyron check." What was done was, in effect, to get the p–t saturation relation by integrating $(dp/dt)_s$ out of r/u, Eq. [18], with r and u from the respective characteristic equations; this was not (could not be) the exact procedure, but so long as the proper theoretical relations were preserved another road led to the same result. It happens that up to 200 deg. cent. this agrees excellently with experiment, as shown by Fig. 7 and by Fig. 13 in Callendar's book. But at the top of his table, for $t = 491$ deg. fahr., p is given as 613 lb. abs., against 625.7 from Goodenough's Equation [21], corresponding with the two per cent divergence in Fig. 7 at this temperature.

The experiments on which was based the primary assumption in the third paragraph preceding were made by observing simultaneous pressures and temperatures during expansion in the cylinder of a steam engine using superheated steam and running as fast as 100 r.p.m. Pressures were measured from the indicator card, temperatures were taken by means of a thermocouple of very fine wires, set into the end of the piston rod so that it would be out in the body of steam, away from wall influence. In order to see how much room or margin such an experimentation has in which to differentiate types of steam equations, it is of interest to examine an adiabatic from a modern formulation. For this purpose the adiabatic for entropy $s = 1.7$, which cuts the saturation line at 30.7 lb. and 251.6 deg. fahr., is taken from the Goodenough table as follows:

ADIABATIC FOR ENTROPY $s = 1.7$, FROM GOODENOUGH TABLE

p	T	v	$\frac{pv}{T}$	$\frac{\Delta pv}{\Delta T}$	$m+1$
500	1362.3	1.598	0.586		
				.....	.....
320	1233.1	2.256	0.585		
				0.602	4.48
200	1107.6	3.23	0.584		
				0.595	4.38
125	992.9	4.64	0.583		
				0.587	4.31
80	893.7	6.51	0.583		
				0.584	4.24
60	834.7	8.11	0.583		
				0.583	4.21
40	757.7	11.04	0.583		
				0.583	4.20
(1)	(2)	(3)	(4)	(5)	(6)

In this tabulation absolute pressure p is the argument, temperature T and volume v are got by interpolation. Column (4) shows that pv bears nearly a constant ratio to T; what variation occurs is more clearly shown by the interval ratios in column (5). Index

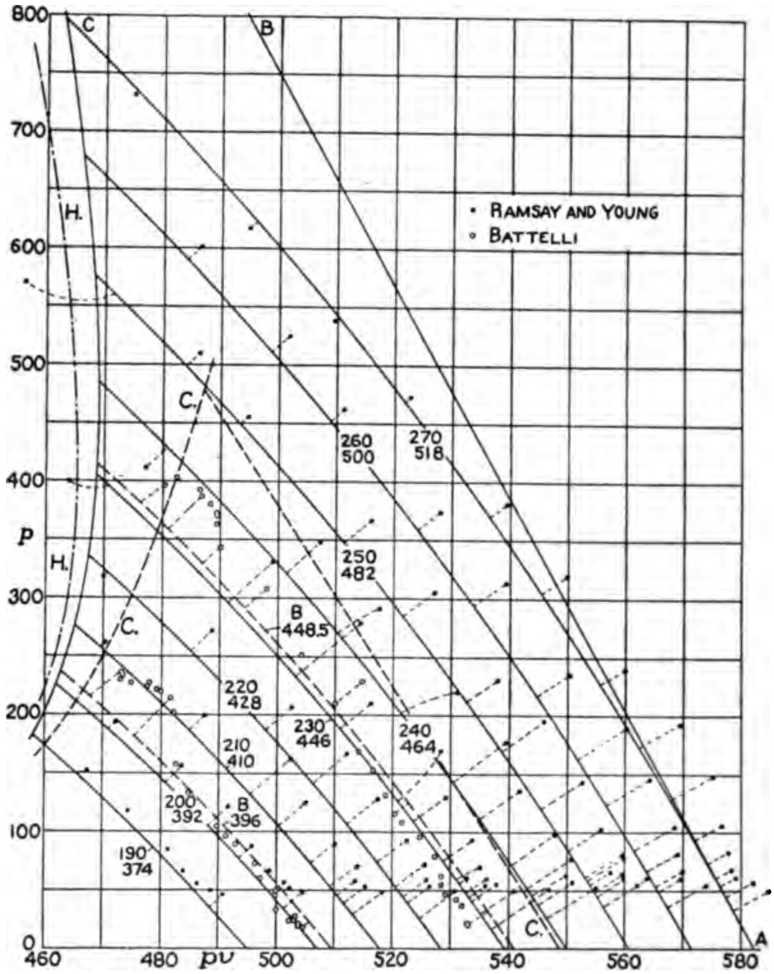


FIG. 25 RAMSAY AND YOUNG VOLUME MEASUREMENTS

$m + 1$, computed from p and T for each interval, is the last quantity General particulars for a series of these adiabatics are given in the second tabulation, showing ranges of variation of the two computed results (4) and (6).

PARTICULARS OF SERIES OF GOODENOUGH ADIABATICS

s	p-range	p-sat.	pv/T	$m + 1$
1.5	800–400	331.4	0.527–.523	4.21–4.19
1.6	800–125	105.5	0.568–.564	4.36–4.22
1.7	500–40	30.7	0.586–.583	4.48–4.20
1.8	125–9	8.8	0.592–.590	4.32–4.17
1.9	48–3	2.56	0.594–.593	4.32–4.17
2.0	14–1	0.76	0.594	4.32–4.16

Study of the pressure ranges indicates that where the experiments were probably made the computed value of $m + 1$ will range about from 4.30 to 4.20, corresponding to 1.30 to 1.31 for the primary adiabatic index $k = c_p/c_v$ (if the two were related as for a perfect gas). With one coördinate depending on an instrument no more precise than the steam engine indicator, and with at least a reasonable doubt as whether the performance observed was quite 100 per cent adiabatic, it does not appear that within such a narrow margin of range the type of experiment can have furnished a positive or exclusive indication of type of steam equation.

On the question of curved isotherms and of absolute steam volumes the data in Fig. 25 are worthy of presentation. The drawing was made for the author's paper of 1913, hence the principal body of curves belongs to the formulation here labeled (H2). On this have been drawn in the Callendar saturation curve, also that from Table 4. Points are tied to isotherms of the temperatures of observation. The evident inaccuracies, fairly consistent, are easily accounted for by the minute amounts of substance handled. Quantitative value is low, but the whole indication favors both shape and dimensions of the newer formulation.

THE GOODENOUGH EQUATIONS

On the close agreement between the (G) and (H) *results* no further comment need be made, except perhaps to repeat that it is fortuitious, not intentional or exactly confirmatory, at high pressures. For all present practical purposes and until new high-pressure data become available, the Goodenough table can be accepted with a considerable degree of confidence as to the absolute correctness of its quantities. In respect to internal consistency it is much better than anything else that is in shape for routine use.

The author cheerfully concedes that he has followed in Goodenough's footsteps, in accepting and only slightly modifying the form of equation of which Goodenough first made full application. He does not concede, however, that either this form or its valuation is a purely empirical attempt to satisfy experimental data. That the form is generally reasonable has been argued in preceding paragraphs. As to valuation, there is good reason for every one of the changes summed up under heading Reasoned Considerations.

Of course there must be dependence upon direct observation, for there is (now, at least) no other possible basis of quantitative fixture. The situation is that each observed quantity is the overall result of a number of combined influences. We should be very glad to take these apart and measure them separately, but that cannot be done. As to the character of some of these influences we have obvious physical hypotheses, not necessarily complete and exclusive and not quantitatively expressible. Some of these influences — or better, perhaps, relations among their effects — are expressed by reasoned theoretical relations among the rates of change of the properties or "coördinates" of the substance.

To the author's mind the several lines of observed relationship among steam properties bear a good deal of analogy to a group of simultaneous equations. Some of them are quite fully expressed, others are rather vague and sketchy; but there are *more than enough* for the determination of a common solution. Suppose that we select the best known physical relations, combine them with the theoretical requirements for the same quantities, and get a definite general solution; then if this solution agrees with the redundant physical relations its probability of correctness is high. As there are more constituents, each a definite thing even though but partly known, the probability of representing all except by an essentially correct general scheme tends toward zero.

Now the process used in getting such a solution must itself be rigorously correct; and here the author continues to believe that Goodenough has done something indefensible in making the variable covolume term help to satisfy Clapeyron's relation. Physically this expedient has the peculiar effect of carrying over into the gaseous state the dimensions of a molecular behavior that occurs in the liquid state: for if used along the saturation line it must be made a component of the whole superheated state also. From the viewpoint of formulation there is the awkward step of taking a quantity that has naturally been fixed as a function of t and using it as a function of p, but without reformulation. As to computation, along the superheat isotherm this small extra variable is an unmitigated nuisance. In mathematical theory there is a real error, in carrying pw through relation [10] as though w were a constant. If w is carried over to the state of superheat it must be as a function of p alone, for

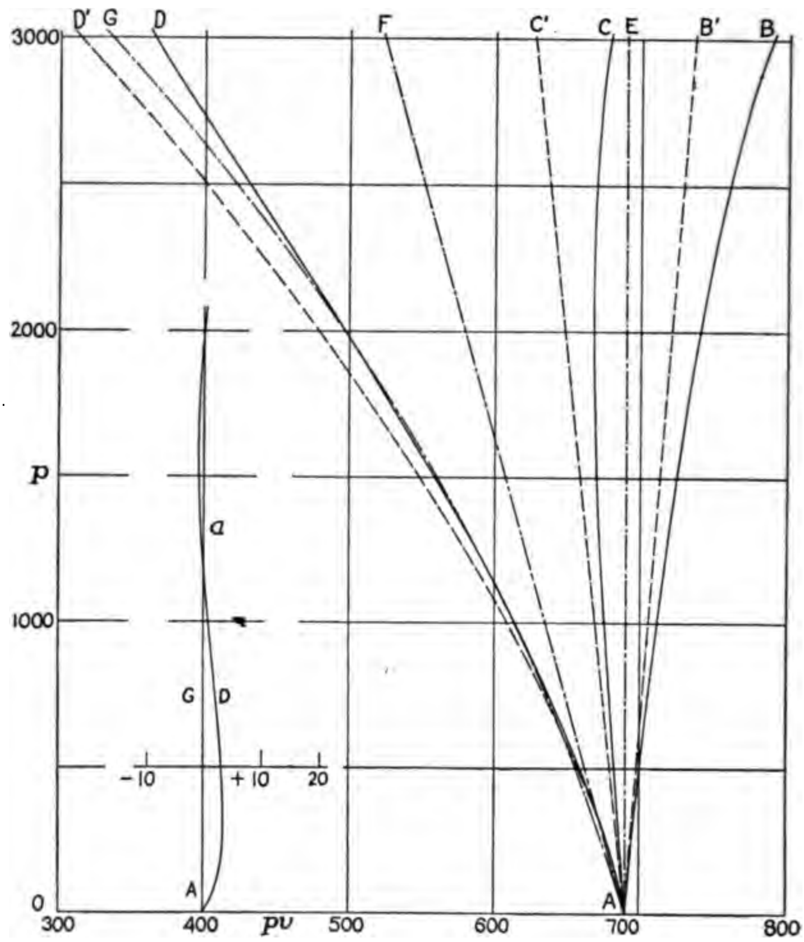


FIG. 26 DETERMINATION OF 700–DEG. ISOTHERM

curve AB of Fig. 26 will be the same in form (simply translated) for every temperature. Then the second term of expression [63] will be, not simple Apw, but $A\int wdp$.

The effect of using variable liquid volume as covolume term is shown by a complete workout of the 700-deg. isotherm of pv on p in Fig. 26; this runs well above the range of claimed applicability of either formulation, but the tendency of a scheme may fairly be judged by extreme extension. The full lines show Goodenough's method, AB for $(BT + pw)$, AC got by deducting Y, then AD by further deducting Z – compare Fig. 3. The dotted lines result from use of constant w_0, which makes AB′ and AC′ straight lines and gives AD′ for pv a markedly different course from AD. For comparison (H) curves are added in dot-and-dash, term BT alone giving the vertical AE, while AF separates terms Y and Z and AG shows pv. Note how very much smaller curvature term Z is relative to slope term Y in the (H) solution; along with the larger exponent of n on T in the (H) form of Z, this helps in a proper diminution of curvature as temperature gets higher.

A matter already discussed namely, the too great curvature of isotherm at low pressures due to use of only 1.5 for exponent s, is here illustrated by portion (a) of Fig. 26, where the (H) curve AG is rectified and the magnified departure of the (G) curve AD from it is plotted. At the top the extra curvature effect introduced by AB more than straightens out isotherm AD above 2000 lb., which is entirely illogical: but with that element omitted and working from a straight line AC′, the effect of low exponent s is further seen in the relatively less curvature of AD′ as compared with AG at high range, in spite of the relatively larger (G) value of term Z.

Whether or not to use the covulume term in steam equations that are not intended to be fully determinative more than halfway up to the critical pressure is a question that will have to await fuller data in the high-pressure region. At the top of Fig. 21, at 600 deg. saturation temperature, the effect of legitimate use of pw_0 and Apw_0 would be to diminish h_s from about 1170 to 1140, or by 2.5 per cent, and $(pv)_s$ from 420 to a little less than 400, or by 5 per cent. From the viewpoint of practical formulation this seems to be the largest question left open.

No. 1770

SELF-SUPPORTING CHIMNEYS TO WITHSTAND EARTHQUAKE

By C. R. Weymouth, San Francisco, Cal.
Member of the Society

The studies of the Japanese Imperial Investigation Committee on the nature of the stresses developed in chimneys in consequence of earthquakes are summarized by the author, who also examines the conclusions of the committee in the light of data on failures of chimneys secured from inspections of wrecks caused by the San Francisco earthquake of 1906. *An advancement to the theory developed by Professor Omori, Chairman of the Japanese Committee, on the design of chimneys to withstand earthquakes, is then presented. This assumes that the chimney, instead of being looked upon as rigid, should be considered as a body oscillating about the center of percussion. Reference is made in the paper to the design of various chimneys where the theory outlined has been applied.*

BECAUSE of the prevalence of earthquakes in Japan, and the immense damage resulting from them to buildings and other structures, the Japanese Government, a number of years ago, created the Imperial Earthquake Investigation Committee with a view to having it undertake a scientific study of earthquake phenomena, as well as of the effect of earthquakes on structures, etc. The Committee's reports would fill a good-sized volume. Many of these have been translated into English, and are obtainable at a few of the leading libraries in this country. Professor Omori, chairman of this committee, is the world's greatest authority on earthquake problems.

2 Although this subject is one of more or less academic interest, there are, however, certain localities in this country, mainly on the Pacific Coast, and many foreign localities, where the design of structures necessitates a careful consideration of the forces due to earthquakes. With the increasing tendency to invest American capital in foreign countries, American engineers will be required to design structures to withstand earthquakes.

3 A brief review of the work of Professor Omori, together with certain data pertaining to the San Francisco earthquake of

1906, a discussion of what the writer believes to be a limitation in a portion of Omori's work, and also an advancement in the theory of the design of chimneys to withstand earthquake are presented in this paper.

4 Earthquakes are due to various causes, the principal one being a slippage of the crust of the earth along fault lines. Earthquakes, of course, vary from mild trembles, barely perceptible and lasting but a few seconds, to violent shocks lasting several minutes and capable of demolishing very substantial structures. The principal movement is in a horizontal plane, but the vibration occurs in a number of directions in the same plane. There is generally a slight, but sometimes a severe, movement in a vertical direction, and also a severe twisting effect. On account of the combination of horizontal and vertical motion, earth waves visible to the eye frequently occur during earthquakes.

5 Earth movements are recorded at most astronomical observatories by means of seismographic instruments which record the displacement of the earth in three coördinates by measuring the amplitude, frequency, and duration of the earth wave.

6 By means of various earthquake records, values have been established for the maximum rate of acceleration of an earth particle, the maximum amplitude of an earth particle during a complete vibration, and the time for the complete period of vibration; and from these data it is possible to estimate approximately the stresses and strains in simple structures subject to earthquake shock. A sufficient number of earthquakes have been recorded and classified during recent years to make known the maximum intensity of shock experienced in certain localities, and the best that an engineer can do is to design a structure based on recorded experience. There will, however, always be the doubt as to whether some future earthquake may not be more severe than those which have been recorded in the past.

7 The most severe earthquake recorded in Japan was the Mino-Owari great earthquake of 1891, for which Omori gives the following estimates:

Displacement of an earth particle................233 mm.
Complete period of the principal motion, being a wave
cycle.................................... 1.3 sec.
Duration of earthquake......................... 4 min. 30 sec.
Maximum acceleration observed at different localities
ranged from.......... 3,000 to 10,000 mm. per sec. per sec

8 For the purpose of classifying various earthquakes Omori has devised an absolute scale of earthquakes intensities as follows:

The following absolute scale of destructive earthquakes, or the relation between the maximum acceleration of the earthquake motion and the damage produced, has been deduced chiefly from analysis of the Mino-Owari earthquake, the intensity being arbitrarily divided into seven classes. It is to be noted that the scale applies principally to Japan.

1 Maximum acceleration, 300 mm. per sec. per sec. The motion is sufficiently strong to cause people generally to run out of doors. Brick walls of bad construction are slightly cracked; plasters of some old dozo (godowns) shaken down; furniture overthrown; wooden houses so much shaken that cracking noises are produced; trees visibly shaken; waters in ponds rendered slightly turbid in consequence of the disturbance of the mud; pendulum clocks stopped; a few factory chimneys of very bad construction damaged.

2 Maximum acceleration, 900 mm. per sec. per sec. Walls in Japanese houses are cracked; old wooden houses thrown slightly out of the vertical; tombstones and stone lanterns of bad construction overturned, etc. In a few cases changes are produced in hot springs and mineral waters. Ordinary factory chimneys are not damaged.

3 Maximum acceleration, 1200 mm. per sec. per sec. About one factory chimney in every four is damaged; brick houses of bad construction partially or totally destroyed; a few old wooden dwelling houses and warehouses totally destroyed; wooden bridges slightly damaged; some tombstones and stone lanterns overturned; shoji (Japanese paper-covered sliding doors) broken; roof tiles of wooden houses disturbed; some rock fragments thrown down from mountain sides.

4 Maximum acceleration, 2000 mm. per sec. per sec. All factory chimneys are broken; most of the ordinary brick buildings partially or totally destroyed; some wooden houses totally destroyed; wooden sliding doors and shoji mostly thrown out of the grooves; cracks 2 or 3 in. in width produced in low and soft grounds; embankments slightly damaged here and there; wooden bridges partially destroyed; ordinary stone lanterns overturned.

5 Maximum acceleration 2500 mm. per sec. per sec. All ordinary brick houses are very severely damaged; about 3 per cent of the wooden houses totally destroyed; a few tera, or Buddhist temples, thrown down; embankments severely damaged; railway lines slightly curved or contorted; ordinary tombstones overturned; ishigaki, or masonary walls damaged here and there; cracks 1 or 2 ft. in width produced along river banks; waters in rivers and ditches thrown over the banks; wells mostly affected with changes in their waters; landslips produced.

6 Maximum acceleration, 4000 mm. per sec. per sec. Most of the tera, or Buddhist temples, are thrown down; 50 to 80 per cent of the wooden houses totally destroyed; embankments shattered almost to pieces; roads made through paddy fields so much cracked and depressed as to stop the passage of wagons and horses; railway lines very much contorted; large iron bridges destroyed; wooden bridges partially or totally damaged;

tombstones of stable construction overturned; cracks a few feet in width formed in the ground, accompanied sometimes by the ejection of water and sand; earthenware buried in the ground mostly broken; low grounds, such as paddy fields, very greatly convulsed, both horizontally and vertically, sometimes causing trees and vegetables to die; numerous landslips produced.

7 Maximum acceleration much above 4000 mm. per sec. per sec. All buildings, except a very few wooden houses, are totally destroyed; some houses, gates, etc., projected 1 to 3 ft.; remarkable landslips produced accompanied by faults and shears of the ground.

In the above scale of the seismic intensity, the earthquake motion has been assumed to be entirely horizontal. This supposition would not, except in places very near to the epicenter, cause sensible errors in the result.

9 A complete report was made of the San Francisco Earthquake by the California Earthquake Commission. This report, which has been published by the Smithsonian Institute, gives numerous charts, diagrams, photographs along fault lines and of wrecked structures, etc., and its study will be of value to any one desiring to pursue this subject further.

10 Seismographic records were almost valueless for the San Francisco earthquake because practically all of the instruments at the local observatories were put out of commission by the intensity of the shock, the full amplitude greatly exceeding their limits of movement. Omori visited San Francisco soon after the earthquake and, judging from the damage done, estimated the complete amplitude of the most severe shock to have been about 4 in., and the complete period of vibration of an earthquake wave about 1 sec. He also noted shear along the fault lines at different points from 16 to 20 ft.

11 Professor Lawson, a member of the California Earthquake Commission, estimated, from the Omori scale, that the acceleration for various formations in San Francisco had been as follows:

Foundation	Acceleration mm./sec.^{2}
Serpentine	250
Made land	1100
Marsh	3000
Sandstone	250 to 600
Made land	2900
Sand	600
Sandstone	400
Sand (Mission Valley)	1100
Marsh	3000
Sundry solid rocks	250

12 It will be noted from these data that earthquake shocks are least severe on structures built on rock foundations and most severe on structures built on loose or filled ground.

13 Omori discusses the force due to earthquake as impulsive or gradual, defining the former as a force applied to an elastic body so rapidly that it is finished in a time interval infinitely small in comparison to the period of natural vibration of the body, and the latter as a force applied so slowly that the body assumes a position of equilibrium without being thrown into vibration.

14 Within moderate limits of intensity of earthquake shock, Omori shows that structures are likely to fail which have a natural period of vibration materially less than the period of the earthquake, and structures are not likely to fail which have a natural period of vibration equal to, or greater than, the period of the earthquake. It is apparent that structures in the former class are short in comparison with their width, or diameter, and, until failure occurs, are accelerated throughout their mass by the earthquake vibration, inducing stresses resulting from the acceleration of the mass of the structure as a whole. Structures of the latter class are generally tall in comparison with their width, are more or less flexible, and, owing to their natural period of vibration with respect to the earthquake period of vibration, are able to yield in a measure to the earth movement, and are not accelerated as a whole at a rate equal to the earth's acceleration, and, consequently, are stressed to a less extent than in more rigid structures.

15 Omori brings out the well-known proposition that a force impulsively applied to an elastic body produces a stress double that caused by the same force when gradually applied, and further states that since brick fractures immediately the limit of elasticity is exceeded, the strength of a brick column against a force implied impulsively is half that against the same force applied gradually.

16 Having reduced the investigation of earthquakes to a basis of numerical acceleration and stated periods of vibration, Omori proceeds to show that for any given structure, such as a vertical prism resting on its base, the force necessary to accelerate the mass is to be computed from the well-known formula that the force is equal to the total mass of the structure, multiplied by the acceleration, divided by g; assuming, of course, that the mass is accelerated as a whole. This force may be considered as applied at the center of gravity, and it exerts a turning moment at the base of the prism equal to the computed force multiplied by the height of the center of gravity above the base.

17 In one of his reports Omori gives the results of investigation of forty-nine self-supporting brick chimneys. These failed at points from 24 per cent to 94 per cent of the height of the chimney, the average failure being at a point about two-thirds of the height of the chimney, while practically none of the chimneys failed at the base. He points out that the point of failure corresponds approximately to the center of percussion of the chimney with reference to a horizontal axis at the base of the chimney. Unfortunately, no self-supporting steel chimneys nor reinforced-concrete chimneys were in existence in Japan during the severe earthquake shocks of 1891 and 1894.

18 Omori shows that the failure of chimneys at the center of percussion instead of at the base of the chimney indicates a tendency to oscillate, as it were, about the center of percussion. There is a whip-snapping effect of the upper portion of the chimney which may cause a condition of resonance. In chimneys which have failed portions of the top of the chimney have been thrown down, striking the ground never at a great distance from the base of the chimney.

19 The writer investigated the failures of a number of chimneys in the San Francisco earthquake of 1906, and found that brick chimneys failed in about the same manner as observed by Omori. The tallest chimney investigated was 240 ft. in height, and failed at a point about two-thirds its height. Many of the large brick chimneys in the San Francisco District failed, although a few on rock foundation and especially well built did not fail by having a portion of the chimney height demolished; but the chimneys that did not suffer in this way were at least cracked either vertically or horizontally.

20 There were no reinforced-concrete chimneys in San Francisco at the time of the earthquake, and but one large self-supporting steel chimney, this being 10 ft. in diameter by 120 ft. in height. This chimney, although brick-lined, was not injured by the earthquake. A 150-ft. self-supporting steel chimney at the Mare Island Navy Yard was uninjured by the earthquake. A 217-ft. brick-lined self-supporting steel chimney at Salinas, Cal., likewise was uninjured by the earthquake.

21 Omori made an interesting series of experiments with brick columns on a shaking table, duplicating in effect earthquake motions of varying intensities. The results of these experiments are too elaborate for presentation in this paper. They showed conclusively that estimates of the tensile strength of brickwork are

exceedingly uncertain and undependable, owing to the variation in the character of workmanship in laying the brick.

22 Omori has measured the natural period of vibration of a number of self-supporting brick chimneys. One chimney 50 ft. in height was found to have a natural period of vibration of about one second.

23 Recently a very tall concrete chimney has been erected in Japan for the Kuhari Mining Co., Saganoseki, in the Oita Prefecture, Kyushu. The main shaft has a height of 550 ft.; the inside diameter at the top is 23 ft. 6 in.; the base is circular and 95 ft. in diameter; the natural period of complete vibration of the chimney was observed by Omori to be 2.5 sec. The range of motion or double amplitude of the top of the chimney was 1 in. at a strong gale of 24 m. per sec., but it reached 7.7 in. at a hurricane wind blowing 35 m. per sec. The chimney is located on hard rock, on which it is estimated an earthquake movement would be comparatively weak, probably having an intensity of less than 500 mm. per sec. per sec. A 100-ft. reinforced-concrete chimney 3½ ft. inside diameter at the top was found to have a period of vibration of 0.8 sec.

24 Some interesting observations of a chimney at Providence, R. I., are given by Dr. D. S. Jacobus, in his paper on Counterweights for Large Engines.[1] The stack was built on a mass of concrete which extended under the building structure. Referring to the vibration of the chimney due to the reciprocating engine, Dr. Jacobus states:

> At a point near the extreme end of the foundation where the No. 1 engine was located the foundation was found to shake 0.008 in. Measurements made near the top of the chimney, which was erected to the height of about 175 ft. above the ground, showed that the maximum shake with the engine running at its ordinary speed of 90 r.p.m. was about 0.02 in. After measuring the vibration of the chimney with the engine running at its ordinary speed the engine was shut down and a marked result took place when its speed fell in harmony with the time of vibration of the chimney. When this occurred, the chimney shook to such an extent that the motion was beyond the range of the special instrument. The total movement of the pointer of the instrument was such that the chimney was shown to move more than one-eighth of an inch. In constructing the chimney the workman had noticed that when they came to a height of about 130 ft. the vibration was much greater than it was after the chimney was built higher. This made it appear that at the height of 130 ft., at which there was the most shake, the time of the vibration of the chimney was in harmony with the number of revolutions made by the engine.

[1] Trans.Am.Soc.M.E., vol. 26 (1906), p. 531.

25 Professor Omori has also collected many data with respect to the effect of earthquakes on bridge piers.

26 Following the San Francisco earthquake and fire the city of San Francisco installed an elaborate high-pressure fire-fighting system, including two high-pressure salt-water pumping plants; the structures were designed to withstand earthquakes by utilizing the latest information obtainable from the San Francisco Earthquake Commission, an acceleration of 6.2 ft. per sec. per sec. being stipulated. Pumping plant No. 1 included four chimneys specified not to exceed 90 ft. in height, to minimize the earthquake risk, and measuring 64 in. inside diameter at the top. These chimneys were finally made of reinforced concrete and self-supporting.

27 As the writer's firm undertook to build a substantial part of this pumping plant he engaged a local engineer of prominence to make a special design of chimney to withstand earthquakes, and in the course of the next few years several other chimneys were designed in the same way. These designs followed Omori's theory, as outlined above; the bending moments due to earthquake were calculated for all sections, assuming the lateral displacement of the chimney to result from the earthquake acceleration imparted to the mass of the structure as a whole and the chimney shaft to remain vertical at all times. This theory gave a strength of chimney at the base materially greater than at the higher sections, and in view of Omori's observation of the frequency of breakage at two-thirds of the height, the reinforcing was empirically increased at this section. No severe earthquakes have tested the accuracy of the design. Recently the writer had occasion to examine the subject further in connection with the erection of a chimney 220 ft. in height and 14 ft. inside diameter at the top.

28 The chimney in question was constructed during the war time, and on account of the prohibitive price of self-supporting steel chimneys, decision was made in favor of reinforced concrete.

29 The writer concluded that while the chimney might have a tendency to sway somewhat about the base as an axis, the principal design should be made on the basis of accelerative rotation of the chimney about its two-thirds point, instead of a motion of accelerated translation parallel to itself, as heretofore assumed. The theory of rotation harmonizes with the observed failures of chimneys, whereas the theory of translation does not.

30 Considering the chimney as a rigid body and assuming the two-thirds point to be at rest and the acceleration of the earth to

be known, then the angular acceleration about the two-thirds point becomes immediately known, and from the principles of moment of inertia, the bending moment at different sections is easily computed. Such theory, however, would only apply if the chimney were under no strain at its base or foundation. If such rotation did exist, the chimney remaining rigid, then there must occur either a yielding of the soil under the chimney foundation, or a deflection or fracture of its base, or both. An investigation of the foundations of chimneys which had failed in the San Francisco earthquake did not lend support to this theory, and the writer finally concluded that the element of flexibility of the chimney shaft played a most important part in chimney design.

31 A further important consideration in chimney design is the deflection of the chimney from the base to its neutral point, assumed to be at two-thirds of its height. It is only on the assumption that a tall chimney will deflect within its elastic limit one-half the amount of the earth's vibration that its stability during a severe earthquake can be maintained without rupture.

32 The failure to find serious horizontal cracks near the base of the shaft or in the foundations of some large self-supporting brick chimneys can only be explained on the theory of deflection. This theory is not touched upon by Omori in any of his reports.

33 Fortunately it is comparatively easy to make an approximate estimate of the elastic deflection of a self-supporting chimney, but unfortunately the writer is not able to indicate the factor of safety to be applied to this deflection as compared with the earth's vibration.

34 The writer consulted E. P. Lewis, Professor of Physics, University of California, who submitted a statement as follows:

Since seeing you I have considered the questions raised in the memorandum which you gave me regarding the design of chimneys. As I told you, the problem seems so complex, and the factors so obscure, that I can do no more than make some suggestions regarding the physical principles involved.

In the first place, referring to Omori's view that the center of percussion of the chimney remains at rest under the impulses of an earthquake, Omori evidently makes this assumption by considering the case of an impulse at one end of the uniform rod on which no other forces act. In this case the center of percussion is a fixed axis of rotation, as is easily shown by considering that the resultant moment of momentum, due to the impulse, must vanish if taken about the point at which the impulse is applied. In the case of the uniform rod this is one-third the length of the rod from the opposite end. I think, however, that Omori's assumption is incorrect. The above applies to a body free from external forces after the application of the impulse, whereas the chimney remains fixed to its base, and the principle applies to a perfectly rigid body, which the chimney is not.

As a matter of fact, I am confident that something like this happens:

The impulse comes along; an elastic wave runs up the chimney; during its first passage the center of percussion has a minimum motion; the earth movement ceases; the inertia of the chimney carries it forward until it is brought to rest by elastic reactions, which cause it to vibrate in the opposite direction, and any effect at the center of percussion vanishes. The amplitude of the motion of the top of the chimney (ventral segment of the vibration) probably is many times greater than that of the earth. If the earth vibrations are approximately synchronous with those of the chimney, the amplitude may become very great; if the frequency of the earth vibration is three times that of the chimney, "overtones" will appear with a nodal point one-third of its length

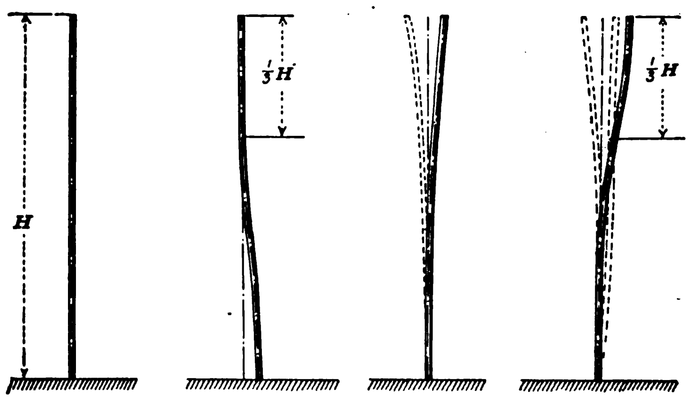


FIG. 1 VIBRATIONS OF AN ELASTIC ROD FIXED AT ITS BASE

a Initial position. *b* Effect of center of percussion. *c* Effect when center of percussion vanishes. *d* Effect of a 3:1 overtone.

from the top of the chimney (if of uniform section), thus adding to the possible effect due to the center of percussion.

A wooden or glass rod clamped to a massive base, which is then subjected to a shock or vibration, behaves as I have indicated above, and I am sure that a tall chimney behaves in qualitatively the same way.

I think you are right in considering that the vibratory motion of the chimney is of fundamental importance, and that account must likewise be taken of vertical displacements. Below [See Fig. 1] I sketch some of the phases of the motion of the chimney as I imagine it.

It occurs to me that especially violent effects might occur if succeeding non-periodic shocks come at such phases of the chimney vibration due to the first shock as to suddenly increase the acceleration.

I have given my ideas so far as they are clear to me. I do not speculate further because the problem is so difficult and obscure that I might easily suggest some error.

35 The sketches submitted by Professor Lewis are shown in Fig. 1. At *a* is shown a vibrating rod, fixed at the base, in its initial position; at *b* the effect of the center of percussion; at *c* the effect when the center of percussion vanishes; and at *d* the effect of a three-to-one overtone.

36 The ordinary chimney having much less flexibility than the experimental rod considered by Professor Lewis, it is the writer's opinion that the extreme effects which Professor Lewis mentions would not be experienced by a chimney with intensity of shock equal to that of the San Francisco earthquake.

37 In completing the design of the chimney in question, it was necessary for the writer to make certain assumptions, as follows:

1 That the minimum strength at any section must be equal, of course, to that required to withstand wind pressure; that an earthquake would not occur at a time of the hurricane wind velocity for which the chimney is designed; and that the addition of these forces would be unnecessarily conservative.

2 That the minimum strength at any section to withstand earthquake, should correspond to the equal acceleration of the chimney as a whole, considered as a rigid body, the axis of the chimney remaining vertical at all times; due to its flexibility, the chimney is not accelerated equally throughout, so that the writer assumed that this calculated moment should not be doubled, as customary for vibration.

3 That the minimum strength at any section to withstand earthquake should correspond to that required for the angular acceleration of the chimney about the center of percussion, as explained above; allowance being made for the doubling of stresses due to the vibratory or whip-snapping action of the chimney; a factor of safety must still be applied on account of the uncertainty of such calculations, as further outlined below.

4 That the vertical acceleration of the chimney should be considered as resulting from the vertical component of the earth wave. The stress computed in this manner will be generally found to be very small.

5 That in the case of reinforced-concrete chimneys the temperature stresses must be computed for temperatures corresponding to the average overload on boilers,

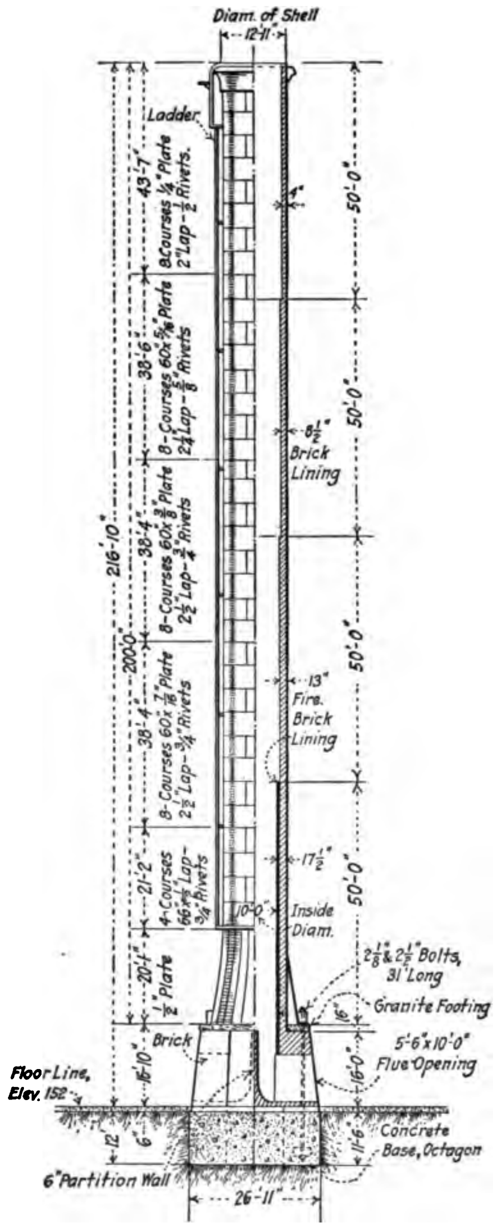


FIG. 2 SELF-SUPPORTING STEEL CHIMNEY, SPRECKELS SUGAR CO., SPRECKELS, CAL.

and this condition should be assumed as being coexistent with a possible earthquake, the temperature stresses being added to earthquake stresses.

6 That the foundation or base of the chimney should be very carefully investigated with respect to the moments produced at the base of the chimney by the earthquake, and transmitted through the foundation.

7 That the elastic deflection of the chimney through its two-thirds height, measured from the base, should be somewhat greater than the amplitude of an earth particle, the amplitude being taken as half the total movement of an earth particle.

38 As a check on the above theory of design, investigation was made along parallel lines of the brick-lined self-supporting steel chimney of the Spreckels Sugar Company, at Salinas, Cal., details of which are here reproduced, with the permission of the Spreckels Sugar Company, in Fig. 2. This chimney was erected in 1897, and was presumably designed to withstand wind pressure only. Having an inertia effect comparable with the chimney under consideration, and having successfully withstood the action of an earthquake of intensity approximately the same as that estimated for the reinforced-concrete chimney, it afforded a standard of comparison for testing the computed strength of the reinforced-concrete chimney. For any section, the ratio of the assumed elastic limit of the steel to the computed tensile stress, according to the above theory, gave a factor in all cases greater than unity, indicating either that, if the above theory is correct, the earthquake intensity could have been multiplied by this factor before the elastic limit of the steel would have been reached; or that the whip-snapping and inertia effects, due to earthquake, resulted in stresses greater than calculated, by a ratio numerically equal to this factor.

39 If the factor were not applied it does not necessarily follow that the reinforced-concrete chimney would be unsafe. It may, and probably does, indicate that the strength of the Spreckels chimney was somewhat greater than required to withstand the San Francisco earthquake.

40 The factors of safety, mentioned under (3), finally used, were as follows: Starting with a factor of 1 at the 20-ft. section (measured from the ground line) these factors were gradually increased up to 2.5 at the critical section, or two-thirds of the height, and this factor was continued to the top of the chimney. The amount

of steel required for this increased safety in the upper or critical sections was relatively small and inexpensive. On the other hand it seemed the part of wisdom to apply to the chimney such factors as would produce a stability against earthquake approximately equal to that of a chimney which had withstood earthquake shock.

41 Experience extending over a long period of time will alone indicate the proper factors to be applied in chimney design in order to insure the structure against earthquake acceleration. This, for the reasons stated above, was impossible in San Francisco. According to Omori, fifty years may elapse before another severe shock; the preceding one was in 1868.

42 The method used in computing the temperature stresses in the reinforced-concrete chimney is that given in Turneaure and Maurer's Principles of Reinforced-Concrete Construction, except that the temperature reinforcement was increased materially beyond that indicated in this textbook. The allowed tensile stresses of the reinforcing steel, as well as the compressive stresses in the concrete, due to earthquake, were normal. The temperature stresses alone were slightly above normal for concrete, and less than normal for steel. When these were added to the earthquake stresses, the final result gave a stress safely within the elastic limit for steel, but considerably above what would be considered a safe allowance for the compressive stress of concrete. Investigation of the temperature stresses shows that the above condition is unavoidable, and the conclusion is that while a chimney so designed would not fail by falling during an earthquake, it is easily believable that the combination of earthquake and temperature stresses would result in serious fractures at critical portions of the chimney.

43 A very important point in the design of such chimneys is the assumption of the difference between the temperature of the gases within the chimney and the temperature of the outer shell of the chimney. Some tests made by the writer indicated that this temperature difference is greater than was assumed in the example given by Turneaure and Maurer, and further data along these lines would be of much interest to designers.

44 The writer was unable to find any new material of light weight and of low conductivity to be used in lieu of the ordinary concrete or brick lining of a reinforced-concrete chimney; for manifestly it is highly important to minimize the mass of the chimney subject to vibration. It was finally necessary to assume a brick lining. The question then arose as to the height to which this should

be carried to minimize temperature stresses in the unlined upper portion of the chimney, which, from experience with earthquakes, is the weakest section of the chimney. It was considered impracticable to carry the chimney lining to this critical section; which indicated a manifest defect of the reinforced-concrete chimney for withstanding earthquakes.

45 It has recently been suggested to the writer by Mr. C. A. G. Weymouth that in the building of reinforced-concrete chimneys to withstand earthquakes it would be advantageous to use terra-cotta concrete, as developed by the Concrete Ship Section of the Emergency Fleet Corporation. As is well known, terra-cotta concrete materially reduces the weight per cubic foot, and in addition has a high compressive strength. It should be pointed out, however, that any saving in the weight of the chimney shaft, as compared with the ordinary concrete construction, would, of necessity, be made up by an increase in the weight of the base of the chimney to give the necessary stability against overturning from wind pressure.

46 The requirement under (7) for the deflection of the chimney shaft should be made with some margin of safety. No data are at present available, but it should be borne in mind that any increased elasticity of the chimney should not be secured at the sacrifice of strength actually necessary to resist moments in the lower portion of the chimney shaft.

47 It is hardly necessary to state that the chimney designer must give due consideration to the shear produced by earthquake, as well as the shear due to wind pressure; other stresses, however, are usually the determining factors.

48 It is of course obvious that a chimney to withstand earthquakes should be constructed of materials of maximum strength, of minimum weight, and with maximum elasticity. So far, this combination seems to be best supplied in the use of self-supporting steel chimneys, regardless of whether the conditions of operation require a chimney lining. In California, at least, the earthquake evidence testifies eloquently in favor of the use of steel chimneys.

49 In closing, the writer will state that while the ultimate solution of the earthquake chimney problem appears impossible in an accurate sense, it is nevertheless to be hoped that some of our engineering physicists will give further study to the rather special questions which it involves.

No. 1771

POLICIES FOR FUTURE POWER DEVELOPMENT

By Col. John Price Jackson, Philadelphia, Pa.
Member of the Society

The primary purposes of this paper are (1) to present the problem created in this country by the present acute shortage of central-station electric power, and the large growth of demand for such power; and (2) to set forth the physical, public, and financial relationships that appear essential for relieving the present stress and meeting the continually enlarging demand with reasonable economy and conservation of natural resources and labor. The first purpose is dealt with in the first half of the paper, and includes a comparison with the situation in England. Treatment called for by the second purpose begins with the heading Natural Power Districts, and extends to the end of the paper. The latter is largely an exposition of the possibilities and advantages of unified power systems for supplying the power required by natural power districts.

IN a large industrial district with which the writer is familiar there is a large load ready and waiting to be connected with the central-station systems, and were there a wholly adequate and reliable supply of power at the disposal of the power companies in this territory, their present large aggregate load would be greatly increased. Expressed opinions of industrial and central-station men, as well as various statistics, justify the conclusion that an equally pressing situation exists in most or all of our important industrial centers.

2 The shortage of central power is serious by reason of the fact that it interferes materially with the natural productive growth of our country and with the prosperity of its several communities. Another even more serious element relating to such power shortage is the fact that it continues the use of a large number of isolated steam and electrical plants, most of which are consuming from two to four times the amount of fuel which would be required were the power supplied from efficient central stations. Moreover, the lack of sufficiently adequate central power systems makes impossible the

Presented at the Annual Meeting, New York, December, 1920, of The American Society of Mechanical Engineers.

economic utilization of many water powers which would become available if they could pump their power in the form of electrical energy into great electrical distribution systems having heavy demands.

3 To so develop the central power systems within a reasonably short time that they will meet the present deficiency, absorb the great amount of unconnected or isolated loads, care for the natural growth of demand, and utilize our fuels and water powers most advantageously, represents, in this country, a problem of gigantic dimensions which will require much wisdom to solve.

THE ENGLISH SITUATION

4 An even more striking situation affects England, where an acute shortage of fuel and inadequate development of electrical power had placed her, by 1917, in a position in which she was unable to keep up with the increased industrial growth demanded for war purposes. In this regard the situation was somewhat analogous to our own a year later, and is therefore of interest to us. The Board of Trade, a governmental body in England, appointed investigating committees of eminent engineers to study the situation and recommend programs for overcoming the difficulty.

5 As a result of these investigations it was found that enormous quantities of fuel were being ineffectively consumed by reason of the use of large numbers of comparatively small central stations and by great numbers of isolated industrial plants.[1] The central stations also were unprepared to act in coöperation, and thus there were found in the area of greater London 70 concerns supplying electricity to the public with 70-odd generating stations, 50 types of systems, 10 different frequencies, and 24 different voltages.

6 As a result of these findings a bill was introduced in Parliament, which later became a law after modification of some of its compulsory elements. This law provides that there shall be a central body, known as the Electricity Commission. which is given wide powers relating to the national electric-power supply; that the British Isles shall be divided into large power districts in which electric-supply undertakings shall be "unified," the districting to be approved by further action of Parliament; that power boards composed of representatives of the power interests and others, shall

[1] Report of Electric Power Supply Committee (Cd. 9062), published in April 1919. (English.)

be constituted in each of these regions; and these boards shall have charge of the operation of all power plants and transmission lines in their respective territories, including isolated plants, as well as the development and promotion of plans for expansions. It further provides for the transfer of central plants and transmission lines to the boards under certain conditions and gives them large powers of enforcement.

DIRECTION OF DEVELOPMENT IN AMERICA

7 Though the problem is much the same, the conditions in England differ in many ways from those in the United States, for here our commercial power companies have, of their own initiative, progressed far beyond those in England. In this country the growth of large, economical power systems is distinctly and actively under way with but relatively little direct governmental supervision. Furthermore, while English municipal ownership of power utilities has been actively growing, in America the commercial power systems have been developing with enormous rapidity. Again, during the ten years ending with 1917, America increased her central-station output from 123,000 to 241,000 kw-hr. per year per employee, with similar reductions in the use of fuel and other supplies per kw-hr., while in England the coördinate development has been relatively less satisfactory.

8 However, though the United States is in a better position than England, the demand here is insistent for even more adequate and reliable power systems, for the purposes of reducing at least the rate of increase of depletion of our natural resources, and for better conservation of our labor. To meet this very proper demand requires concerted action. Much has already been done in concert by way of coöperation and unification, but the time has come when more concrete and highly crystallized coöperative policies can be successfully determined upon by privately owned companies. Only by proper policies of this kind can such extensive governmental interference as has occurred in England be obviated.

9 As the United States is entering into the coming era of peace, indications are that it will face, in its foreign trade end and in its home market, a sharper competition than ever before from the other civilized nations of the world. Such competition is healthy and desirable, for it places us in a position where we must use every endeavor to produce along all lines with the greatest economy. This

involves continued development of power production in the way which will most rapidly increase its efficiency in regard to the use of natural resources and labor, and at the same time provide, as far as feasible, a fully adequate and reliable power supply for all those in need of it, particularly for industrial purposes, wherever they may be located.

THE MAGNITUDE AND CHARACTER OF OUR POWER PROBLEM

10 The United States Geological Survey has recently tabulated the use of fuel for power purposes in the United States, based on estimates of 1919 power production and census figures.[1] It is estimated that 333,000,000 tons of coal were utilized in producing energy during 1919. Steam railways produced about 37.5 billion horsepower-hours and used 148 million tons of coal, or about 8 lb. per hp-hr. This includes the fuel for heating and auxiliaries. Manufactories, not including mines and quarries, and not supplied by central stations, produced about 34.5 billion horsepower-hours by the use of 145 million tons of coal. This coal was utilized in numerous, mostly small, plants at a cost of about 8.4 lb. of coal per hp-hr. In obtaining this figure, fuel used for heating in the manufacturing plants was deducted, but water power was included in the total horsepower-hour output. The fuel rate for the output of the steam apparatus alone was estimated to be about 9.8 lb. of coal per hp-hr. In addition to the above, there was produced by central power stations and electric railways about 53 billion horsepower-hours using 40.5 million tons of coal at a cost of 1.5 lb. of fuel per hp-hr. With water power deducted, this rises to 2.3 lb. for the steam plants alone. With our modern plants this figure is of course much less.

11 The field for savings in fuel that could be made were the aggregate of the loads named above furnished from efficient central-station systems is evident from these figures. A large part of the unconnected load undoubtedly can and will be rapidly connected for central-station supply, connection of other blocks will probably be delayed for economic or other reasons, and part is probably not of a character for economical central supply. Loads of the character of the last two will therefore delay or prevent obtaining the full savings in fuel which would be possible were our total industrial load supplied from economical central-station systems, but nevertheless

[1] Files of U. S. Geological Survey, Power Division, Sixth Ave., New York City (Henry Flood, Jr., Executive Secretary, under date of Nov. 1, 1920).

possible practical savings are very great. In addition to savings in fuel, very material savings in labor also can be made by the extended use of central power systems.

12 The equipment, as given by the same reference, required to produce the energies named heretofore, amounts to about 100 million horsepower, including steam-railway locomotion, of which nearly 20 million are installed in central electric power stations and electric railways, and an approximately equal amount in manufacfactories. Included in this grand total, about seven and three-quarters million horsepower are of water-power equipment. Something less than a third of the central-station equipment is hydraulic, or about five and three-quarters million horsepower. Economic considerations will largely determine the rapidity with which this latter figure can be increased by further development of the available unused water powers of the country.

13 Census figures show that from 1907 to 1917 the output of central stations grew from less than six billion kilowatt-hours to over 25 billion, or more than doubled each five years. It should be noted, however, that electric railways are not included in these as in the former figures. During the same period the value of the central power properties increased from approximately one to three billion dollars and the annual gross income from about 1.75 to 5.25 hundred million dollars.[1] That this growth has been continuing is borne out by an extensive investigation, made by the writer with Henry Flood, Jr., Frederick Darlington, and others, of a typical American power district where the central power business had just about doubled from 1915 to 1920. It was also found in the same district that there was every reason to predict another doubling of the business in the period from 1920 to 1925 and further substantial increases thereafter. Available information leads to the conclusion that conditions similar to those of this district prevail throughout much of this country.

14 The above figures referring to the enormous expected central power demand, and other considerations, indicate that the following are among the most important activities to be undertaken in connection with the power situation as it faces the country today:

a To so design the construction and direct the operation of central power systems as to make possible the highest practical economy and conservation of natural resources and labor; and also to provide a fully adequate supply of power for all purposes.

[1] United States Census, Central Electric and Power Stations, etc., 1917.

b To so arrange by interconnection of transmission systems and otherwise that the maximum amount of water power may be developed. The amount of water power utilized can and will greatly increase with the rapidly growing power demands, but in many important industrial sections of the country, where water power in scarce, it will require much wisdom to cause the increase to become a greater percentage of the total power than now exists. Many watersheds having reasonably constant flow, however, but not accessible for connection normally with central power systems, can possibly become of early value for developing the electrochemical processes of the country, and incidentally in reducing the consumption of fuel.

c To place steam plants, as far as possible, close to coal mines or on waterways in order that energy may be transmitted as much as possible in the form of electric power rather than in the form of coal, with resultant reductions of freight charges and relief to railroad traffic.

d To supply practically all industries with power from central power systems, which would result not only in an enormous coal saving, but also in a large saving in man power, maintenance supplies, and in transportation.

e To largely electrify the railways. This would make possible a great reduction in the fuel used for transportation purposes and relieve the railroads from carrying a large dead weight of coal freight for their own purposes, and also would make available the materially increased duty of railroad trackage and equipment which is possible by electrification.

f To not only make the central stations adequate for all of the above purposes and the expected future growth, but fully reliable.[1]

g To conserve a larger proportion of the valuable chemical constituents in the coal; such products are now recovered from only a small percentage of the total coal mined. Closely related with this problem is the conservation of our oil and natural-gas supplies. A discussion of these problems is not directly within the sphere of this paper, but it is believed that the suggestions for power development contained herein are along lines to coördinate with their solution. The problems are acute, for our oil and natural gas are showing signs of exhaustion, and though under present rates of consumption

[1] J. W. Lieb set forth emphatically the need of "absolute continuity of service" in the discussion of Electric Supply at the meeting of the American Institute of Electrical Engineers in February, 1920.

our coal may last a few thousand years, if the present increase in the rate of demand continues, it will be exhausted in a very limited time.

h To work out programs which will enable the financing of our power systems to be handled with facility. By the census there were three billion dollars invested in 1917 in the central-station industry, not including electric railways. About double that is needed now or as soon as the equipment can be produced, say, in 1922 or soon thereafter, and at least four times the 1917 capital will be required well before 1930 if reasonable progress is made.

15 The large sums of money indicated as necessary during the next few years for power development will in a large measure be spent even though no concerted effort be made to solve the power problem by a fully coördinated plan. But without concerted action there would be no possibility of such excellent returns, either in the form of earnings for the public and security holders, or in conservation of natural resources and labor. Were we advanced to a point of highly efficient central electrification of all power absorbers, with reasonable use of water power, the saving in fuel alone would amount to vast sums of money per year, with probable total savings of from two to three times as much if labor and all economies were included. The following suggestions with regard to power districts are intended to indicate the lines of a concerted policy of unification which would encourage these savings and at the same time accelerate central power development along the natural lines in which it is now advancing.

NATURAL POWER DISTRICTS

16 The trend of power development during the past few years, and the present tendency, is toward the enlargement and unification of the power systems in large natural power districts. The boundaries of a natural power district are largely determined by the considerations that the region should be sufficiently large and the load of a character to enable it to sustain and properly use generating plants of a size and character necessary to give high economy, and at the same time have a territory sufficiently compact that the exchange of power between the several parts can be carried on conveniently and effectively. Due regard must of course be paid to the location and character of natural power resources. The actual area of a district to fulfil these conditions may vary widely; thus, the conditions of load, population, and natural resources demand a much

larger territory for economical operation in California than is required by a district centering in New England.

17 A unified central power system supplying the power requirements of a natural power district inherently tends toward highly economical service, because such a system makes it possible to reduce to a minimum the aggregate amount of fuel used in the power plants through the possibility of raising the use of factors of the most economical plants of the region and lowering others, to take advantage of load diversities in the several parts of the district, to maintain a minimum of reserve equipment, to place necessary new plants at the points most effective from the standpoint of natural resources and the load, to more fully utilize water powers, and to permit a higher degree of reliability.

18 The selection of power sites for accessibility to coal mines, condensing water, water power, etc., has already become difficult with the present large stations and power demands, and with the great expected growth of these demands this problem will become more difficult. Unified action in a power district is of much value in this regard.

19 Those in charge of operations in a natural power district should design the unified system with the view to obtaining as far as practicable the same efficiency of production and delivery of power to all uses or users of power over the whole district, as is now attained in the big centers of power supply, if reasonably satisfactory conservation of our resources and the best welfare of our country and the power companies is to be obtained. It is evident that such objects require full standardization of voltages, frequencies, systems, etc., throughout the district, or better, throughout the whole country. Such standardization mostly exists in the United States, but it has not reached perfection.

20 At the present time the growth and unification of power districts is proceeding steadily, as may be noted by the situation in the New England States, the tying together of various companies and successful coöperation in western Pennsylvania and eastern Ohio, the getting together of a number of power companies in a group of the Southern States, in the unified control of the central-station systems of the Pacific Coast, etc. This kind of coöperative development was given a distinct and its most important impetus forward, during the war, by the Power Division of the War Industries Board, headed by Frederick Darlington, which was faced with the problem of relieving an acute shortage of both power and fuel,

and the development of additional power to meet the rapidly growing demands caused by war activities. Mr. Darlington, after having examined the subject, concluded that the wisest procedure was to take advantage of and promote the natural tendency toward unification and coöperation of power interests in natural power districts, and with that object in view, dealt with several sections of the country as distinct units, or districts, including those referred to above.[1]

21 Experience during the war and various results found in peace times are sufficient to lead to the conclusion that both investment and operating costs will usually be so reduced by unification in properly limited districts as to make this kind of power supply attractive from the financial point of view.

UNIFIED POWER SYSTEMS

22 By the term "unified power system" (or unification) as used heretofore in discussing natural power districts, is meant a system in which the several component parts or companies combine and interconnect to obtain the efficiency and service which would be rendered if the entire district were operated by a single power company. Further, the terms used and discussions of the power problem herein are to be understood as confined to the generation and trunk transmission of power only, and not to include the distribution from the trunk-line load centers and its retail sale unless otherwise indicated.

23 The purpose and scope of this paper do not warrant entering into a discussion of technical engineering features, but the transmission of the power in quite large unified systems is feasible with existing equipment; thus, in a certain natural power district, a practical transmission system with switching and substations was recently laid out for a future growth to over a million and a half kilowatts. The technical work in laying out this system was largely done and was all approved by some of the best transmission engineers of the country. It is believed that projects which exceed this can be handled safely, though the difficulties to be met and the skill required for their solution should not be minimized.

24 The most simple method of handling a unified system, covering a natural power district, is by placing the entire generating

[1] Report on the Power Situation During the War made by General Charles Keller to the Secretary of War.

and transmission apparatus in the hands of a single power company, either by purchase, lease, or the equivalent; where this is the condition no comment is necessary as to methods of operation or construction programs. Where it is not found advisable to arrange such a coalescing of the power companies the several systems of the several companies can be unified and operated as though they were under one ownership by the companies combining in the appointment of a power administrator, who is given full authority to direct the generation and delivery of power as he finds will result in the greatest economy to the entire district.

25 Where a power administrator is used, arrangements must also be made under which there will be an equitable distribution of earnings among the several companies. An example of a formal plan for accomplishing this is to have each integral company in the district compensated fully, with a reasonable profit, for its facilities in the way of power stations and trunk transmission lines with their substations, etc., which it holds ready for service. In other words, all company expense such as interest, depreciation, amortization, taxes, and insurance, with the addition of reserves and profits, constitutes readiness-to-serve expense for which the power companies are compensated. In addition to this, each company having generating stations or trunk transmission lines will be compensated in full for every operating expense incurred for power generated or transmitted under orders of the power administrator. The companies then pay for the power which they take from the unified system load centers, for retail distribution and sale, at rates per kilowatt-hour which, when applied to the aggregate power generated in the district, pay the combined ready-to-serve and operating charges named above. The price paid by a company for power purchased should, of course, have contained within it adjustments for load and power factors. The greatest difficulty entering into such a formal arrangement is the determination of the value of available facilities, but this need not prove an insurmountable obstacle.

26 A necessary corollary in this example is that the several integral companies in the district join together in laying out programs of construction in order that, as the load grows and additional stations and facilities are required, they may be developed in the way best to procure the highest economy for the district as a whole, and therefore, under the scheme dealt with, for each of the component parts.

27 In such programs of construction or in any arrangement of unification involving more than one company, it may readily be found advisable that the component companies join together in the development of a water power or the construction of a large steam power plant, say, at a coal mine, through the medium of a new company, or otherwise, rather than by settling the entire burden upon one of the component companies. A successful case of such joint action is to be found in the modern steam plant at Windsor, W. Va., on the Ohio River, built jointly by the West Penn Power Co. and the American Gas & Electric Co. through the agency of a third company created and owned by them.

28 Rather than have a single ownership, it may be found more advantageous in a natural power district to form a company which would make the interconnections, handle the interchange of power and settle the intercompany bills, as is done in Switzerland or at Windsor, or to work out other arrangements to meet the special conditions and needs.

29 It should be noted that the two companies at Windsor operate largely in different states, while the several interconnected companies in the entire power district in which they are located cover parts of Pennsylvania, Ohio, and West Virginia. Little difficulty has been encountered by reason of the crossing of state lines.

30 The idea of unification with a central power dispatcher, acting jointly for several companies in a district, is not a new one among engineers; as early as 1911, William B. Jackson, as the result of careful study of certain districts, recommended it and set forth in some detail the financial, operating, and installation advantages, and the possibility through it of reaching small communities not otherwise within the range of electrical supply.[1] Further, we have an example of a unified natural power district using a power administrator, which has proven satisfactory to the power companies within it and to its people. This is on the Pacific Coast, where the several companies have worked excellently in coöperation since 1918.[2]

31 For a recent period of power shortage accentuated by low water, unification arrangements were renewed by a written agreement among 17 power companies in Oregon and California, made

[1] Advantages of Unified Electric Systems Covering Large Territories, by William B. Jackson, Proc. A.I.E.E., vol. xxx, 1911, page 131.

[2] Reports of the Railroad Commission of California for July 1, 1917, to June 30, 1918, and for July 1, 1918, to June 30, 1920.

last March before the State Railroad Commission, a part of which follows:

> We, the undersigned, hereby pledge that we will obey and carry into effect to the limit of our ability all rules, regulations, and orders of the Railroad Commission concerning diminution of service or taking on of new business, and interchange or delivery of power between the several companies.

32 The Railroad Commission was voluntarily selected by the companies as their agent and arranged to exercise the full powers, thus delegated, through a power administrator. There seems to have been no vital reason for having selected a public board as power administrator rather than any other person or body in whom the power companies and people have full confidence.

33 It is altogether to be expected that as unified systems in the natural power districts of the country develop, they also will find it advantageous to make interconnection for emergency and other services. In some cases it may be found that switching arrangements for connecting up the ordinary transmission lines in contiguous districts will be sufficient. In other cases it may be found that several unified districts can, to advantage, join together in the development of a large water power, say, such as on the St. Lawrence and Niagara Rivers in connection with the New England and Middle States, and part of Ohio, and perhaps in the building of a system of plants in the coal regions of Pennslyvania and West Virginia, for supplying more easterly territories.

34 W. S. Murray has suggested that some such arrangements may be desirable between the Atlantic Seaboard districts which he is at present investigating as the representative of the United States Geological Survey.[1] It should be added here that the investigation being carried out by Mr. Murray and his staff, as a bureau of the United States Geological Survey, is in response to a special act and appropriation of Congress. He has been given wide scope to study the present conditions of power utilization in a belt of territory bordering the northeast Atlantic Seaboard with its enormous industrial and railroad activities and to work out plans for electrical supply which appear advantageous. He has surrounded himself with a group of strong advisers whose joint judgment will be generally

[1] Economical Supply of Electric Power, by W. S. Murray et al, Midwinter Convention of the A.I.E.E., February 1920.

accepted. The findings will doubtless give another impetus to the use of central-station power and to unification.

PUBLIC RELATIONSHIPS

35 One of the most important elements which should be considered in bringing about the unification of a power district is that of proper relationship with the public. Effective joint publicity should be inaugurated which would make clear to the people of the district the value of unification, and the ways in which the public can serve to aid in bringing about such a condition of power supply.

36 The prosperity of every industrial community and of the nation is vitally dependent upon success in properly meeting the present and expected power demands of the country. There is every reason, therefore, to believe that redoubled attention, on the part of the power companies, to the subject of public relations — particularly if activities of this sort are under the direction of joint authorities for entire power districts — will create an adequate popular support for unification and other measures necessary to meet the situation.

37 Special attention should be given to the development of laws whereby capital invested in power systems would be reasonably assured by law as to stability and return, instead of being subject to a modified status with every change of administration or public-service commission. One of our most capable and experienced financiers and industrialists in emphasizing this, recently said that, "Public-utility commission actions should be governed by definite laws." . . . "They should be bound to respect and recompense capital invested in public utilities according to plans definitely set out by enacted laws, so that the security to investors in railroads, electric power properties, gas plants, etc., would not be dependent on the personal attitude or judgment of transient commissions." Another man, also of high national standing, makes the statement that: "This is a government of laws and not of men, but there seems an increasing disposition to establish commissions not guided by any fixed or determined principles of law, but which have wide powers and whose decisions are frequently governed by expediency or opinion."

38 Laws should be enacted which would be as liberal in the matter of rights of way for powerplants, transmission lines, and other needs of the power companies, as is the case for railroads or

in various states with regard to water-power plants. It is particularly desirable also that the law be so drawn as to enable the power interests to share economies brought about by the ingenuity, energy, and capacity of their personnel.

39 With regard to this latter item it is well to call attention to the classical example of the South Metropolitan Gas Company of London, which is acting under a sliding-scale law. As the law stood in 1899, when the selling price of gas was not above 3s. 6d. per 1000 cu. ft., the return on the capital invested in the gas companies might be 10 per cent. (These figures have been modified, but the method is still in force.)[1] When the gas companies were able to reduce the price of gas, through economical operation, by a penny a thousand cubic feet, the return on the capital invested might be increased by $\frac{1}{4}$ of 1 per cent; increase of interest on the capital investment continued with each penny lowering of price, while raising of the price lowered the permissible dividend. It may be of interest to note that the returns on the capital gained under this system are so distributed as to give an incentive to the management and employees to make economies. A somewhat similar scheme is used by the Boston Gas Company of this country. The arrangement has the advantage of rewarding good service and at the same time bringing financial profits to the community. In the case of light and power companies the application of these methods is more complicated by reason of the varying rates charged for electric power, but the obstacles are not insurmountable and the principle is well worth considering.

40 The people of each power district should be brought into a closer relationship with the power companies by the distribution of securities to a maximum number of holders. If arrangements are made for payment in small installments and the stability and returns of the securities of most of our power companies are made known, there should be no difficulty in obtaining their wide distribution. Indeed, it is doubtful if there are many building and loan or savings bank associations which can show such excellent returns with reasonable safety as can our best organized electric-light and power companies. This, and obtaining financial support from large consumers, may not secure a major part of the necessary capital, but it will bring excellent returns in the nature of public support.

[1] Report on Profit Sharing and Labour Co-Partnership in the United Kingdom, 1912. (English.)

FINANCIAL CONSIDERATIONS

41 In order to procure the large amount of money necessary for obtaining adequate and reliable power in the United States, it is essential not only that the best plan be adopted from the engineering and business standpoints, but that it shall be one which is acceptable to the public and to public-service commissions. The plan must be one which will gain the definite approval of the latter, and the securities offered by power companies in obtaining necessary funds should, as said, not only have the approval of the commissions, but be practically safeguarded as to earnings through stable action of the Government. The financial situation, under a properly designed unified plan for power districts with proper inter-district connections, such as have been suggested, would better command the support of the public and protect their investments than would be the case were the necessary facilities installed and operated by several independent companies; in other words, there is undoubtedly better opportunity for such safety of investments and support by the public and the public-service commissions for a single well-worked-out plan covering a whole district, which would be clearly the right way of performing the work economically, than there would be for three or four different plans put on foot by several independent companies operating in one natural power district.

42 Since the success of the electrical power business in meeting industrial needs depends upon securing the necessary capital, this feature with regard to financing may easily be a controlling factor in the results. The power companies are practically obligated by their franchises and other contracts to supply the facilities needed to provide an economical and sufficient supply of electrical power. If they should not do this through failure to make the necessary financial arrangements, or for any other reason, the results which would naturally follow would be most unfortunate, and would probably lead to undesirable and in the end nationally injurious curtailment by the Government of the advantages of the strong personal initiative and incentive which now exist in the power industry of this country.

DISCUSSION

E. H. DeLany suggested the founding of a Federal Bureau of Engineering and Research. "I think," he said, "that every technical bureau that has ever been inaugurated by the Federal Government has proved itself worthy, and has the support of the people." He urged the creation of the Federal Bureau on the grounds that no private corporation or society could afford to advance the funds necessary for experimental purposes and that such a Bureau could do a great deal more than either corporations or societies in the way of securing the legislation necessary to carry on its work.

John W. Lieb, in referring to Colonel Jackson's statements of the conditions in London where there are a number of plants with various frequencies and voltages, said that we face no such difficulties in this country because the systems in our great cities are fairly thoroughly consolidated. He called attention to the fact that the British Commission assumes control, not merely of "linking in," as the general term is there, but also assumes to say what type of installation an industry should use and also whether the industry ought to be allowed to install such a plant or whether it should be made a part of a system which already exists. Furthermore, they assume the right to say what shall be the voltage and frequency in order that the proposed plant may form a part of the future prospective original system. Referring to capital expenditures and the availability of capital, Mr. Lieb said: "What the public wants is service, and the American public is willing to pay a reasonable and proper price for that service. But to obtain it, it must be possible to have adequate rates under proper supervision and regulation." Mr. Lieb also stressed the importance of the reliability of service, saying that it is absolutely essential in our great metropolitan centers to have absolutely reliable service, and that from an engineering standpoint a complete reliance on a transmission line alone, without an adequate steam stand-by, was not a safe and proper thing. The usual method of tying in and linking up of systems was also commented upon by Mr. Lieb. He urged the carrying of a joint reserve instead of each power company carrying its own reserve and meeting the capital charges thereon. "There is an important economic advantage," he said, "in having a joint reserve that is common to all." As to the tying in of systems, Mr. Lieb stated that what was necessary was not merely to have two original systems and then incorporate them by tying them together at the ends, but that the tie line should be of large capacity and should connect the systems at important generating points so that the load could be shifted back and forth at the points where it was needed. Mr. Lieb also referred to the work being done by the Superpower Survey and stated that if we wished as a nation to be leaders in industry we must go ahead with such comprehensive plans, which are essential to our economic national life.

Walter N. Polakov urged that the coal supply be conserved by utilizing distillation processes for the recovery of products now lost. We should put our coal, he said, through a low-temperature distillation process and thus obtain not only a smokeless, dustless, and odorless coal, but also fuel gas valuable for domestic and industrial purposes and a residue valuable for fertilizers. Benzol, which can be used as a substitute for gasoline, as well as coal tar could also be obtained.

Harold L. Doolittle spoke of the systems now employed in California for the interchange of power. The scheme functions through the Railroad Commission and practically all of the companies in the state have agreed that future transmission lines shall be so constructed that ultimately a 220,000-volt bus line will run from one end of the state to the other and that in this way it will be possible to take advantage of any diversity in the load. Experiments have already been made on the line, he said, and there is no question as to the practicability of the scheme. In regard to economy of operation, he pointed out that there should be some method adopted which would reward public-service corporations for increased efficiency. To accomplish this in California, he said, the Southern California Edison Company had placed before the Railroad Commission the proposition of establishing some standard of operation, so that if any saving were made the corporation would be allowed to put that money in a fund. In other words, create a surplus, so that at the end of the year rebates could be made, 50 per cent to the consumer, 25 per cent to the stockholders in the company, and 25 per cent to the employees. Mr. Doolittle stated that this proposition, however, had not as yet been acted upon.

W. S. Murray referred to the waste of power through the use of isolated plants and also by the railroads, and stated that the whole object and incentive of the Superpower Survey is to see that this may be corrected. Regarding the development of our water power, he said that the water power available in the superpower zone represents but a possible 20 per cent of the whole, and that therefore we must confine ourselves to a most careful consideration of the highest economic development of steam power. Mr. Murray also contributed a written discussion of Colonel Jackson's paper, extracts from which follow:

In connection with the railroads, the fuel consumption by steam engines against electric engines, respectively, in the passenger, freight, and switching service, has been shown to be 2, 2½, and 3 to 1, and this with a thermal efficiency of the central stations driving them not greater than 8 per cent. Today the efficiency can be made to be 18 per cent, and this means that those ratios will be in excess of 4, 5 and 6 to 1. The best steam freight locomotive when burning 5 tons throws 3 tons away.

Mr. Henry Flood, Engineer-Secretary for the Superpower Survey, has placed in my hands a collaborated statement which shows that had the superpower system been in service as of 1920, there would have been saved 30,000,000 tons of coal per annum. In 1925, based upon the growth of power on a conservative basis, these savings grow to 33,000,000 tons of coal, and as of 1930 the conservation that can be effected is estimated at 50,000,000 tons per annum.

These preliminary estimates of the savings possible through the centralization of power supply do not consider the complete electrification of the railroads in the district, nor the complete electrification of industry; for instance, on the heavy-traction railroads it is estimated that 6000 miles can be economically justified for electrification in 1920 of the total 30,000 miles within the region, and it is further estimated that the increase in electrification would grow only at a rate of traffic growth for this region up to the 1930 period, so that it considers the electrification in total by 1930 of only about 20 per cent of the total track mileage. In industry these estimates consider taking over to central power-supply sources only 50 per cent of the total power in the manufacturing industry, so that it is reasonable to state that these estimates are conservative and that they do not carry out the electrification of industry to the point of saturation.

Colonel Jackson points out the difficult situation in England with its innumerable small stations and varying frequencies. It is a lesson to us what we should avoid. Our plants are larger; our main frequencies are but two, and yet it is amazing to know that the capacity of the average-size plant in the Superpower Zone is not greater than 7900 kw., and its consumption of coal per kilowatt-hour is nearly 3 lb. The superpower stations projected will be on the order of 300,000 kw.; their economy should touch 1.5 lb. of coal per kw-hr. Again, a load factor of 40 per cent carried by a central station requires that station to burn 85 per cent of its coal before its kilowatt capacity has been taxed 35 per cent, and the remaining 65 per cent of its kilowatt capacity is used while burning the remaining 15 per cent of the coal. I think this visualizes what I have just said regarding the employment of the present large stations, which, if they are not in a position to furnish large blocks of kilowatt-hours, stand ready, however, as peak-load plants to furnish the capacity with which a very small amount of energy (kw-hr.) is involved.

Colonel Jackson says, "The time has come when more concrete and highly crystallized coöperative policies can be successfully determined upon by privately owned companies." I am in perfect accord with this statement, and I wish to emphasize at this point that the Superpower System as proposed stands alien to none, helpful to all, and that its construction will not be upon a competitive basis. The entity and franchise rights of the present central stations must be respected. The Superpower System through delivery and exchange will place energy upon the distributing mains of the present public utilities, and there its functions cease. Its duty will be the generation of a maximum amount of power for a minimum consumption of coal. It is to be purely a power and transmission adjunct and will have no relations with the customers of power, except that those customers be public utilities.

Through the uneconomical production of power, coal is in the yards, on the sidings, and on the main lines of the railroads. It is competing with the raw and finished commodities for cargo space. Think what a saving of 50,000,000 tons of coal per annum in 1930, within a district between Boston and Washington, inland 150 miles, means! If that amount of coal is not required to be mined, first we have conserved nature's storehouse; second, we have relieved the railroads of its transportation and created cargo space for our other high-priced commodities which industry and its expansion demand; and third, through the new process of power production as recommended in the Superpower System we will put a very large percentage of the remaining coal required for power and lighting in the form of electricity and thus create an overhead carrier system and automatically release cargo space for other important commodities.

Colonel Jackson's summation in Par. 14 of his paper is splendid, and the integration of these indeed furnishes an epitome of the specifications of the work undertaken by the Superpower Survey, viz.:

(*a*) Allocation of wastes and their amount, due to the present improper form of power production and distribution, and (*b*) The recommendation of the regional plant through the means of which these wastes may be eliminated.

I am struck by his reference in Par. 7 to the average cost of a kilowatt installed by central stations up to 1917. This is $333, including the distribution system. In the past when building a plant we have been required to build, due to the lack of interconnection, a plant and a quarter, the additional quarter being necessary for spare capacity. Through the proposed interconnection offered by the Superpower System, not only will this additional quarter capacity disappear, but through diversity the plant's capacity may be run below the 100 per cent line, and instead of our small plants (averaging but 7900 kw.) at a cost of possible $200 per kilowatt, the larger plants may be built at probably less than half that amount. Indeed, we have arrived at the period for the construction of enormous plants to carry the great base loads. The present unassociated load factor (and by unassociated is meant the independent operation of the industries and the railroads) now existing at 15 per cent will belifted to 50, and thus it will not be difficult to visualize a 4,000,000000-kw-hr. base load in a pool of power aggregating 31,000,000,000 kw-hr.

Colonel Jackson has said that special attention would be given to the development of laws whereby capital invested in power systems should be assured of a reasonable return, and in this statement he strikes a note of insistant requirement. Finance and engineering go together. It is indeed a case of *E Pluribus Unum*. Viewing the Superpower System from a commercial aspect and entirely apart from its basic and national importance, namely, the conservation of the millions of tons of coal previously referred to, the savings to be effected by it in money will represent $300,000,000 per annum, even at as early a period as 1925, which amount will steadily increase as years multiply the units of power required.

Charles W. Thomas referred to the newly created Federal Power Commission and its duty of granting preliminary and permanent licenses for the exploration and final development of water powers. He said that in his opinion influence should be brought to bear upon the Commission so that permanent permits would not be issued to those who might finally develop important water-power sites in an incompetent way.

Henry Harrison Suplee, referring to flood control of great rivers, particularly the Mississippi, the Ohio and its tributaries, stated that it had been suggested that by controlling the run-off rainfall much nearer the source and by supplying proper catchment basins, power could be developed, while at the same time losses due to floods could be entirely avoided. Mr. Suplee also spoke of the difficulty of securing local interest in one part of the country for the benefit of another and suggested that it would probably require a Federal bureau with appropriations from Congress to develop such a scheme.

H. P. Quick contributed a written discussion in which he reviewed the method of financing hydroelectric enterprises that have been built in this country, Canada, Mexico, Brazil and Spain, by money furnished partly by bankers, partly by capitalists, and partly by small investors. The promoters of these enterprises secured their money, he said, by first getting a majority control of existing going enterprises and future consumers of their power, and then issuing bonds and selling stock in new development companies, or railway, light and power companies organized for the purpose of hydroelectric development and power distribution in the various municipalities. These bonds and stock were sold as the reconstruction or improvement of the going concerns or as the new construction demanded, and the revenues increased with which to pay interest on the bonds. Of course the purchase of stock or the gift of stock having great prospective value increased the income of the concerns, and the ability, standing and character of the managing engineer-president-financier and his associates furnished the additional incentive to investors to open up.

As to the attitude of American bankers toward such enterprises Mr. Quick wrote as follows:

Did American bankers assist to any great extent in these developments of foreign enterprises? Not a bit of it, and perhaps they were wise in view of the revolutionary possibilities and actualities in some of these countries in recent years But plenty of foreigners — Canadian, Englishmen, Frenchmen, Belgian bankers and others invested in them, as they saw a large profit and thought the countries stable and were accustomed to such investments.

Now, if those same developments had been undertaken by the foreigners' own nationals, how would the undertakings have been financed? Why, by their governments, by securing control of going enterprises, or by special taxation, of utilities or communities to be benefited; and this revenue from taxation, or perhaps the revenue from imports and exports, or the income from foreign stocks and bonds as collateral, would be used or set aside periodically to pay the interest on new government bonds issued to bankers of other nations who would take them and sell them to their customers, clients, stockholders, etc., or they would be taken by the construction companies and sold in the same way.

Now that fact of government issued bonds with guaranteed income at high rates of interest from the start is the inducement that leads the United States bankers to take the securities of foreign stable governments, as they have been doing recently, which funds are to be used for hydroelectric developments that will enable electrification of railroads and industries, and make great savings in fuel. For, what bankers are as a rule looking for is securities that they can take at a profit and recommend to constituents as a profitable investment, either from their own investigation or that of the governments or municipalities they represent.

I believe the United States Government or its Federal Power Commission must do as the foreign governments are doing in order that the funds may be raised for these enterprises. In other words, having sanctioned the enterprises, issued licenses, fixed the rates for interest and for power, guaranteed the efficiency and caused the cost or value to be filed with it, they must go further and find the funds to develop by first finding or arranging for the money to pay the interest, then issuing the securities to be sold to bankers, and they in turn to investors, on the strength of Government backing and guaranteed interest from the start.

Those who provide the money for interest, through taxes or assessments, must expect to get it back in some way, either through enhanced values of property, or improvements, lower rates, cheaper power, etc. The Commission will then have to regulate the use of funds during construction by the engineers and contractors, and lease the enterprises to operating companies for the term of years, specified in the Act.

William B. Jackson. The matter of water powers in their relation to the subject of this evening's paper has been referred to in previous discussions, but I believe this phase of the subject should receive more emphasis. The fact has been touched upon that the construction of electric transmission interconnections and net works over this country means the development of many water powers *which cannot otherwise be utilized*, but this means much more than the mere statement would indicate. We gain, of course, a reduction in the use of fuel on account of kilowatt-hours of energy produced by such water powers, but this is a small part of the advantage obtained by their utilization.

In almost all cases the inherent characteristics of a water power make it possible to reduce steam-plant capacity to a far greater amount than the kilowatt-hours of energy the water power may be able to develop would indicate, while at the same time giving opportunity for economically developing hydroelectric energy to the fullest practicable extent. The full developed capacity in water-power equipment is always available, and it can be operated at full capacity for short intervals, even with low stream flow and relatively small water storage capacity. It can therefore always care for peak load to its full installed capacity, which consequently need not be provided in the steam-plant capacity.

A third and very important advantage derived from the development of water powers which arises from the situation explained above, is the possibility for more economical per kilowatt-hour operation of the steam plants, both in the matter of fuel and labor on account of the greater amount of base power permitted from steam plant operation. There is a still further advantage, which applies more particularly in less densely settled districts, in the possibility of reduction in transmission line investment by the development of water powers located along the lines, many of which would be otherwise substantially worthless.

These can be so operated as to reduce the distances over which the heavy loads need to be carried by the transmission lines, and to therefore reduce the transmission-line capacity necessary.

My thought in the above remarks is to bring categorically before you the diverse factors which make water powers so important in unified electric systems beyond the matter of mere saving of a few kilowatt-hours of steam-station output. The more thought given the matter the more clearly the advantages of developing our water powers all over the country become impressed upon you as being a matter that should be encouraged as far as possible, and well-planned unified electric systems are one of the greatest factors in making possible such developments.

No. 1772

EFFECT OF LOAD FACTOR ON STEAM-STATION COSTS

By Peter Junkersfeld, Boston, Mass.
Member of the Society

The following paper discusses the factors affecting steam-station costs, particularly the load factor. Financial loss, the author states, will result whenever it becomes necessary to operate a power station at a substantially higher or at a substantially lower annual load factor than that for which the station was properly designed. The load factor should be carefully considered in locating a central station as well as in the selection of equipment. Fuel, usually an item of great expense, is dependent upon the load factor, the efficiency in the use of fuel being greater at the higher load factors. Curves are given to show relative generating costs and boiler rating as affected by load factor, typical week-day load curves and load factors, and the load factor as it occurs from day to day and month to month.

THE annual load factor is an indication of the number of hours per year that the dollar of capital expenditure in a steam station can be kept at work. The expense of using this dollar is just as great when it is working 100 hr. as when working the full 8760 hr. of the year. The returns are, however, in direct proportion to the number of hours per year that this dollar is kept at work, or in other words, to the annual load factor. For example, if the annual load factor of a power station is increased from 30 per cent to 60 per cent the fixed charges per kilowatt-hour are cut in half.

2 Load factor is the ratio of the average to the maximum output during any definite period of time. It is usually based upon an average demand for, say, 15 min., and upon output for a day, week, month, or year. It may apply to a system having many generating stations and serving many communities, or merely to one power station or even one unit in a power station.

3 The annual load factor of a central-station system nearly always changes very slowly. Special effort to secure a well-diversified business together with the new uses of electricity which are con-

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stantly arising contribute substantially to improvement in load factor.

4 The ordinary factory, working a 44-hr. week with the usual layoffs for holidays, keeps a power station busy for only one-quarter of the 8760 hr. per year. Residence and street lighting tend to improve load factor in the summer, but are less and less of a help as the days grow shorter, until they finally change from a blessing to a burden upon the plant, during the months when the lighting peak overlaps the end of the factory or business day.

INVESTMENT AS AFFECTED BY LOAD FACTOR

5 The cost of energy is made up of two principal elements, the fixed charges and the operating and maintenance expenses. In the case of the fixed charges we must use annual load factor because fixed charges run on year after year. Suppose that one power station has cost $100 per kilowatt. At 15 per cent per annum the fixed charge per kilowatt would be $15 each year. With an annual load factor of 30 per cent, the fixed charge per kilowatt-hour would be 0.57 cents. If the load factor is increased to 70 per cent, we find this fixed charge per kilowatt-hour cut to 0.24 cents.

6 This comparison of course is not wholly a fair one, because a station properly designed to operate at a 70 per cent load factor would cost more to build, although less per kilowatt-hour to operate, than one of equal capacity but properly designed for a 30 per cent load factor. Its boilers would have been rated differently, turbines of different individual capacities and perhaps different efficiencies might have been selected, the condensers would have been designed more liberally and higher-efficiency equipment would have been selected wherever economical with the higher load factor. Even supposing that those features boosted the first cost $20 per kilowatt, we would still save probably one-tenth of a cent per kilowatt-hour in the combined fixed charges and operating expenses in comparison with an attempt to impose a 70 per cent load factor on a plant properly designed for one of 30 per cent.

7 The curves in Fig. 1 show the relative costs of energy from a power station designed for a 70 per cent annual load factor when operated at various other load factors. Note at 40 per cent annual load factor the lower cost of energy obtained from a station designed for a 40 per cent annual load factor. The lower fixed charges on the 40 per cent station more than offset its higher operating expenses.

8 Financial loss will result whenever it becomes necessary to operate a power station at a substantially higher or at a substantially lower annual load factor than the annual load factor for which the station was properly designed. Suppose that the station mentioned previously were one of several stations supplying a light and power system. Assume that at the time of its construction it were intended to operate it on a base load at a 70 per cent annual load

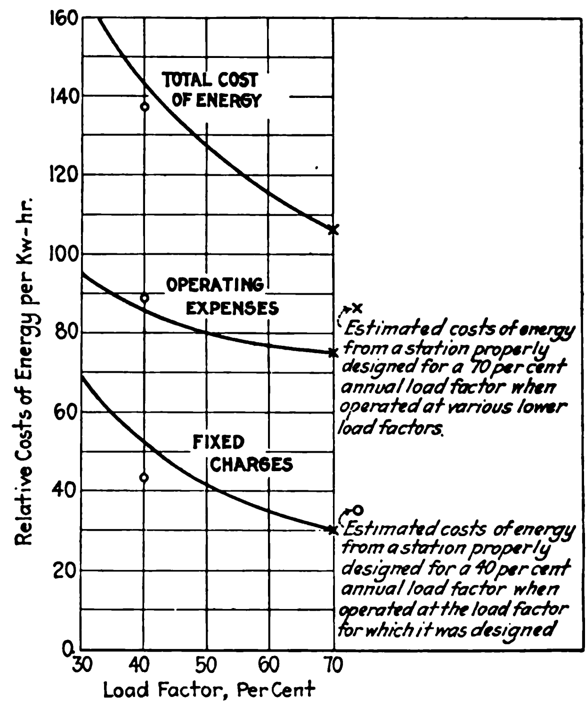

Fig. 1 Relative Generating Costs, Showing the Effect of Designing a Station for Too High a Load Factor

factor, whereas the other stations supplying the system were to be operated only to carry peak loads on account of their inferior efficiency. Assume also that within a few years, on account of rapid growth of load, another larger station, not previously contemplated, were built having greater efficiency and that the station first mentioned no longer were operated on the base load but were operated at a 30 per cent annual load factor. We at once see that we are then using an unnecessarily expensive tool where one costing $20 less per kilowatt would do the work just as well and at less total cost per kilowatt-hour.

9 Similar financial losses may occur in a single power station supplying a rapidly growing load. When a new unit is installed in a power station it is important to know approximately how soon another will be required. Each succeeding unit is frequently larger than the preceding ones, up to the practical limit of size available for the particular purpose. The increased size is usually accompanied by increased efficiency. The units of highest efficiency carry the bulk of the load. The extra price paid for extra efficiency in a power-station unit sometimes results uneconomically, due to another larger unit being installed earlier than anticipated.

10 The cost of an error of judgment, or the extra expense due to an unexpected change in the annual load factor at which a station or unit is operated may therefore be considerable.

CENTRAL-STATION LOCATION

11 The location of a central station relative to its raw-material supply and to the point of delivery of its finished product should be influenced greatly by load factor. In the selection of a site, condensing water is usually given first consideration. Transporting it adds greatly to the station fixed charges, while cooling it artificially causes inefficient operation. A very low annual load factor, likely to prevail over a long period of years, sometimes justifies artificial cooling of condensing water if the station may thereby be located near the load with a minimum investment in transmission lines or cable. With a large annual load factor the additional operating expense due to cooling the condensing water might be so large as to entirely outweigh the fixed charge on the transmission line; so that the plant would be located where condensing water is available. In weighing the relative costs, the operating expense at either location must be added to the fixed charge before the more advantageous site can be determined.

SELECTION OF EQUIPMENT

12 Load factor determines the quantity of fuel consumed by a power station and has considerable influence upon the efficiency with which it is burned. Starting, stopping and standing by as well as operating equipment only partly loaded, as is required at low daily load factors, inevitably result in a high coal rate.

13 One of the first problems, if the station be a new one, is to determine the boiler pressure. High efficiencies can be obtained with high boiler pressure. Construction cost, on the other hand, increases rapidly with increases in pressure. The economical pressure should usually be determined by the extent of the use of the equipment, that is, by the annual load factor.

14 An arbitrary assumption of steam pressure or an arbitrary selection of certain principal items of equipment may so definitely restrict the design that the full financial benefits of a consideration of load factor in the later selection of additional equipment may not be realized. The most far-reaching power-station engineering is done during the conception of the project, and it is at this time, when fundamental decisions are being made which will either help or hinder the engineering construction and operation of the station, that load factor should be given the greatest consideration. This means not only the immediate load factors, but also the load factors that could reasonably be expected during the probable life of the equipment.

15 Reference has been previously made to the differences in boiler rating for which central-station boiler plants would be designed for operation at different annual load factors. This depends upon the cost of the boilers and combustion equipment, their efficiencies at various loads, the cost of labor and maintenance at various loads, and the fuel required for banking and starting. Fig. 2 shows for a particular case, selected as an example, the effect of load factor on the most economical rating at which boilers should be operated to carry the station peak load. Note the manner in which this economical rating increases as the load factor diminishes.

16 The rating is only one of a number of boiler-plant problems that arise which must be solved on the basis of load factor. The type and design of the boiler, the length of its tubes, the number of rows of tubes wide and the number of rows high, the combustion equipment, the location of the bridge wall and the baffles, and the size and shape of the furnace, should all be determined by reference to the total load factor on the station as well as the anticipated load factor on the individual boilers. The use of economizers even in large stations cannot be justified at very low load factors.

17 The economic selection of turbo-generator units may be made upon the basis of size, speed and other factors. The speed influences both the cost and the efficiency of the unit, and is very important in relation to the average annual load factor at which the machine will opreate over a period of years.

18 The selection of a surface condenser should be made largely from the point of view of the annual load factor. A high load factor anticipated for a unit for a long period of time would justify a condenser of large surface, so as to obtain a high efficiency in operation.

19 The foregoing does not attempt to mention the many power-station appurtenances contributing to higher efficiency which

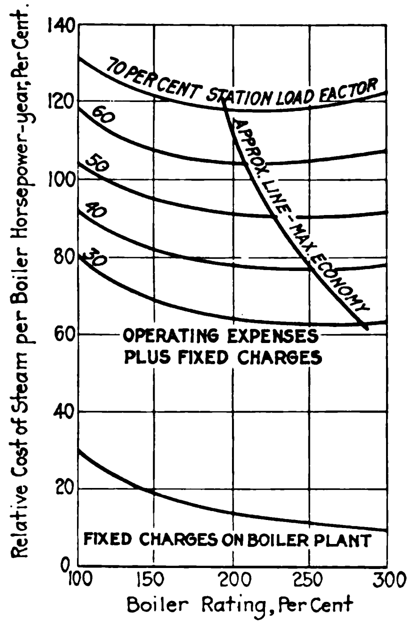

FIG. 2 BOILER RATING AS AFFECTED BY ANNUAL LOAD FACTOR ON THE STATION

should be omitted from the station designed to operate at low annual load factor, but which might be justified if the station annual load factor were higher. On the other hand, it should not be overlooked that there are some instances in which it is more economical to buy more expensive equipment for low load-factor conditions than for higher load-factor conditions.

20 The engineers who design a power station fix the upper limit of its operating efficiency by the selection and arrangement of equipment. If this is properly done for the particular load and load factor, the total cost of power from that station, including the fixed charges upon the investment, will be a minimum if the station is properly operated and maintained. If the selection and arrange-

ment are not made wisely, the station thereafter will be handicapped either by excessive fixed charges or excessive operating expenses.

OPERATING EXPENSES

21 The expense of superintendence and station administration is more or less fixed for a given size of station, so that a change of load factor would not influence it in total amount. Boiler labor may be considered as composed of two parts, one of which depends upon the capacity of the station alone, and the other of which, for a station of given capacity, would vary with the daily load factor upon the station. Mechanical labor would be influenced by daily load factor to a less degree, but would depend upon the number and size of the main generator units. The electrical labor would be influenced more by the number of feeders or the number of kinds of current distributed than anything else.

22 Fuel is usually an item of greater expense than all others and, as pointed out previously, it is dependent upon load factor. Efficiency in the use of fuel is greater at the higher load factors. As an example of the effect of load factor upon station operating expenses, a hypothetical case may be taken of a station containing units of various sizes which might have operating expenses as shown in Table 1. Note the fuel expense as compared with the total.

TABLE 1 OPERATING EXPENSES AS AFFECTED BY LOAD FACTOR

	Operating Expenses, Cents per kw-hr.		
Annual Load Factor, per cent	30	50	70
Fuel	0.91	0.85	0.71
Labor for Operation	0.10	0.07	0.06
Maintenance. Material and Labor	0.08	0.05	0.04
Miscellaneous, Material and Labor	0.04	0.03	0.03
Total	$1.13	$1.00	$0.84

23 Load factor alone does not by any means determine the amount of fuel which a central station of a given size should consume. The shape of the daily load curve is of almost equal importance in its effect upon the efficiency with which the fuel is burned. The shape of the load curve is in turn dependent upon the character of the business served by the central station. This is illustrated by the two examples of central-station load curve given in Fig. 3. In both instances the lighting peak overlaps the day load, but on ac-

count of the predominance of factory load in the business of one of these stations an exceptionally favorable shape of load curve is obtained as well as a relatively high load factor for the day.

24 It is interesting to note, however, that the load factor for the year is practically the same in these two cases. This is due to

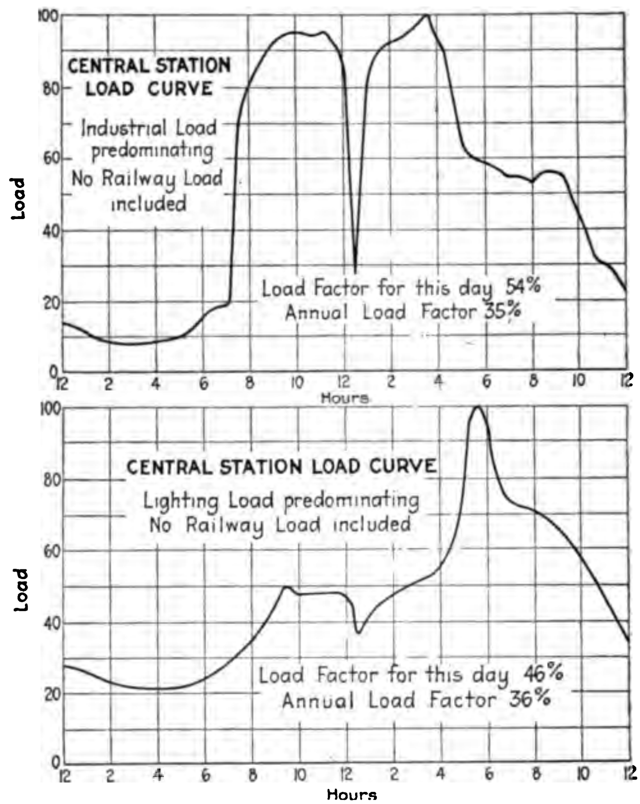

FIG. 3 TYPICAL WEEK-DAY CURVES AND LOAD FACTORS

the effect of the extremely low loads which obtain on Sundays and holidays. For these two conditions, illustrated by Fig. 3, having practically the same annual load factors, the better annual coal rate would be obtained with the one having the more favorable shape of load curve, that is, with the one in which the factory load predominates.

25 Fig. 4 shows how the load and load factor vary from day to day and month to month for a business in which factory load

predominates. The beneficial effect of high load factor on Thanksgiving Day, for example, is much more than offset by the relatively small load on the station on that day. Note how the load factors for the week and for the month are practically the same. The load fac-

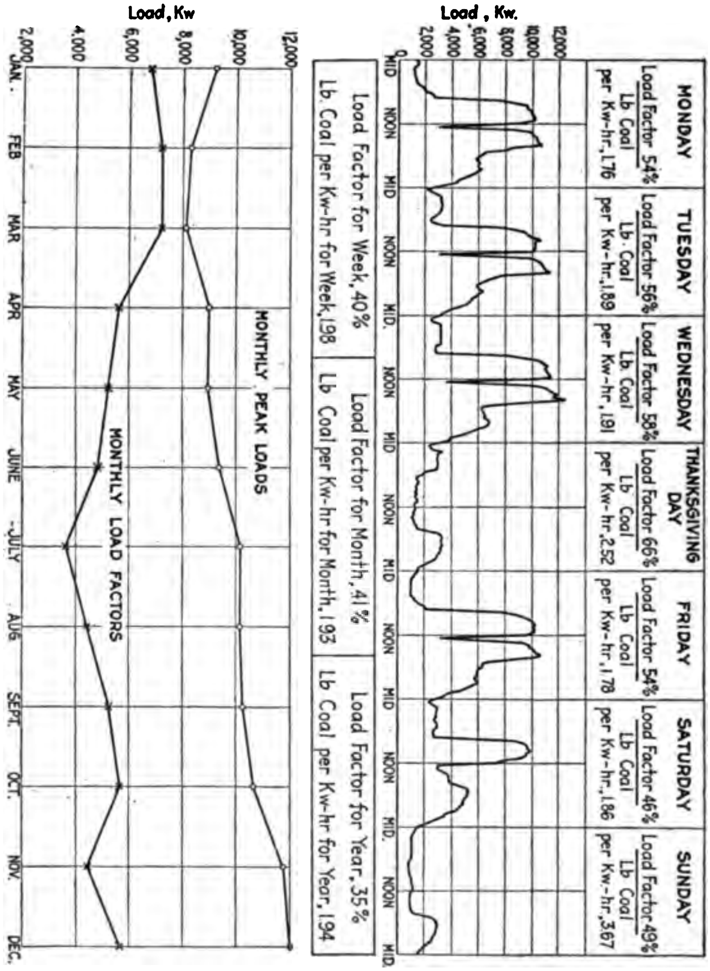

FIG. 4 LOAD FACTOR AS IT OCCURS FROM DAY TO DAY AND MONTH TO MONTH

tor for the year is somewhat lower, due to the variation in maximum demand throughout the various seasons. In general the better coal rates are obtained with the higher load factors, although this is greatly influenced by peak load, quality of fuel, and skill in operation, all of which are interesting subjects but beyond the scope of this discussion.

DISCUSSION

CHARLES PENROSE stated that it is perhaps the foremost duty of the engineer to act as conserver of the natural resources of the nation, and by the same token to obtain the greatest possible economy in power-station design. If this is true, he said, then, instead of designing a power plant for existing loads, it should be designed for high loads and then the load should be developed. In some cases this can be secured by a combination of circumstances and loads, including railroad electrification, street-railway load and industrial uses. There has been secured, he said, within a short radius from New York City a load factor which at the present time, exclusive of Saturday, Sunday and holidays, has the high figure of 78 per cent.

JAMES T. ENES referred to the relation between the investment and load factor, which in turn depends upon fixed charges and operating expenses. Since we are now entering a new era of increasing prices, he said, it will be very difficult and perhaps impossible to determine such charges. We have in mind prices and the actual value based on the entire life of the plant. The problem is thus complicated because of the fact that we are now entering upon a period of increasing prices.

No. 1773

POWER APPLICATIONS TO COTTON-FINISHING PLANTS

By Leo Loeb, Philadelphia, Pa.
Junior Member of the Society

The following paper presents an analysis of the power applications to cotton-finishing plants in which, under ordinary conditions. the steam required for processing is greater than that for development. The generation of power and of steam and the applications to the driving of process machinery are traced briefly from the boilers to the motor applications. Charts are presented showing the steam consumption and the proportion of the exhaust steam converted into work and available at the exhaust for different types of prime movers. Illustrations of typical drives are included.

POWER applications in the textile industry fall into two main subdivisions: (*a*) Those primary processes, such as spinning and weaving, in which mechanical power is used to a great extent and in which steam for processing and for the heating of manufacturing space is relatively less important; and (*b*) The finishing processes, including bleaching, dyeing, mercerizing, printing, calendering, etc., where process steam and the steam used for the heating of manufacturing space may be from two to four times the steam required for power.

2 This paper will deal with power applications to the finishing of cloth, in which the solution of a proper heat balance involves more factors than a primary power problem.

3 Power surveys at a number of finishing plants have established the fact that at all seasons of the year the demand for process steam will be several times the demand for power steam. Just what relation exists between these steam requirements will be affected by such factors as:

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a Type and economy of prime movers
b Kind of cloth finished and nature of finishing process
c Efficiency of the process machinery
d Mean outside temperature
e Skill and competency of master mechanic's department
f Strictly local conditions.

4 Under the latter are included such items as availability of water and hydroelectric power, either in the form of indentured water rights accompanying land grants, or cheap hydroelectric development by a utility company.

5 Opinion as to the lowest working steam pressure for process work consistent with steady output and a minimum production of "seconds," is almost as varied as the number of plants. It has been found, however, that with ample heat transfer surface, pressures of from 10 to 15 lb. gage, with an average of 12 lb., will take care of fully 80 per cent of the requirements in finishing.

6 It therefore follows that all power may be generated as a by-product by utilizing engines or turbines as reducing valves. Steam at boiler pressure is thus reduced to the low pressure of the exhaust line without thermal loss other than the heat converted into work and, in case only of reciprocating engines, the loss in cylinder condensation.

7 From the economic standpoint power cost will be merely the fuel equivalent of conversion and condensation, labor and material costs in attendance upon and maintenance of equipment, and fixed charges on power-plant machinery. Under these circumstances purchased power can seldom compete with self-generation, provided the installation be large enough to keep the overheads and operating labor costs to a low figure per kilowatt-hour of output.

BOILERS

8 For obvious reasons water-tube boilers are coming into more general use in the boiler houses of finishing plants. The same points of superiority over horizontal or vertical fire-tube boilers which manifest themselves in central station use apply in the selection of boilers for a finishing plant. The working steam pressure should range from 150 to 200 lb., the higher figure being preferable where its use does not add unduly to the first cost of boilers. A moderate degree of superheat is desirable, although not essential. Superheat reduces condensation in steam lines and results in somewhat better

economy of engines or turbines, but there may be objections on account of the superheaters and their maintenance.

9 Among the items of minor equipment which are becoming increasingly popular may be mentioned soot blowers and reliable feedwater regulators.

FUEL AND FIRING DEVICES

10 The kind of fuel used is intimately associated with its method of burning. The use of steam sizes of anthracite, except where mixed with bituminous coal, is not increasing. The market is restricted, and few types of stokers for this fuel have been developed. High boiler ratings easily reached with bituminous coal cannot be duplicated with anthracite; hence the installed boiler capacity and the first cost of boiler plant must be greater than with soft coal. Increasing fuel costs, higher freight rates, and labor advance all work under normal market conditions toward the reduction in consumption of the lower grades of anthracite, particularly at points remote from the coal fields.

11 As in all other industries bituminous coal leads in the proportion of fuel utilized by finishing plants. The newer installations for soft coal have underfeed stokers, of which there are now several reliable types on the market. Where the cost of large brick or steel chimneys is prohibitive, induced-draft fans may be installed, and this apparatus is always provided with economizer installations.

12 Fuel oil is the ideal fuel, but high price and possibility of shortage during times of emergency have combined to restrict its use. There are many advantages and economies in its use, and very recently there has been evidence that the large oil producers are entering the field to secure new business. Quick steaming, cleanliness and reduction in labor are the foremost economies, but no large plant should go to oil fuel without providing means for quickly converting a portion of its capacity to coal burning in case of acute and prolonged shortage of oil.

13 The use of powdered coal as a boiler fuel is just beginning to emerge from the experimental stage. Under favorable conditions the efficiency of combustion will be high, but offsetting this are such points as: liability to dust explosion; cost of drying and grinding; nuisance from fine ash which will be deposited on adjacent property, and high brickwork and refractory maintenance.

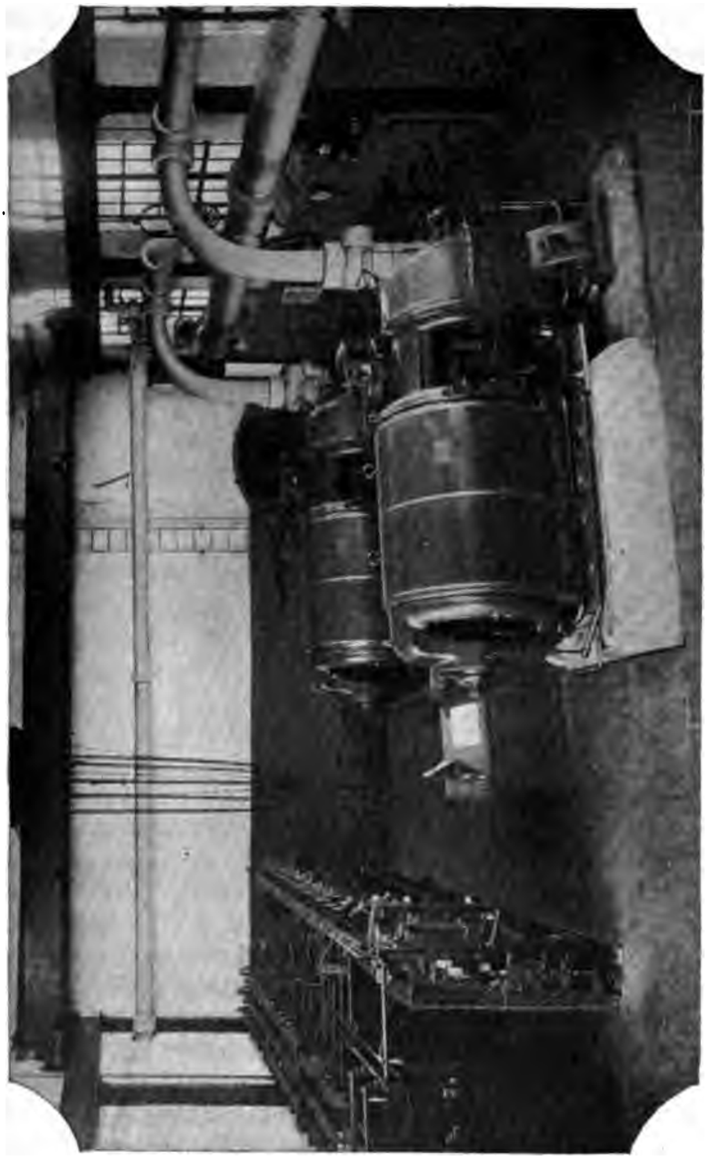

FIG. 1 INTERIOR OF POWER STATION FOR A FINISHING PLANT. TWO 1250-KW. TURBO-GENERATORS AND AUXILIARIES

PRIME MOVERS

14 The selection of primary drive is a matter which will depend to an extent on the size of the plant and the schedule of operation. In certain districts water power is still available during a considerable portion of the year, and this power is usually utilized by the mills as a part of privileges and grants dating back to the original acquisition of the mill property.

15 The water-power companies will likewise sell surplus water power under contract conditions, but the charges for such excess power will likely be in excess of rates for purchased power. This is due principally to the fact that individual mills have small wheels of low efficiency and the power company, where itself engaged in the hydroelectric business, can develop a horsepower hour through its larger and more efficient machinery on less water than the individual mill. Likewise conditions of load factor and diversity, together with control of storage water, generally put the power company in a more favorable position to utilize the water than to sell it.

16 Where water-power privileges have been acquired in terms of horsepower without time limit as to use, the individual mill can generally afford to continue any existing development up to its original privileges, particularly if some departments, requiring power in excess of process steam, are likely at times to operate when the rest of the plant is idle. Water-power privileges form a convenient means of developing power for night lighting and for the overtime operation of mechanical departments.

17 Naturally the available water power is subject to considerable seasonable variation, and a plant desiring continuity of operation must of necessity have sufficient steam-power stand-by to take care of its requirements under low-water conditions. The chances for duplication in investment are therefore obvious, so that the development of power from water should in every case be subject to careful analysis.

18 Steam power in the smaller mills may be developed in economical engines, but for plants requiring any considerable amounts of power the turbine drive from all points of view is preferable.

19 The turbine will extract from the steam only such energy as is converted into work and a small amount of casing radiation, retaining the remainder of the latent and sensible heat in the exhaust steam.

20 There is a field of application for the bleeder type of turbine in which steam may be extracted from an intermediate stage for processing or for building heating requirements. The excess steam to make up power demand passes to a condenser. The latent heat in this condensed steam may in part be recovered by utilizing circulating water for processing and for make-up boiler feedwater.

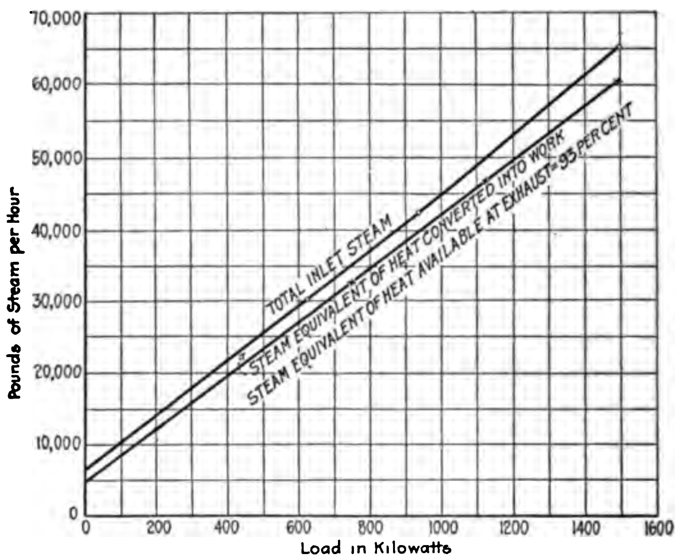

FIG. 2 STEAM CONSUMPTION AND AVAILABLE EXHAUST FROM A 1500-KW. NON-CONDENSING TURBO-GENERATOR

Steam pressure at throttle, 175 lb. per sq. in.; back pressure at exhaust, 12 lb. per sq. in.; steam flow at full load per hour, 65,000 lb.; equivalent steam available at exhaust, per hour, 60,600 lb.; inlet steam available at exhaust, 93 per cent.

There will, however, be at all times an unavoidable loss of heat rejected at condenser outlet.

21 Fig. 1 shows the interior of a power station for a finishing plant. Figs. 2 to 5 show for prime movers of four different types the relative steam consumption per unit of output and the relative heat consumption under conditions where exhaust steam can be fully utilized at all seasons of the year. Assuming in each case that the primary drive is connected to an electric generator, the relative costs based on a 1500-kw. size, will be approximately as shown in Fig. 6.

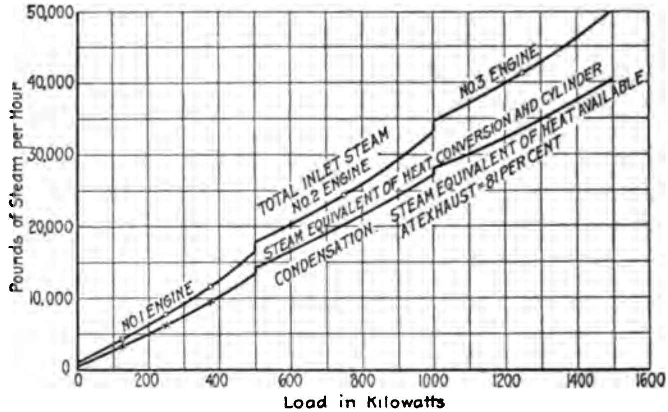

Fig. 3 Steam Consumption and Available Exhaust from Three 500-kw. Non-Condensing Engine Generators

Steam pressure at throttle, 175 lb. per sq. in.; back pressure at exhaust, 12 lb. per sq. in.; steam flow at full load, per hour, 50,000 lb.; equivalent steam available at exhaust per hour, 40,600 lb.; inlet steam available at exhaust, 81 per cent.

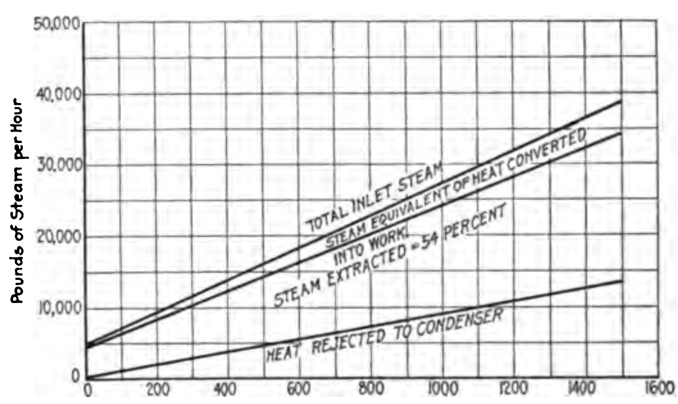

Fig. 4 Steam Consumption and Available Exhaust from 1500-kw. Extraction Turbo-Generator

Steam pressure at throttle, 175 lb. per sq. in.; extraction pressure, 12 lb. per sq. in.; vacuum in condenser, 28 in.; steam flow at full load, per hour, 39,000 lb.; steam extracted at full load, per hour, 27,000 lb.; inlet steam extracted, 69 per cent.

ELECTRIC DRIVE

22 Except in smaller mills where belt drive may be used in connection with a steam engine, the tendency is now entirely toward electrification, both by groups and by individual machines.

23 The electrical characteristics of generators and motors must be determined after a study of the local requirements for variable-speed as against constant-speed drives. In certain instances the selection will be influenced by the amount of electrical equipment

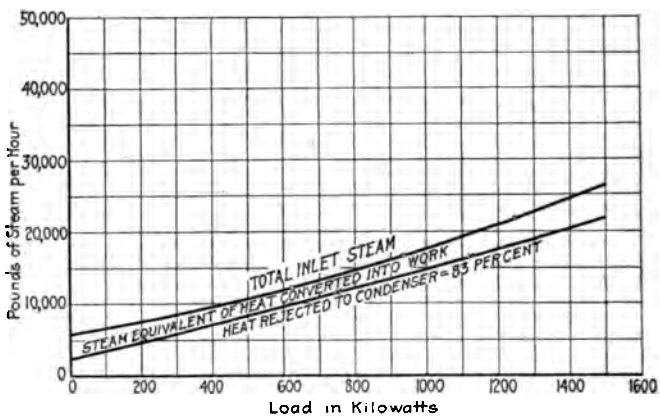

FIG. 5 STEAM CONSUMPTION OF A 1500-KW. CONDENSING TURBO-GENERATOR

Steam pressure at throttle, 175 lb. per sq. in.; vacuum at exhaust, 28 in.; steam flow at full load per hour, 26,400 lb.; inlet steam rejected in circulating water, 83 per cent.

already installed in the mill and the power characteristics of the local public utility on whom it may be desired to depend for stand-by service.

24 The three main subdivisions of power applications to finishing plants are:

a Such power as is required to drive groups of machines or individual process machinery at constant speed. This applies to bleaching, washing, short tentering, damping, jig dyeing, pasting; also to ventilation, color mixing and pumping

b Such power as is required to drive process machinery with moderate speed variation not in excess of 2 to 1 to accommodate variation in fabric and process. This includes certain forms of mangles and cans, sheet ranges, starch mangles, mercerizers, khaki, sulphur and indigo dyeing machines and steamers

c Such power as is required to drive process machinery where the demands for variation in speed are in excess of 2 to 1. This class of equipment includes long tenters, aging boxes and multi-color printing machines.

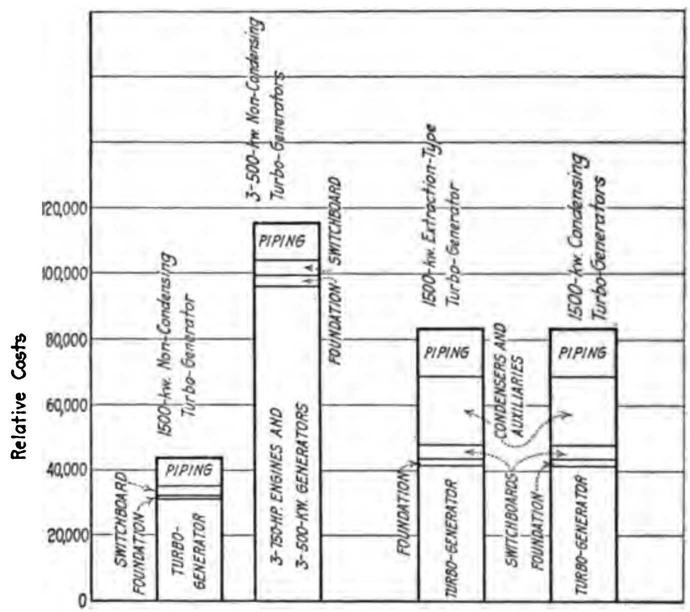

FIG. 6 RELATIVE INSTALLATION COSTS OF FOUR DIFFERENT TYPES OF PRIME MOVERS

25 For variations in speed in excess of 2 to 1, the direct-current motor occupies the leading position, unless it be desired to resort to some form of mechanical speed-changing device.

26 In the class of finishing plants considered, the operations of tentering and printing will frequently require speed ratios outside the range of commercial forms of alternating-current motors. Hence the proportion of power consumed by motors driving tenters and printing machines to the entire mill requirements, is an important factor in deciding whether the entire electrification shall be direct current, or whether a combination of alternating and direct current be applied. A suitable direct-current voltage is from 230 to 250 volts.

27 In the larger mill projects the electrical characteristics of generators are generally 3-phase, 60-cycle, with voltages varying according to the choice of the designer and his method of solving the power problem.

28 The voltages in use are 220, 440, 550 and 2200, the two intermediates being the most popular. The lowest voltage calls for large sizes of conductors and conduits, and the highest cannot be used directly on all mill motors, hence transformers for stepping down to motor voltage must be installed. In such a development direct current may be obtained from rotary converters or motor-generator sets. Although the rotary converter has a higher efficiency and may occupy less floor space than the motor-generator set, the use of the latter is preferable because of its ability in effecting power-factor correction.

29 Modern alternating-current generators are designed on a basis of maximum current output, hence in such finishing plants as have a large proportion of inductive motor load, it may be well to specify a load power factor of 70 per cent or 75 per cent when selecting generators, rather than the more usual 80 per cent used in central-station practice.

30 There are two methods of providing power for night lighting and the overtime operation of mechanical departments. One is the installation of a small turbo-generator of sufficient capacity to take care of lighting and shop load. This machine will have exactly the same electrical characteristics as the main units, but will have a lower water rate for the off-peak load.

31 Another method successfully applied is to provide some excess capacity in the steam exciter and by means of a selector switch transfer the lighting load from the normal supply of three-wire 100-volt alternating current to two-wire 110-volt direct current. Likewise the shops may be equipped with a 110-volt direct-current motor taking its power from the turbo-exciter and driving the shops when the manufacturing departments are idle.

32 Where the size of the equipment justifies the expense, generator coolers should be installed in connection with high-speed alternating-current turbo-generators. These generator air washers not only permit of high capacity of the generators during extremely warm weather, but likewise remove coal and ash dust, lint and any other foreign matter whose presence might damage the generator windings.

MOTOR APPLICATION

33 Constant-speed groups or individual drives are best operated by induction motors with the type of drive illustrated in Fig. 7. Such motors are controlled by motor-starting switches mounted conveniently to the operator and protected against injury by overload and low-voltage trips.

34 For constant-speed drives having heavy starting duty it may be necessary to use slip-ring motors with drum controllers and

Fig. 7 Installation of Squirrel-Cage Motor for Constant-Speed Drive

with resistances in the controller circuit for starting duty only. A safety or oil switch should be provided in the primary circuit.

35 Another group of drives falls in the class where the requirements of the process are met by a speed reduction of two to one, or where the lowest speed needed at any time will be 50 per cent of the maximum speed. It is not always necessary under these conditions to resort to the direct-current motor, since a slip-ring motor with secondary resistance banks designed for continuous running at any point of the controller has been found suitable for mangles, cans, open tenters and sheet ranges.

36 The control equipment consists of a drum controller and resistance for the secondary circuit. The incoming feeder should be

protected by an oil switch with overload and low-voltage releases. A variable-speed alternating-current motor application is shown in Fig. 8.

37 It should be borne in mind in applying slip-ring motors for variable speed that, except for the limitation of maximum speed imposed by the frequency of the alternating current, a slip-ring motor has very similar speed and electrical characteristics to the

FIG. 8 INSTALLATION OF SLIP-RING VARIABLE-SPEED A.C. MOTOR

series type of direct-current motor. Consequently variations in load arising from such adjustments as the set on rolls or the friction of cloth over cans and rollers, will produce variations in speed without any change of controller position.

38 The applications of variable speed with higher ratios by means of direct-current motors opens up a wide field in the selection of control equipments. Various manufacturers have developed outfits to suit local conditions and the ideas of the designer. These controls vary all the way from the simple faceplate rheostat for

varying the field strength of a shunt-wound direct-current motor to the more elaborate push-button control contactor panels such as are illustrated in Figs. 9 and 10.

39 A form of safety device which should never be omitted is some form of push-button or snap-switch release whereby the back

FIG. 9 CONTROL FOR VARIABLE-SPEED D.C. MOTOR

tender on printing machines or tenters may from his position instantly shut down the entire equipment.

40 It is usually convenient to use two motors for driving long tenters, one driving the tenter equipment proper and the other operating the mangle and cans. In order to preserve proper synchronism between the two motors, a compensating equipment such as is illustrated in Fig. 11 may be applied. This consists of a "dancer"

roll actuated by the tension in the cloth between dry cans and tenter. The vertical movement of the dancer roll operates a rheostat which in turn varies the speed of one of the motors.

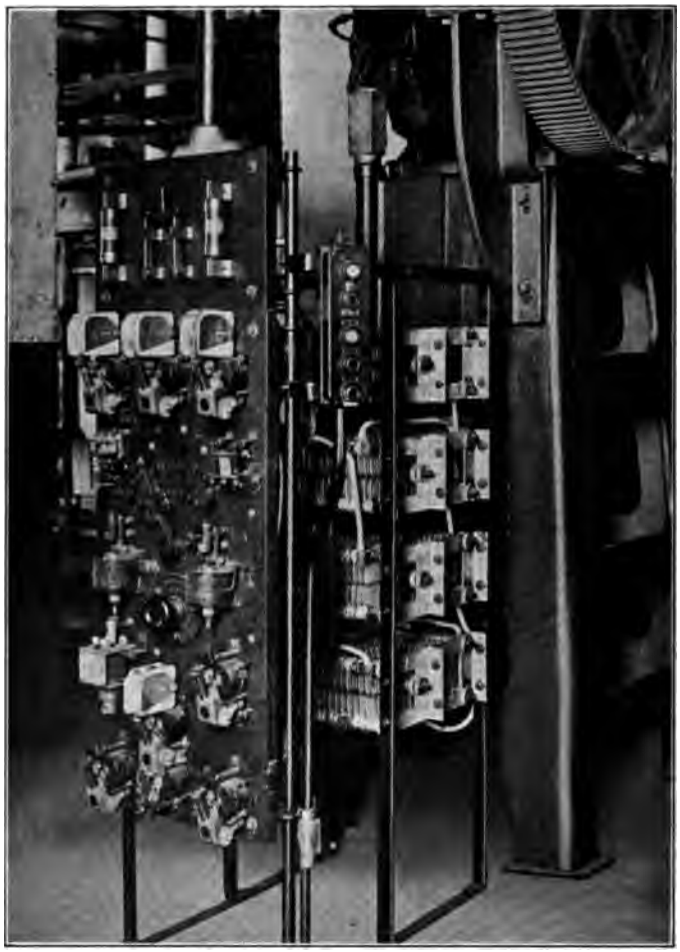

FIG. 10 CONTROL PANEL FOR VARIABLE-SPEED D.C. MOTOR, PUSH-BUTTON CONTACTOR TYPE

STEAM PIPING

41 There are no unusual difficulties in the design and installation of steam piping to take care of a proper heat balance in a finishing plant. Since under normal conditions process steam will be

considerably in excess, there is required merely a satisfactory form of reducing valve to automatically make up the deficiency at a point in the low-pressure main beyond the turbine exhaust.

42 In every mill there are certain processes which with the existing equipment function better on high-pressure steam, and for this purpose it may be necessary to have some high-pressure piping beyond the points where steam is bled into the exhaust main. This

Fig. 11 Installation of Compensating Roll for Synchronizing Two Variable-Speed D.C. Motors

steam may be used at full boiler pressure or reduced to 80 lb. or 100 lb. gage by additional reducing valves.

STEAM DISTRIBUTION AND BALANCE

43 Since in the raw materials utilized in finishing plants steam and power rank second only to gray goods and drugs, the necessity of measuring the power and steam consumption of all departments is obvious. This can be accomplished by the simple application of integrating wattmeters to power feeders and of flow meters to steam

lines. The number of electric and steam meters need not be excessive and their cost will soon be returned in the form of better knowledge on the part of the executive as to plant conditions, and in the ability to immediately locate and check wastes and losses. An

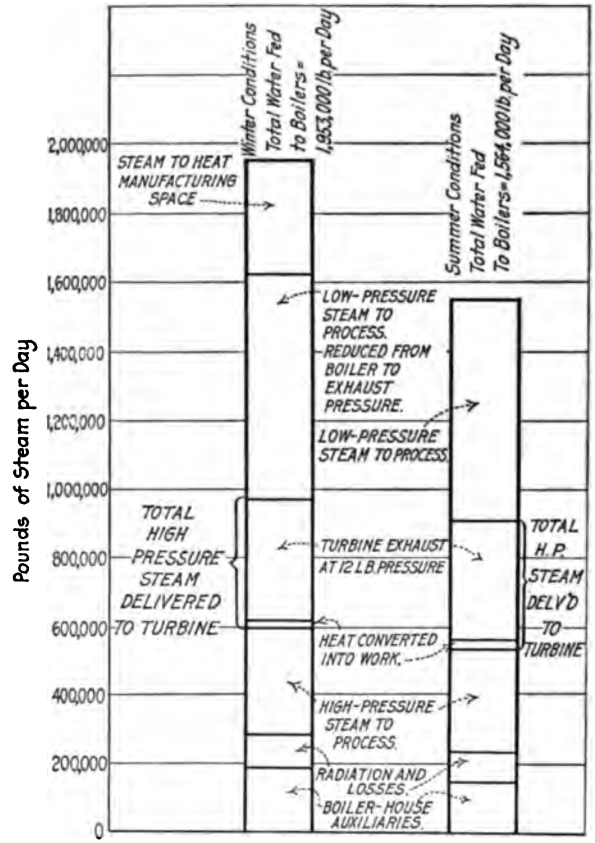

FIG. 12 STEAM DISTRIBUTION (HEATING AND NON-HEATING SEASONS) FOR FINISHING PLANT WITH CORRECT HEAT BALANCE

X: Total high-pressure steam delivered to turbine.

actual distribution from the point of view of sources of steam is shown in Fig. 12 for both the heating and non-heating seasons.

44 There may in some cases be a tendency toward thermal unbalance occurring during periods of low production in the non-heating season, or during periods of the working day when the

power load is building up more rapidly than the process steam consumption. Any loss of this kind may be checked readily by the installation of a storage heater, in which the heat-transfer surface is proportioned in accordance with the rate of discharge of exhaust steam, and the volumetric or storage capacity in accordance with the duration of the period of unbalance and the heat to be absorbed in that period. This hot water can be drawn upon for washing or wherever hot water is required in the process.

45 A general plan of power application based on a complete heat balance such as has been described in this paper will lead to very substantial economies in fuel, labor and maintenance costs in the power department over the more general use of engines for group or individual drive. Particularly at this time, because of advancing freight rates, scarcity of fuel, high prices prevailing and unsatisfactory transportation conditions, have fuel costs and power economies assumed national significance. Indeed, the Government has recognized the prime importance of this problem in its relation to the conservation of natural resources through the superpower survey being conducted by the Department of Interior.

46 It therefore behooves every mill executive to weigh the power problems of his plant with the same rigorous scientific standards as have, as a matter of course, come to be applied to the more strictly manufacturing phases of the finishing industry.

DISCUSSION

F. R. Low. The writer is glad to see this paper call attention again to the possibilities of heating and processing for maintaining a lower temperature level to which steam generated at higher pressures and temperatures may be worked with the inexpensive production of power.

Thirty years ago, Chas. H. Manning told the Society that by this means he was developing 1200 hp. and the men in the fire room could not tell whether the engine was running or not, there was so little difference in the consumption of steam whether it was sent direct to the dye house or through the engine.

The writer recently visited a plant at which 50 tons of coal, are burned per day to generate steam at 100 lb. pressure which is used at low pressure for heating and manufacturing processes, without any attempt to utilize the energy which might be generated by the heat fall, notwithstanding more than this amount of power is purchased. The simplest kind of turbine might be used as a reducing valve, and whatever power is produced, and it would be a great deal, would be almost pure gain.

Charles H. Bigelow said that his experience had been that the bleeder turbine worked better than the non-condensing turbine for the reason that the demand for steam for the bleachery did not always coincide with the demand for power. In one plant of which he knew and in which was a non-condensing turbine, either the make-up valve was in operation or the turbine was exhausting to the atmosphere. The master mechanic of the plant thought that the load was steady, but when recording steam flow meters were installed, he saw that the requirements varied more than he had supposed.

Paul A. Merriam, in speaking of the variable demands for power and process steam, said that at certain times during the day the amount of steam required in a system was in excess of that produced, while at other times there would be exhaust steam coming out of the relief valve. He thought the real solution to be a large storage heater which would absorb the surplus exhaust, store up the heat which was otherwise going to waste, and make it available later as hot water. Storage heat of this kind, he said, could be likened to a flywheel on the heating plant, and would help maintain the heat balance during the day.

Relative to superheaters, he did not feel that they had any application in a finishing plant. In other words, the fact that the demand for exhaust steam was greater than the exhaust steam available made it seem inadvisable to decrease the water rate of the turbine. The money spent for superheaters would better be spent for economizers.

In connection with the choice of boilers and prime movers, he thought that attention ought to be brought to the matter of floor space. In two finishing plants with which he was familiar, the entire power plant, both boilers and turbines, had been placed in the space that had been occupied by the boilers alone. This had been done by taking out the tubular boilers and putting in water-tube boilers and turbines in part of the space. The space formerly occupied by the engines had been made available for other purposes.

He thought direct current more desirable than alternating for finishing plants because so many machines required careful control of speeds.

F. A. Chiffelle said that speed variation would influence the cost of production more than a variation in power consumption. The aim should be to run at the maximum speed of the machinery in use at all times and take care of variations by the amount of air or heat supplied to the machine, rather than to vary the speed of the machine to suit a predetermined amount of heat supplied. It had been his experience that a great deal of the speed variation ordinarily demanded could be eliminated, and by constant-speed operation the cost of production would probably be cut down much more than by varying the speed of the machine. Incidentally, much complicated machinery for varying the speed would be eliminated.

The Author. Mr. Bigelow refers to the use of the bleeder rather than the non-condensing turbine. The particular type to be chosen depends, of course, on the average relation of process to power steam. In most cases it will be found that the slight unbalance which is likely to occur on starting up in the morning and closing down at night is insufficient to warrant the extra expenditure in piping, turbine, condenser and auxiliaries; to say nothing of the maintenance on the equipment. However, this is a point that each engineer must decide for his layout.

Mr. Merriam's suggestion of a storage heater to absorb the fluctuating steam demands is excellent in the cases where the heater can be kept within reasonable size, and expensive supporting structure avoided. A slight amount of superheat will be of advantage in insuring dry steam at turbine inlet and thus reducing erosion of first-stage blading, which has in some cases resulted in costly reblading after a few years of service. Thermodynamically there is no gain of consequence, the maintenance item being the prime consideration.

Mr. Chiffelle's comments on speed variation apply when the plant output is more or less uniform. But in an establishment serving a diversified trade, the power finishing process will vary with the season of the year, changing fashion and the fabric offered by various customers, so that a wide speed variation may be required on some machines, in order that minimum production costs may obtain.

No. 1774

ORGANIZATION AND CONSTRUCTION OF DYE HOUSES

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The machinery organization, location, construction, ventilation and piping of dye houses are topics treated in the following paper in a general way from the engineer's viewpoint. The fundamental principles which are laid down can be applied to individual cases according to the conditions which surround them. Dye houses are of two kinds: the converting plant, which does dyeing and finishing for the trade; and the plant dye house, which dyes and finishes for one mill only. The operation of dyeing enters largely into the four great branches of the textile industry, cotton, worsted, woolen and silk. The author lists the machinery used in dye houses and describes the processes followed in cotton, worsted, and woolen dye houses. The location of the dye house is discussed in a very general way because it is largely influenced by the particular conditions of each house. Dye-house construction and dye-house ventilation are discussed more in detail. The paper concludes with a brief consideration of dye-house piping.

THE purpose of a dye house is to apply and fix colors to textile materials of all kinds, although the scope of any one plant is usually limited to one or two kinds of materials. The chemistry of dyeing is rather complicated and will not be touched upon. The purpose of this paper is to discuss from the engineer's viewpoint the machinery organization, location, construction, ventilation and piping of dye houses. These topics are treated in more or less of a general way and certain fundamental principles are laid down which can be applied to individual cases according to the conditions which surround them.

2 The average dye house of the past was a damp, gloomy room, with murky windows, a series of shallow puddles, a confusion of trucks and machines, water dripping from the ceiling and pipes, and clouds of vapor rising from the kettles. Quite often the view

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consisted of one vast cloud of white vapor with a few shadows scattered through it. Fortunately that condition is fast disappearing and it is now realized that a dye house will function better, give greater production, with less damage to stock in process, if it is a well-lighted, well-ventilated room, free from steam, with reasonably dry floors and ceilings, and be a desirable place for a man to work in with reference to his health and mental condition.

3 This change has been brought about partly by statutory requirements, partly by demands of the employees for better working conditions, but largely by the advance in engineering practice and the closer coöperation of the engineer and the mill owner. There has been at the same time a revolution in dyeing processes, both as to methods and materials, which has simplified many of the problems.

4 The equipment mentioned later may not be complete in every detail since there is more or less special machinery used for special purposes and processes. There is considerable individuality shown by dyers in their methods and a wide variation exists in the methods employed to obtain similar results. As a rule the machinery used by all dye houses to dye raw stock, yarn, and cloth is very much the same, but the method of handling and the process vary largely.

KINDS OF DYE HOUSES

5 There are two distinct classes of dye houses, one being the converting plant which does dyeing and finishing for the trade; and the other, the plant dye house which dyes and finishes for one mill only. The first generally covers a wider range and is prepared to do a great variety of work, while the latter is naturally of a narrower scope and is usually limited to dyeing the product of one plant. Converting plants are sometimes equipped to do bleaching and mercerizing, which add complications to their problems. For the purpose of this paper the dye-house problem will be discussed from the standpoint of the plant dye house.

6 The operation of dyeing enters largely into the four great branches of the textile industry, cotton, worsted, woolen, and silk, and in many cases these lap over each other in the individual plants because of the great variety of combinations of stock used in making cloth. Cotton may be dyed in the raw stock, as yarn, or in the piece. Worsteds are dyed in the top, as yarn, or in the piece. The bulk of the woolen fabrics are dyed in the stock, but yarn and piece dyeing are also common. Silk is generally dyed as yarn.

MACHINERY ORGANIZATION

7 Following is a list of the kinds of machinery used in cotton, worsted and woolen dye houses. It will be noted that many of the machines appear in more than one of the lists. It must be borne in mind that most dye houses handle more than one kind of stock and in more than one form and will contain machines from more than one group.

Cotton-Dyeing Machinery

For Yarn:

- Boiling-out machines
- Doubling machines
- Chain dyeing machines
- Splitting machines
- Beam dyeing machines

For Piece Goods:

- Jigs
- Continuous dyeing machines
- Padders

For Raw Stock:

- Pressure kettles
- Rotary raw-stock kettles
- Hand tubs

Miscellaneous Machinery

- Dye cans
- Mangles
- Washers

Worsted-Dyeing Machinery

For Yarn:

- Rotary skein-dyeing kettles
- Spool-dyeing kettles

For Piece Goods:

- Piece dye kettles

For Slub Dyeing:

- Top-dyeing machines

Other Machinery:

- Crabs
- Steamers
- Flat and string washers
- Extractors
- Carbonizing machines
- Dry cans
- Tentering dryers

Woolen-Dyeing Machinery

For Yarn:

- Rotary skein dyeing kettles
- Spool-dyeing kettles
- Skein dyeing in machines in which the liquor is circulated

For Piece Goods:

- Piece dye kettles

For Raw Stock:

- Rotary raw-stock kettles
- Stock dyeing in machines in which the dye liquor is circulated
- Hand tubs

Silk-Dyeing Machinery

For Yarn:

- Rotary kettle for skeins
- Hand tubs.

8 This list might be added to for special work, but practically all dye houses have a machinery organization made up of a combination of machines from these lists. It will be noted that all of these machines with the exception of the dry cans and carbonizing machine are so-called wet machines, and steam is used in all of them with the exception of the extractors, doublers and splitters.

PROCESSES

9 Of all dye houses, the one handling cotton only presents the fewest problems, due to the simple construction of the machines, the small amount of water and steam required, and the comparatively low temperatures at which the dye liquors are used.

10 *Cotton Dye House.* The cotton materials are delivered to the dye house as raw stock, as yarn in ball warps or beams, and as cloth in rolls. The machinery for this organization is given in the above list and requires no special description since it is quite common. In cotton dye houses the drying is usually done in the dye house proper, and for some classes of dyeing heat is required to develop the colors. Dry cans of various widths are used for this purpose, made up in both horizontal and vertical sets. Pits and trenches are required for many of these machines, but not to so large an extent as in worsted or woolen dye houses.

11 *Worsted Dye House.* A dye house for worsteds is somewhat complicated by the wet-finishing machinery required before and after piece dyeing and which should be so placed as to be convenient to the dyeing machinery. Considerable yarn dyeing is done either on skeins, in a rotary kettle, or on spools as in the Franklin process, but the bulk of the work is cloth, dyed in piece dye kettles.

12 Before dyeing the goods are crabbed, steamed, washed and sometimes carbonized. The machinery for these operations is usually in the dye house, but partitioned off from the kettle room. All this machinery requires water and steam in large quantities and calls for a building of somewhat the same construction as a dye house. The kettle rooms contain all the piece, yarn, and top-dyeing kettles and the extractors or squeezers. The drying equipment consisting of dry cans and tenter frames should be conveniently located.

13 *Woolen Dye House.* In woolen mills the bulk of the stock dyed is scoured wool and noils, although there is more or less piece dyeing and a fair amount of yarn dyeing. The piece and yarn kettles are the same as used in a worsted plant. The only other machinery that should be in the kettle room of a woolen dye house is the extractor.

14 Much more might be said about the machinery used in a dye house, but it would consist largely of the description of machines which are well known. It might be said, however, that as a whole the machinery is exceedingly simple but must be well built to resist the hard usage to which it is subjected by the use of steam, acid, and water. It must be thoroughly reliable because of the damage which can be caused by a breakdown while stock is being dyed.

LOCATION OF DYE HOUSE

15 The location of the dye house in the plant is influenced by (1) sequence of operations; (2) water supply; (3) steam supply; and (4) drainage. It is impossible to lay down any fixed rule as to the location of the dye house in a plant in its relation to other operations, because most dye houses handle the stock at more than one stage of the process. Each dye house becomes a problem in itself and its location must be decided upon by the particular conditions surrounding it and the purpose for which it is to be used. If a dye house handles its product all in one form such as raw stock, yarn or cloth, its location is easily fixed, and if it handles stock in more than one form there is generally a preponderance of one kind which determines its location. In general, the position of a dye house for various kinds of plants would be as given below, from which it will be seen that the choice of location depends upon the form of the bulk of the product.

Kind of Plant	Product of Dye House	Location
Cotton	Dyed yarn	Near warping, dressing and slashing department
Cotton	Piece goods	Near gray room and finishing department
Worsted	Yarn	Near warp preparing department
Worsted	Piece goods	Near gray room and finishing department
Woolen	Raw stock	Near scouring and picking department
Woolen	Yarn (unusual)	Near dressing room.

16 A dye house requires an abundant supply of clean soft water. If this is available from a canal or river the dye house can often be advantageously located to secure its supply by gravity and thus avoid pumping. If the water is taken from a pond, it usually has to be pumped and therefore the location of the supply has no influence on the location of the dye house. When the water is so hard as to require a softening plant, pumping is required and the same thing holds true with regard to location.

17 When the water supply is taken from a river that is polluted by sewage and waste from cities and mills upstream, it is sometimes necessary to install filter plants. This might have some bearing on the location of the dye house as well as its construction, since filter beds are sometimes placed in a part basement under the dye house.

18 A considerable portion of the steam used in a plant which has a finishing department is used in the dye house, either as live or exhaust steam for boiling liquor or for drying materials. While

it is convenient and economical to locate the dye house near the power plant, it should not be a factor of any great importance. High- and low-pressure steam can be transmitted through pipes properly proportioned and covered for a considerable distance with very little loss in heating value or pressure.

19 The waste water from a dye house often presents a problem which must be taken into consideration in locating the building. It particularly affects the height of the main floor, which must be such that the water in the trenches and cross-drains will run by gravity to the point of discharge into a river or sewer. This becomes further complicated by the fact that the future tendency is to compel manufacturers to treat the mill waste before discharging it into the rivers.

20 This part of the problem has a direct relation to the water supply when the latter is a gravity supply, and considerable care must be used in properly adjusting the levels. If the dye house is joined to another building from which it receives stock or to which it delivers stock, the same floor levels should be maintained if possible.

DYE-HOUSE CONSTRUCTION

21 *Type of Building.* Dye houses are built both single- and multiple-story and are successfully operated either way. The multiple-story building is satisfactory if it is not too wide and the operations on the upper floors are not interfered with by ventilating ducts passing through the floors. The upper stories in such cases are generally used for the driers and storage of dyed stock. When the plant is cramped for space, a dye house of this kind is justified. But on the whole a one-story building is much better for both light and ventilation and is the type of dye house best fitted to give satisfactory results.

22 *Size of Building.* In determining the general dimensions of the building a machinery organization plan must be made of the equipment to be installed. The width of the bay is not essential and is usually about ten feet. The length of the span from column to column and column to wall should be about 25 ft., providing space for one row of kettles and an ample working alley. (Fig. 1.)

23 The kettles should be installed in pairs, right and left hand, with the drives whether belt- or motor-driven, outside, and an alley of 4 or 5 ft. between ends of each pair of kettles. If there are but few kettles, they can be economically arranged in two rows

in the center of the room, the fronts facing the outer walls with a 12- to 16-ft. working alley. The backs should be set 6 ft. apart. If there are a large number of kettles, they are best arranged in four rows, the two outer rows facing the walls with a 12- to 16-ft. working alley, the two inner rows facing the center of the room with a 16- to 20-ft. working alley. The height of the room should be from 16 to 18 ft. at the eaves. If the building is three spans or more in width there should be a monitor in the middle to light the inner rows of kettles, as the window light is largely cut off by the machines and ventilators. (Fig. 2.)

FIG. 1 WORSTED DYE HOUSE

24 *Materials and Construction.* The most satisfactory type of building for a dye house is that of slow-burning mill construction consisting of brick walls, wood roof and concrete foundations and trenches. The use of exposed steel is to be avoided in every way and for this reason wood sash are preferred to steel, because of the corrosion due to moisture and acid fumes. An ordinary flat roof with a monitor, using hard-pine beams and 4-in. spruce plank, works out satisfactorily. The extra thickness of roof plank is necessary in cold climates for insulating purposes. Spruce or Douglas fir is preferable to hard pine as the latter contains resin which is softened by the heat and is likely to drop down upon goods waiting to be dyed.

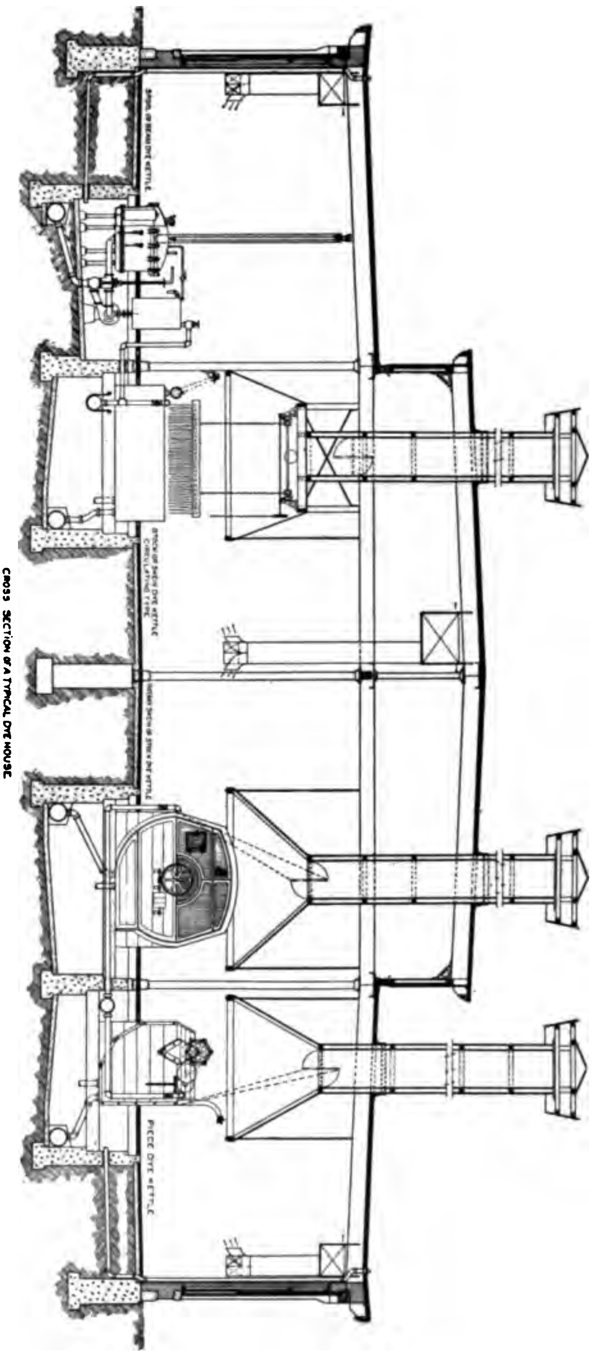

FIG. 2 CROSS-SECTION OF A TYPICAL DYE HOUSE

All timber and plank used in the roof and the monitor should be kyanized after the framing has been done and before putting into place. This necessitates that much of it be done on the job. It is the best safeguard against decay and well worth the expense.

25 *Sash.* It is desirable that dye houses located in cold climates should be equipped with double sash to prevent condensation. The extra sash is put in from the inside and can be made removable if desired. Double glazing is sometimes used to accomplish the same purpose and will be fairly effective if the space between the two lights of glass is about ½ in.

26 *Dyer's Office.* An office should be provided for the dyer, located on the north side of the room if possible, having a large window, preferably of plate glass, at which he can match sample of work taken from kettles, for shade and color.

27 *Drug Room.* Space should be provided for the storage of dye stuffs and chemicals used in the dye house. This room should be near the dyer's office and partitioned off from the kettle room. It should be well lighted and have a steam and water supply to convert dry colors into liquid form before they are placed in the kettles.

28 *Laboratory.* The modern and well-equipped dye house of today is provided with a laboratory where the dye stuffs and chemicals purchased may be tested and small samples of stock dyed. This should be located next to the dyer's office and connected to it.

29 *Trenches.* Practically all types of dye kettles and washers are set so that much of the machine is below the floor level. In times past it was customary to cut a hole in the floor which was 4 or 5 ft. above the ground and set the machine on some wood bolsters or masonry piers. The waste was allowed to run out on the ground and find its way to an open drain. This sort of an arrangement made repairs very difficult and expensive and prevented the proper upkeep of the machines.

30 This is avoided now by building concrete trenches about 11 ft. wide by 6 ft. deep in which supports are provided for the machines leaving a clear space underneath for the flow of the waste, and ample room all around to get at all parts requiring attention.

31 In large dye houses and also in those where, because of local ordinances, it is necessary to separate the hot waste dye liquor from the rinse water, an auxiliary drain should be provided in this trench. This is done by installing a separate drain pipe on one side or partitioning off part of the trench. The kettle is then provided

with two waste pipes, one of which is connected to the closed drain and one opens into the trench. If the hot liquor is run off into the open trench, it gives off clouds of steam which rise around all the machines. A main intercepting drain crosses the dye house, with which all the trenches are connected, and in some cases two such drains are provided to keep the wastes separate.

32 *Floors.* In a dye house of the above description the trenches are planked over after the machines are placed and all spaces between the trenches are backfilled and the floor laid directly on the ground. The floor consists of a concrete base with a top of either granolithic or paving brick, pitched to drain into the trenches. A granolithic top is only good in small dye houses where the trucking is light, because such floors crack more or less and break down at the edges under traffic, causing depressions which hold water and make trucking difficult. A granolithic floor that will stand up under heavy trucking can be laid by placing cast-iron grids on the concrete base and putting the granolithic finish over the grids and floating flush with the top. This floor is expensive but will stand a great deal of abuse. The best floor is obtained by using a vitrified paving brick bedded in cement mortar and well grouted. These brick should be not less than 2 in. thick and thoroughly vitrified. A common hard-burned brick will not answer the purpose.

DYE-HOUSE VENTILATION

33 The subject of dye-house ventilation is a broad one and can hardly be covered within the scope of a paper of this kind; but it is most vital in the successful operation of a dye house and deserves a great deal of attention. This problem, so far as it applies to keeping the atmosphere clear of steam or vapor and the prevention of moisture from gathering on the ceiling and overhead structure and dripping to the floor below, is largely a matter of confining the steam at its source, discharging it outside of the building by the shortest possible route, and replacing the large volume of air which is carried out with the steam, by warm, dry air. (Fig. 3.)

34 The chief cause of obscurity in a dye house when kettles are boiling is not due to steam in suspension but to the dense fog caused by the precipitation of the moisture in the saturated air when it comes in contact with cold ceilings, floors, walls, pipes and cold air seeping into the room. Some of the hot vapor rising from the boiling kettles and tubs will condense on coming in contact with the

overhead structure and the water will drip to the floor. That which is not expelled by fans or ventilators will cool off to the room temperature, creating a fog which settles toward the floor, gradually filling the whole room.

35 Many kinds of ventilating schemes have been used and are in use today, some of which are successful and others only partially so. The shape of the roof, the height of the story, and materials

FIG. 3 DYE HOUSE WITHOUT VENTILATION

being handled are factors to be taken into consideration. For instance, a piece dye kettle is probably the worst machine to ventilate, while an enclosed kettle for dyeing spooled yarn requires no special ventilation. The other machines give off varying quantities of steam.

36 The simplest form of ventilation is to depend on the natural ventilation from the windows and monitor. In very small dye houses this gives fair results in summer, but in cold weather it is inadequate. It can be helped by a large heating system designed to keep the temperature of the air above the point where fog will form, or by putting the driers in the dye house, but at the best it is unsatisfactory.

37 It is very often attempted to clear the air by using exhaust

fans in the monitor, and while this scheme will remove the hot, oppressive vapors, it does not clear the fog which lies in the lower half of the room. The air necessary to replace that driven out by tho fans, comes in at the doors, windows or other inlets and as soon as the cold air comes in contact with the warmer saturated air a fog is produced which cannot be removed by fans.

38 Another scheme is to put large wooden hoods over the machines, singly or in pairs, with a vent from each hood going up through the roof and well above it. The idea is to catch the hot vapors before they are dispersed and by means of the natural draft due to the difference in temperatures, to get them out of the room. This works out fairly well and will give good results under normal conditions in medium-sized dye houses. There is still some fogging due to cold air in winter being drawn into the room to replace that carried out with the steam, but the humidity is much lower than with open kettles so that this is not so serious.

39 Where the attempt is made to remove the steam from the dye house by means of hoods and ventilators over the kettles, or with exhaust fans, or a combination of both, a partial vacuum is created in the room which causes cold air to be drawn into the room through cracks and crevices around doors and windows, through open doors and oftentimes down some of the ventilators over the kettles which is the cause of most of the fogging. This partial vacuum also retards the natural flow of steam and air from the ventilators. (Fig. 4.)

40 Some large dye houses have been ventilated without the use of hoods over the machines by means of warm air blown in along the ceiling and delivered in large quantities about 7 ft. from the floor toward the machines. The steam, fog and surplus air escape through the monitor windows or ventilators in the roof. By this method the ceiling is kept dry and the fog is eliminated to a distance of about 6 ft. from the floor. The shafting, piping and equipment in the upper part of the room are obscured by the fog which is moving upward.

41 The system of ventilation which has given the best results is to place over the kettles hoods having vents up through the roof and to distribute warm, dry air over the room and in a blanket on the under side of the ceiling in sufficient quantities to produce a slight pressure. This prevents any cold air from coming into the room and causes the steam to discharge directly up through the vents, keeping the ceiling free from appreciable condensation and

the room free from visible vapor. The temperature of the air to perform this work should not be over 80 deg. fahr.

42 The air is furnished by a steel-plate fan taking its air through a sectional heater in which any desired number of sections can be heated depending on the temperature, and distributed about the room by a system of air ducts suspended close to the roof, having outlets delivering a blanket of warm air along the ceiling, eliminating all condensation from that source and having branch ducts drop-

FIG. 4 DYE HOUSE WITH VENTILATION

ping down to within 7 ft. of the floor with outlets discharging towards the hoods and about the room. During the summer months the windows should be kept closed and the apparatus kept in operation but without steam on the heater. With the room clear of all condensation, vapor or steam, individual or group motor drive for the dye-house machinery may be used with safety.

43 The fan may be driven by either a motor or turbine. The motor gives a constant speed while the steam turbine permits of a variation of speed which is desirable under intelligent supervision. The exhaust from the turbine is used in the heater in winter, and in the summer, or when any surplus exists, it is delivered into the low-pressure steam system and used in the dye house for boiling kettles or heating water.

DYE-HOUSE PIPING

44 Each kettle has a hot-water, a cold-water and a steam connection, which means a good deal of piping to care for, and it should be laid out so as to be accessible and permit of changes. All cold-water piping must be below ground and is usually of cast iron run along the inside of the trench on the same supports as the machines. The high- and low-pressure steam and hot-water piping are run overhead and covered.

45 *Steam Supply.* The objection to the use of low-pressure steam in dye houses for boiling kettles has about vanished, and where an ample supply of clean low-pressure steam is available, as from either bleeder or non-condensing turbines, a considerable economy can be effected. For worsted piece dyes this steam must be free from oil, but in stock dyeing for woolens the exhaust from reciprocating engines can be used without injurious results if proper precautions are taken. If low-pressure steam instead of high- is used at the kettles, the pipe sizes must be increased. Low-pressure systems have been condemned because of a failure to take this into consideration. A careful study should be made of the steam requirements of the dye house under consideration, especially with reference to peak loads, and the pipe sizes proportioned to meet these conditions. The piping should be ample to provide for the future growth of the dye house and the possible rearrangement and changing of machinery. It is difficult because of the varying load in a dye house to maintain an economical heat balance, but much can be done in the way of economy by careful arrangement and the use of surplus steam to heat the water supply.

46 *Warm-Water Supply.* There should be provided, either close to the dye house or as part of it, an ample supply of warm water for washers, crabs, rinsing and such dye baths as can start with warm water. The simplest way to accomplish this is by means of wood storage tanks located high enough so that the water will flow by gravity to the machines. The water can be heated by exhaust steam or any other means. If there is a surface condenser in the power plant, the circulating water can be used and with a small sacrifice in vacuum can be delivered at about 120 deg. and pumped from the hot well to the tanks. The effect of this hot-water supply is to speed up the dye-house operations and reduce the peak load on the boilers.

DISCUSSION

C. E. Bliss. One feature of particular interest, as presented by the author, is the conclusions arrived at in regard to placing of hoods for ventilation over open dye tubs. A dye house without hoods over tubs presents so many good features that the subject is of interest both to engineers and manufacturers, and as ventilating systems without hoods over tubs have been installed in a number of large plants and have operated successfully, it seems possible to equip most any dye house in this manner and obtain satisfactory results.

Such a ventilating system to be successful must eliminate visible vapor throughout the room, and also prevent condensation at roof. This can be accomplished by the circulation of large quantities of heated air in a uniform manner throughout the room, and at the same time, a blanket of heated air must be maintained below the underside of roof to prevent condensation.

The air circulating system must be installed so that heated air will be delivered to every part of room. It must be delivered near the floor and in sufficient quantities to force out or absorb all steam as fast as it arises from dye tubs. The greater part of moisture-laden air in the room must be forced out with the volume of heated air being discharged into it, as only a small percentage of the moisture in the air in room can be absorbed. As stated by the author, the room must be kept under pressure to prevent the entrance of the cold air around windows and at doors.

A representative of one of the large companies who manufacture ventilating equipment for industrial plants states that, with the exception of two, all installations of ventilating equipment which his company have made in dye houses in the last few years have been without hoods.

A further objection to hoods over dye tubs is that if the machine is stopped, condensation from hoods which is heavily charged with acids may drop on cloth in the upper part of the machine and spot the cloth. Hoods placed over dye tubs obstruct the light greatly.

The average size of piece dye tubs is approximately 7 ft. square, and hoods are generally built to overhang the tubs. These hoods must, therefore, be very large.

If possible, it is best to locate circulating air fan, together with heater, outside of dye house. If located inside, a leakage of air is liable to occur through the housing enclosing the heater, and as this air is charged with acid fumes it will attack the fan and duct material wherever these are of sheet metal.

W. S. Brown. The author has intimated that double sash (that is, two single sash separated by an air space) is preferable to a double-glazed sash. Is this statement based on relative heat transmission, and if so, have authoritative experiments demonstrated the superiority of the former type of sash?

Some tests made at the National Bureau of Standards four or five years ago, and reviewed in the A.S.R.E. *Journal* at that time, showed that for an air space 24 in. high, the most effective thickness was between $\frac{3}{4}$ in. and 1 in. That is, if the two lights of glass were placed at a greater distance apart, the heat loss would actually be increased, due no doubt to rapid convection currents set up.

Of course, as stated, these tests were subject to the limitation of an air-space height of 24 in. and it is possible that with double sash having air space dimensions considerably greater in both directions than the above, the transmission rate might be lessened. On the other hand, it seems likely that, notwithstanding the thicker air space, the increased convection currents set up thereby might be so great as to increase rather than decrease the total transmission rate. The question of air leakage is also involved and it would appear for double sash, especially with interior one movable, that as great or a greater loss from this factor would result.

The point is, has the proper spacing to obtain maximum thermal resistance with double sash ever been determined experimentally, or otherwise, and if so, does this spacing result in an appreciably greater resistance to heat loss through it than for double-glazed sash? The cost of double sash being considerably greater than double-glazed sash should, in order to make the use of the latter warranted, be compensated for by some well-established advantageous feature which the author may have in mind.

Has dehumidifying apparatus ever been tried in a dye house to take the moisture out of the air, and is it likely that recirculation during a portion of the year will ever be found practicable? The writer has in mind a system such as is used in ammunition plants. Of course, in a dye house, the volume of air to be handled must be relatively enormous and probably the thermal cycle of dehumidifying would be inefficient. Its practicability, therefore, seems open to serious doubt.

A. N. Sheldon. There is apparently no mention made in the author's paper as to the use of sawtooth roof construction for dye houses. We find that often good natural lighting with minimum direct sunlight is a paramount consideration; also sometimes the machinery layout is such that on account of interference from numerous hoods, ventilating stacks, etc., good lighting with monitor construction is difficult to obtain. For this reason, many manufacturers prefer the sawtooth building, and our experience has been that, contrary to a somewhat current impression or prejudice, this type of building has not been found difficult properly to ventilate and heat. Sometimes the addition of a basement is also worth while, as it affords a place to locate water, steam and effluent pipes, all of which are so arranged, together with the machinery itself, as to be easily accessible for repair. Shafting and air ducts may also be placed in the basement, which results in less congestion in the room where the various processes are going on.

The statement is noted that "the most satisfactory type of dye house building is one of slow burning mill construction." Our experience would be to modify this, giving due consideration to the value of reinforced concrete. We have built many dye houses of both types, the choice depending upon the relative importance of the various pro's and con's entering into each individual csae. Two advantages of reinforced concrete construction are its permanency and less risk from a fire protection standpoint, sprinklering being not usually required by the Underwriters. Against it is possibly a somewhat greater first cost, including the extra insulation required, but this extra expense is very often worth while.

Regarding methods of wood preservation, recent tests by the U. S. Bureau of Standards, research work undertaken by Fred J. Hoxie, Engineer and Special Inspector with the Associated Factory Mutual Fire Insurance Companies, National Electric Light Association and others, have clearly demonstrated the great value of creosote oil, and that it has certain points of superiority over kyanizing. Within a month the writer has seen samples of 3-in. plank taken from a consignment received from a large and reputable kyanizing concern, each plank being stamped, but when tested revealing that the average penetration did not exceed ¼ in. Bichloride of mercury being colorless, one method of ascertaining its penetration is by sawing of the end of a plank and treating it with ammonium hydrosulphide, which reacts so as darkly to discolor that portion of the wood which has been treated. If these samples are typical of average results to be expected, the ultimate preservation of lumber by this method certainly seems very questionable.

Charles H. Fish. There does not seem to be any great problem involved in the ventilation of a dye house if we are willing to go to the expense necessary to accomplish the results. As the author states, there is little or no trouble in warm countries or during the warm weather here in the north. The air which is supplied through open windows or doors to take the place of air exhausted through ventilating fans is sufficiently warm so that there is no condensation, and the atmosphere is clear. It is merely a question, therefore, of supplying, during the cold weather, a sufficient amount of warm air — say at summer temperature — to replace that which is thrown out by the ventilating fans.

Some years ago the writer had experience in ventilating and warming a large dye works by means of an independent heating plant, using the so-called Cummer system for heating air in large quantities. The Cummer apparatus was located in a separate building similar to a small boiler house. It consisted of an arrangement of corrugated iron plates of large area, heated by coal fires automatically stoked. A large-volume exhaust fan drew the outside air in and across these plates and sent it through a 6-x 8-ft. air duct, from which it was distributed to the various departments of the dye house and finishing plant. This large volume of air was heated to about 75 deg. fahr., and during the time the apparatus was in operation it kept these departments entirely free from fog, doing away with the necessity for ventilating fans over the machines, all the ventilating being done by natural draft; in fact all the ventilating fans in that part of the works were stopped. The expense, however, for the operation and upkeep of the apparatus was very great; and in addition, an occasional leak in or around the heating plates sent traces of coal gas into the works, which, as may be imagined, caused more or less havoc with certain shades or colors, especially the delicate or light shades; so that the apparatus was discarded, but the system was practically followed by substituting smaller heating units, which provided an ample quantity of warm air.

H. A. Burnham. Reference by the author to raw stock in a cotton dye house brings up the question of fire hazard in connection with raw cotton from the bale brought into the dye house to be dried.

Often there are only a few bales brought in by truck and there are no large opened cotton bales. Sometimes, however, cotton is transferred from a distant picker-room to the dye house and in this case the conveying system should be so arranged that the cotton will not pass through the fan. A paper on this subject was presented to the Am. Soc. M. E.[1] in 1913 by the writer. This feature can be arranged by the use of a condenser working entirely under suction and best located to discharge directly into the dye tub. If the discharge must be on the floor it should be within a completely enclosed bin of noncombustible construction.

There is some fire hazard due to spontaneous combustion of cotton piece goods which have been dyed aniline black and dried on cans and left too long without aging or steaming.

There have also been some fires resulting from handling of chlorates of potash and soda in connection with a mixture of aniline dye liquor.

There have also been cases of fire in connection with fans used for exhausting vapor from the wooden hoods sometimes used over dry cans. The hoods in such places should be of noncombustible material as far as possible, or the ventilation of the room arranged so that the hoods will not be necessary. The same idea of using some light, noncombustible material for enclosures often found necessary for tenter frames is also suggested.

Although many of the common manufacturing hazards are not present in a modern well-equipped dye house, the above will show that sprinklers as a rule will be needed for proper protection against fire.

It is recommended that the drug storage be so planned that there will be no large storage of chlorates or other extra-hazardous drugs, and no opportunity of accidental combinations with acids in the presence of very valuable drugs or colors.

With automatic sprinklers through the dye house, the drug storage room floor might well be raised above the general floor level and sharply drained to avoid an accumulation of water on the floor and consequent excessive damage to sensitive drugs stores in case of sprinkler leakage. Enclosures for the drug room, office and laboratory should be of non-combustible material, such as wire lath and cement or hollow tile in order to avoid unnecessary woodwork.

The use of slow burning construction without any steel work, and with fairly high roofs of thick plank, treated to avoid rotting, has worked out excellently in practice and is much preferred to rein-

[1] Trans., Am. Soc. M. E., vol. 35, p. 513.

forced concrete construction. even if it does bring about the need of automatic sprinklers.

The use of reinforced concrete has been tried out in some cases to avoid trouble from rotting of a wood roof and to avoid the need of sprinkler protection, which is more difficult to maintain in a poorly ventilated dye house than in an ordinary manufacturing room.

The slow burning construction with sprinkler protection and with the plenum system of ventilation and hooding of vapor producing machines, such as is advocated in the paper under discussion, has been found most effective, and a dye house so arranged offers no great difficulty in the maintenance of the sprinkler system necessary for the fire protection.

D. M. Bates. The author speaks of having a laboratory near the dyer. While it is desirable to have a place for mixing and with facilities for testing, it has been the speaker's experience as a superintendent in a dyeing and finishing plant that before the war it was undoubtedly necessary for dyers using German dyestuffs to have the quality tested by laboratories far removed from the dye house. Therefore he would not suggest having a large and expensive laboratory near the dyer's office.

J. A. Campbell said he was pleased to note that the author had shown the need of sufficient ventilation in dye houses. Not twenty per cent of the numerous dye houses of the country had proper ventilation. There was a standard construction of sprinkler pipes, and frequently heating pipes, overhead, and where there was no ventilation, there was frequently trouble with spotted goods and corroded pipes. For the ventilation of kettles, asbestos board was now becoming standard. One kettle so hooded, he said, was still in condition after nine years' service. The author had spoken of the large quantities of steam used and the speaker thought that some data from tests made on kettles dyeing worsted goods would be of interest. The steam used per set averaged 4500 to 4600 lb. The process took about 4½ to 5 hours. The speaker said that he believed in the use of a turbine or steam-engine drive for ventilating fans owing to the ease in varying the speed.

George H. Perkins asked the author about his experience with concrete roofs for dye houses, and also if he had any data on the relative heat transmission through a concrete roof and through the heavier planking he had recommended for roof construction.

THE AUTHOR replied that he had never built a dye house with a concrete roof, so he had no personal experience from a structural standpoint. From personal observation there was much to be desired in such a roof and much to be overcome. Charles N. Norton had made a number of experiments about two years ago, he said, as to the radiation from different kinds of material for roofing. For 3-in. spruce plank, the results showed 4.61 B.t.u. for 24 hr., for 4-in. spruce plank, recommended in the present paper, the loss was 3.87, and for terra cotta tile, 8.35. For 6-in. concrete, the loss was 12.7, for 4-in., 12.2, and for 4-in. as ordinarily used, 13.8. For 3-in. concrete the loss was 16.2. In other words, the radiation from the concrete surface was much higher than from the ordinary spruce plank. This could be overcome somewhat by using a sufficient quantity of cork, but the construction was expensive and therefore not considered worth while. With regard to the longevity of concrete compared to spruce plank, it would be fair to contend that the spruce plank would last as long as was necessary. The cost of a ventilation system such as mentioned in the paper varied from 7.3 cents up to about 9 cents per cubic foot. This would average about a dollar and a half for a building such as outlined in the paper, based on present prices for an equipment of the kind mentioned.

In answer to a question as to whether the workers in the dye house objected to the blast of air blown on them by the fan, the author replied that the construction was such that the air did not blow on them. It blew toward the men, but did not come in direct contact with them. So far as he knew, there had been no objections from the men.

CHARLES H. BIGELOW. In a recently constructed finishing house, a concrete roof was covered on the outside with lumber, leaving an inch air space in which was placed some seaweed paper. With this construction there resulted a saving of one thousand dollars in the heating of the building, and there was practically no condensation.

No. 1775 *a*

A PHOTOGRAPHIC STUDY OF THE WOOD-WORKING INDUSTRY

By Frederick F. Murray,[1] St. Louis, Mo.
Non-Member

As a general introduction to the Woodworking Session there was presented a photographic study of the methods employed in logging and in the manufacture of lumber. The field was covered briefly and with but little detail, for the aim was to create some appreciation of the magnitude and diversity of just a part of one of the greatest industries of the country.

The views were selected from a set of several thousand photographs of recent date, each chosen for its value as a representative and largely self-explanatory subject pertinent to the business of the Session. With this selection very decidedly limited as to numbers, there was an obvious deficiency in depicting manufacturing sequences, and there were numerous voids and seemingly wilful omissions of various phases of the industry, all of which were necessary in order to make the argument of reasonable length. In the very limited time at the disposal of this meeting, however, and in justice to the remainder of the program, this photographic study could hardly have been made otherwise.

An outline of the scope of the study follows:

LOGGING

- Characteristic woods topography
- The logging railroad
  - Construction and general difficulties encountered
  - Grading machines, trestles, etc.
- Felling the tree (a word about mechanical fellers)
- Moving the tree from where it falls
  - Ox teams, horses, chutes, flumes, gasoline tractors, steam skidding

[1] Mechanical Engineer, *Lumber*.

Presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers.

machines, self-propelled railroad skidders, self-propelled log car loaders, skyline and high-lead settings, the electric logging engine
Logging locomotives
The general locomotive, simple rod locomotive, articulated Mallet
The motor truck in logging.

MANUFACTURING

At the Sawmill

The log dump
Floating and piling logs
Log slip and cable haul-up
Log washing
Sawmill power plant
Boiler room, engine room
Fire- and water-tube boilers, reciprocating engines, mixed-pressure turbines, high-pressure turbines, the turbine condenser
Primary saw
Circular mill, band mill, log carriage, log turners
Electric drive for the saw
Character of load, synchronous condenser and its use
Rollers and chain conveyors behind the saw
Hand and electric control of jump chains and stops
The edger
Special high-powered electric drive
The selective trimmer
Hand control, pneumatic control, electric (solenoid) control
Resawing and remanufacturing
The gang saw
Sorting or grading table
Drying
Kiln drying and air drying (artificial and natural drying)
Yard distribution and piling.

At the Planing Mill, Box Factory, Etc.

Handling systems, rough side of planer
Bridge crane, narrow-gage railroad, dolly carts, etc.
Shavings disposal
Smooth side of planer
Direct car loading, storage
Battery tractors — planer and sheds to shipping platform
Motor-truck shipping.

Wastewood and Reclamation

Lath manufacturing
The hog to reduce scrap for fuel purposes
Utilization of wood waste for fuel outside of the sawmill
The passage of the wood burner to dispose of wastewood.

No. 1775 *b*

ENGINEERING IN FURNITURE FACTORIES

By B. A. Parks, Grand Rapids, Mich.
Associate-Member of the Society

THE woodworking industry, including the manufacture of wooden furniture, is one of the oldest industries extant and yet it has shown the least development and has been the slowest to adopt modern principles of manufacturing of any industry of which the writer has knowledge. There are several causes for this condition, the most important of which are the general lack of accurate cost data in all but a very few plants and the absence of technically trained men in the executive positions.

2 The lack of accurate cost data is naturally reflected in the margin of profit, which in the majority of cases has been so small that no surplus has been available for plant improvements and very little for even proper maintenance. In several plants which have come under the writer's notice, only such repairs have been made as were necessary to avoid actual breakdowns, and machinery, power-plant equipment, lighting, heating, drying, sanitary facilities, etc., have been entirely inadequate and inefficient and would not be tolerated in even the average modern plant of most other industries.

3 In the several woodworking and furniture-manufacturing plant organizations that have come under the writer's observation, he does not recall a single man with a technical education or training. The development of practically all modern industries to their present high state of manufacturing efficiency has been accomplished by engineers, as witness the metal-working industry, and who would dare say that the metal-working industry would ever have reached its present state of development without the guiding hand of the technically trained engineer?

4 This lack of engineering ability in the furniture-manufacturing organization shows its effect throughout the entire plant; in

Abstract of paper presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For complete paper see Mechanical Engineering for February 1921.

fact, the writer is convinced that the average manager of a furniture plant is more interested in marketing his product than in manufacturing it. Furniture is constantly changing in style, also most plants manufacture quite an extended line, and consequently a large variety of product must be handled in any given plant. A point which most managers overlook and which is primarily due to lack of engineers in the organization, is the possibility of reducing the variety of parts to be manufactured through standardization of design, interchangebility of parts, and greater limitation of line. This reduction of variety of parts would not only directly reduce manufacturing costs but would also tend toward the development of automatic machinery, better utilization of raw product, economies in handling parts in process of manufacture, etc.

5 It is not the writer's purpose to discuss the manufacture of furniture, however, but only to point out some of the influences which have militated against the more rapid development of the industry in the past, and then proceed to a discussion of some of the principles involved in the design of a new furniture-manufacturing plant and the improvement of existing plants.

6 A furniture-manufacturing plant, as in fact any industrial plant, is in the final analysis simply a workshop for turning out a certain product of a specified quantity and standard of quality in a given time at a minimum of cost. While the problem is easy to state, the solution is very complex, and there are so many factors to be considered that the final solution must of necessity be more or less of a compromise.

7 The efficiency of any general plan for a furniture-manufacturing plant, or in fact any industrial plant, may be measured by the degree in which the following requirements are fulfilled:

- (*a*) Provision for proper arrangement of the necessary machinery
- (*b*) Provision for receiving, handling, storing, and transporting material
- (*c*) Provision for extending the plant and increasing manufacturing facilities without serious disturbance to the original plant or manufacturing routes
- (*d*) Provision for generation and transmission of power, light and heat
- (*e*) Provision for fire protection
- (*f*) Provision for comfort and accommodation of employees.

No. 1775 c

THE USE OF WOOD IN FREIGHT-CAR CONSTRUCTION

By H. S. Sackett,[1] Chicago, Ill.
Non-Member

The following paper is a discussion of the relative values of steel and wood as materials for the construction of freight cars. The author states that from recent studies of the subject wood is still regarded as the most economical material, and this fact is elaborated upon by a review of the merits of steel and wood in the construction of both open-top and closed-top cars. The use of graded lumber in freight-car construction is also touched upon and the paper is concluded with a review of the advantages of wooden cars and a word of caution as to the importance of using nothing but well-seasoned lumber in all construction.

THE fact that over two billion feet of lumber and timber are used annually in the United States for the maintenance of freight equipment and for the construction of new cars, representing an annual outlay for material alone of over $50,000,000, is ample evidence of the importance of wood in this big industry.

2 The first freight cars used for railroad transportation in this country were built of wood and this great natural resource has been an important factor in their construction ever since. In recent years steel has competed with wood, especially in the construction of gondola and other types of open-top cars, but whether this change has been an economical one or not remains a question.

3 From recent studies made of the subject by disinterested commissions, the indications are that even for open-top cars, wood is still the most economical material. With the growing scarcity of timber, however, and the apparently unlimited supply of steel available, it is probable that this latter material will always be a prominent factor in the construction of cars, and especially so, if

[1] Asst. Purchasing Agent, Chicago, Milwaukee and St. Paul Railway Company.

Abstract of paper presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For more complete account see Mechanical Engineering for February 1921.

some agent is found to reduce materially the present high rate of loss from rust and corrosion.

4 In the construction of the first cars wood was almost universally used for all parts, except of course the running gear. Within the past decade, however, with the introduction of heavy motive power which spells long and heavy trains, it has been found that wood is no longer capable of withstanding the heavy shocks incident to such operation, and it is generally conceded by all car builders that the freight car of the future must be of steel underframe and steel draft rigging.

5 This limits the use of wood to unimportant parts of the underframe, such as intermediate sills and cross-beams, and to the superstructure of the car. In the case of open-top cars this means only posts and side and end plank and decking, but in box, stock, and furniture cars, etc., it means also siding, lining and roofing.

GRADED LUMBER IN CAR CONSTRUCTION

6 It seems pertinent to say a word at this time as to the grades of material required for the different parts of the body of the car. In the early days when the supply of wood in this country was thought to be inexhaustible, clear grades of wood were generally demanded by the car builders, and in the construction of new cars this is generally the practice at the present time, especially for car roofing and siding, — sound-knotted stock being used for decking and lining. Some of the more progressive railroads, however, have gone a step further in recent years and many are now using sound-knotted stock for siding and roofing for repair and maintenance work. There is no doubt but what such a practice is economical and based on sound judgment, for certainly there is no necessity for using clear material for the repair of many classes of freight equipment, the life of which may not be in excess of seven to ten years. One large railway system which not only constructs its own cars but does repairing on a large scale, has adopted the following practice as to grades:

FOR NEW CARS

Siding	No. 2 Clear and Better Fir
Roofing (double board roof)	No. 2 Clear and Better and Select Common Fir
Roofing (metal-covered)	Select Common Fir
Lining	Select Common Fir
Decking	Select Common Fir

FOR REPAIR WORK

Siding	No. 3 Clear and Select Common Fir
Roofing	Select Common Fir
Lining	No. 1 Common Fir
Decking	No. 1 Common Fir

On this particular railroad the saving on this amended practice amounts to over $500,000 per year, and its economy is apparent.

7 As far as specifications for end sills, door and side posts, belt rails, etc., are concerned, practice varies widely among the various railroads. Some still insist on oak for such purposes, and others have gone to fir and pine. With the rigid steel underframe it is believed the use of soft woods is more justified than ever for posts and belt rail, but unless well protected with steel plates it is believed the use of oak end sills is still to be recommended. Oak and other hard woods are becoming more scarce and higher in price each year and it is felt that the general practice very shortly will be pine or fir end sills properly reinforced with steel.

SOME ADVANTAGES OF WOODEN CARS

8 An important feature of car construction which is decidedly in wood's favor, is its general ease of working and adaptability to repair, and this is brought about not only from its qualities which make it easy to cut, saw and shape, but also by its almost universal availability. We have been a wood-using nation for over a century, and artisans familiar with its properties and able to work it, are always at hand.

9 It is also rather curious that the salvage value of wooden cars is greater than that of steel cars, and no one doubts this who has seen the two types of cars in a seriously wrecked condition. The wooden car may be quickly and easily repaired, while the steel car is only rehabilitated at a high cost, or for the most part is fit for the junk pile.

No. 1775 *d*

MACHINING RAILROAD CROSS-TIES

By D. W. Edwards,[1] Washington, D.C.
Non-Member

Railroads both in this country and abroad are realizing more and more the importance of tie conservation. Years of observation and tests on both treated and untreated ties, have proved that there is marked economy in the machining of railroad cross ties by trimming, adzing, boring and branding. The machines for these operations are installed in either of two systems, portable and stationary. The author gives the advantages of each, describes the machines employed and outlines the processes followed in both systems.

THE cost of railway-track maintenance has vastly increased in recent years and one of the largest single items of expense involved in this work is the cost of cross-ties. Not only is their cost increasing with the diminishing supply of the most suitable timber and the growing scarcity of labor, but their life when unprotected grows shorter because of the greater destructive effects of heavier wheel loads and more frequent trains. Also the labor cost of renewing ties, exclusive of the value of the ties themselves, has advanced.

2 A consideration of these facts leads to the conviction that there is no department of railway administration in which there is such an opportunity for large saving as in the field of tie conservation. It is encouraging to observe that the number of plants for the chemical treatment of ties is constantly increasing, heavy tie plates are coming into more general use, and a growing appreciation of the importance of improved rail fastenings is apparent.

3 There are two causes of tie deterioration, decay and mechanical wear, and there is no economy in increasing the resistance to one without also increasing the resistance to the other. In some localities decay proceeds more rapidly than mechanical wear, and in arid sections ties wear out before they decay, but as an average

[1] Greenlee Bros. & Co.

Abstract of paper presented at the Annual Meeting, December 1920, of The American Society of Mechanical Engineers. For complete paper see Mechanical Engineering for February 1921.

the two destructive agents may be considered of practically equal importance.

4 It is evident that efficacy of treatment can be realized by doing all cutting before the treatment takes place so that the chemical may present an unbroken barrier to the attacks of decay spores. The majority of all ties are so winding or crooked that they should be adzed to secure proper bearings for the rails, but to do this after treatment is folly as it nullifies the effect of the treatment at the points where it is most needed, around the rail fastenings.

5 Smooth, safe, and durable track demands the equal distribution of the load over all ties. This result can only be obtained by providing a perfect and equal bearing for each rail on every tie. By so doing it is possible to eliminate half-moon breaks in rail bases, to greatly reduce rail cutting and to lower the labor cost for upkeep, because surface is more easily maintained. This can be done accurately only by a machine designed for the purpose. It cannot be satisfactorily done by hand adzing.

6 Tests made by the United States Bureau of Forestry, and independently by many railway companies, show that the holding power of common square spikes is increased by driving them into previously bored holes of suitable size. The resistance to transverse flange pressure is also increased because the spikes have a backing in the solid wood and not against the torn and distorted fibers. When the spikes are driven into bored holes the gage is better maintained, respiking is less often necessary and the ties are not so soon spike-killed.

7 Where ties are to be treated it is essential that they be bored for the spikes before treatment as the holes permit the chemical to enter the interior portion of the tie. When the spikes are driven there is no possibility of the grain of the wood being opened to moisture beyond the penetration depth. The elimination of decay around the spikes removes the most fruitful cause of the necessity for tie renewals. A lighter total treatment will give the same or better results because the effect is concentrated where most needed.

8 Briefly, ties are adzed to assure perfect rail-plate bearing; bored to permit chemical penetration, provide correct gage and perfect spike support; and trimmed for appearance and inspection. That the trimming operation is of the utmost importance and worthy of the most serious consideration cannot be too greatly emphasized.

9 It has been almost universal practice on the English and many European railways to machine their ties before attaching the rails or rail chairs to them. The consensus of opinion there is to the effect that such preparation by machining is of marked economy. Years of observation, and tests on both treated and untreated ties, have proved that longer life is obtained by the machining operations.

10 American railroads that have assisted in the development of machines for adzing, boring, and trimming ties believe that great economy in track maintenance is derived through their use. Every road that has made an installation of equipment of this character, after a period of practical tests through use of machined ties, has added to its original plants, and is endeavoring as rapidly as possible to provide for machining all ties used in main-line track.

11 The operations performed on machined ties are trimming, adzing, boring and branding. Standard machines, developed through ten years of more or less experimental work, are now on the market and a great many installations are in successful operation in various tie yards throughout the United States. The trimming, adzing, boring and branding machines are in the main built in individual units and so installed that the ties pass automatically from one to the next. Any one or more may be either installed or operated without consideration of the rest.

12 There are two distinct systems followed in installing these machines, each of which has advantages peculiar to itself, the choice being governed by the conditions under which it must operate. These are installations in fixed locations, and portable installations in which the machine is mounted in a car which may be moved from place to place. Each system has strong advocates but apparently there is little difference in the ultimate economy. The stationary type has somewhat greater output, but this is offset in a great degree by the cheaper handling of ties to the portable type. Since there are more stationary than portable plants in operation at the present time, doubtless this type is deserving of the closer study.

No. 1775 *e*

CREOSOTED WOOD-BLOCK FACTORY FLOORS

By Lambert T. Ericson [1], Toledo, Ohio

Non-Member

This paper deals in a broad way with the problems involved in constructing creosoted wood-block factory floors. The character of the timber commonly used is first discussed; then the problems commonly encountered and the essential requisites for success enumerated. The reasons for the use of thoroughly air-seasoned lumber for solid foundations and for waterproof and elastic binders are also given. The field which may be covered with floors of this character is next described, with the conditions commonly encountered. It is also shown that the conditions encountered in factory floors vary so extremely that it is impossible to lay down general specifications covering the treatment and installation. It is demonstrated that this is a specialty which should be handled by those who are thoroughly expert in the work. Methods are described which will permit engineers to specify and build creosoted wood-block factory floors without entailing any risk of the failure of the same. It is recommended that the manufacturer shall furnish creosoted wood-block factory floors upon an installed basis so as to obviate a division of responsibility and to insure the successful installation.

THE use of creosoted wood blocks for factory floors has been so extensive during the past few years that it is hardly necessary to go into details in regard to the advantages of this type of flooring. The splendid way in which this material withstood the extreme service to which it was subjected in countless munition and shell plants during the war helped to elevate it to its true rank. Most engineers and architects have come in contact with this material at first hand, so that this paper will be confined to a discussion of:

(*a*) The material used
(*b*) The problems encountered
(*c*) The field covered.

2 Wood floors have many advantages over all other kinds—they are comfortable under foot, resilient and do not radiate heat or conduct cold. Wood in the form of planks or matched flooring

[1] Chief Engineer, The Jennison-Wright Company.

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is not, however, adaptable for heavy shop purposes as it disintegrates under heavy traffic and the abuse incidental to shop practice. It is also unsanitary and adds to the fire risk after disintegration commences. Creosoted wood blocks, on the other hand, if properly installed, have all of the advantages cited above, without the disadvantages, as they are practically wear-proof and are very fire-resistant when placed upon concrete.

3 Southern yellow pine has been used almost entirely for this work, except on the West Coast, where Douglas fir and tamarack is used. Long-leaf pine is usually specified, but short-leaf pine is also adaptable for the work. The lumber should be thoroughly air-seasoned before being cut into blocks, and then should be given a preservative treatment with coal-tar creosote oil to preserve the wood from decay. Hard woods, such as gum, beech, and maple, can be used, but they are not favored on account of the difficulty in properly seasoning the lumber and the tendency for the blocks to warp and check after they are cut. The coniferous woods are more homogeneous in grain and texture and are consequently more adaptable. Soft wood compacts under service and consequently it is just as serviceable as hard wood, and has the advantage of not becoming slippery under traffic.

THE DEVELOPMENT OF WOOD-BLOCK FLOORING

4 Creosoted wood-block factory flooring is a logical development from street paving, but in this development a great many mistakes have been made due to misunderstanding of the problems involved. As a general rule the problems are just the reverse of those in street paving. A great many failures in creosoted wood-block flooring in the past have been due entirely to improper methods of treatment and installation of the blocks. Occasional failures still occur, due to lack of understanding, by those in charge of the work, of the problems involved.

5 It should be understood in the beginning that factory flooring with creosoted wood blocks cannot be grouped under one general heading and specification. The conditions under which the floor is to be used must first be studied and the specifications made to suit. Creosoted wood blocks can be laid to meet practically all factory conditions if the conditions are first properly analyzed. The real reason for most of the trouble encountered with this type of flooring is the fact that wood expands and contracts with various

conditions of the atmosphere and the moisture content of the blocks. It is therefore necessary to lay the individual blocks in such a way as to allow for this change in volume, which in extreme cases may be as much as 5 per cent. The individual units in the floor must be bound tightly in place with a binder which will allow this expansion and contraction and which will exclude water from the underside of the floor. As long as the blocks are held tightly and firmly in place and a smooth surface is maintained the floor will wear almost indefinitely, but as soon as the blocks become loose and the surface becomes rough they will break up into sticks very quickly.

CONDITIONS TO BE MET IN A SUCCESSFUL INSTALLATION

6 The three most essential requisites for success are:

(*a*) Thoroughly air-seasoned lumber

(*b*) A smooth, solid foundation base

(*c*) A waterproof and elastic binder to hold the units in place.

In the majority of cases factory floors are dry most of the time; consequently, the lumber should be thoroughly seasoned, in order to keep the shrinkage to the minimum. Blocks cut from green or semi-dry lumber will shrink in volume to such an extent that they will often have to be taken up and relaid. If it is possible to do so, it is advisable to use a concrete base for the installation of these floors. The base should be strong enough to carry the entire load and should be finished smooth and level so that it will not be necessary to use a cushion between the concrete and the blocks in order to secure a level floor and uniform bearing for the individual units.

7 The use of a sand cushion between the blocks and the concrete, which was almost universally used a few years ago, was largely accountable for a great many of the failures. Sand shifts easily under traffic and heavy loading and also affords pockets for the collection of moisture under the floor. Wherever a cushion is necessary between the concrete and the blocks, it is advisable either to use a mixture of portland cement and sand or a bituminous mastic. The latter is preferable in a great many cases on account of being both waterproof and elastic. The sole functions of a cushion should be to level up the inequalities of the base and, in the case of the bituminous cushion, to furnish a waterproofing membrane on the underside of the floor. It is standard practice today to lay the blocks directly upon a smoothly finished concrete base without any cushion whatsoever. It is also customary to give the base a thin, even coating

of coal-tar pitch before installing the blocks, so that the underside of the blocks may be thoroughly sealed and made waterproof.

8 The elimination of cushions and the use of a successful waterproof binder in the joints of the blocks, thereby eliminating the possibility of shifting of the base and a loosening of the units, has permitted a reduction in the depth of the blocks used. Factory floors are now being very successfully installed throughout entire manufacturing plants with blocks as shallow as 2 in. in depth. Furthermore, the service obtained is superior to that obtained in past years with blocks two or more times as deep where the weight of the blocks was depended upon to keep them down and in surface.

9 Successful installations may be made on timber and plank foundations in mill-type buildings, but care must be taken to see that the timber in the foundation is sound and that the blocks are afforded a firm and even footing. A bituminous-mastic cushion between the planks and the blocks is now being extensively used and is proving very successful.

10 For dry locations the blocks should be driven up tightly together when installed; there is a tendency, however, for a slight further contraction in the volume of air-seasoned blocks when placed in the floor of a steam-heated room, and therefore it is necessary to provide a binder in the joints to hold the blocks in place after this contraction has occurred and to exclude moisture and foreign matter from seeping under them.

THE USES OF WOOD-BLOCK FLOORING

11 Creosoted wood-block floors are being installed in machine shops, forge shops, foundry molding and core rooms, casting cleaning and chipping rooms, warehouses, leather and paper mills, automobile assembly plants, garages, loading platforms, etc. The conditions in the above work vary from extremely dry to a saturated moisture state. They also vary from low to high temperatures, and in some cases the blocks are submitted to the action of molten metal. The extreme variation in the demands covered by the above list of industries gives a fair idea why it is necessary to study the individual job at hand and draw a specification to meet the particular work.

12 In machine shops the floor is often subjected to oils, gasoline, and various acids; in forge shops it is subjected to vibration, to abrasion, and to contact with hot metal; in foundries the heat is often extreme and the blocks are frequently sprayed with sparks from molten metal at one part of the operation and with water a

little later. In leather and paper mills portions of the floor are constantly flooded with water and acid solutions. In loading platforms the problems are similar to those in street paving — that is, the conditions are alternately wet and dry. In upper floors of reinforced-concrete buildings, and also in mill-type buildings, the blocks are usually submitted to extremely dry conditions which cause them to contract to their minimum dimensions. All of these conditions must be taken into account when designing the floor.

13 For ordinary factory purposes the proper treatment for the blocks is 6 lb. of creosote oil per cubic foot of timber. This is injected by the Rueping process and is sufficient to preserve them from decay. When they are to be subjected to considerable moisture or to the weather, they should be treated with 12 lb. of oil per cubic foot of timber by a combination of the empty- and full-cell processes. This extra amount of oil insures better waterproofing of the blocks.

14 Floors which are to be subjected to considerable moisture, or to weather conditions, should be laid with ample provision for expansion. It is good practice to provide ample space between the individual units to take care of this expansion. These joints should be flushed full of a waterproof, elastic binder, which should preferably be coal-tar pitch of a consistency which will not soften up under atmospheric or room temperatures. Coal-tar pitch has proven the most successful binder and filler for creosoted wood blocks, as it is a derivative of the same base as creosote oil, and thus readily unites with the oil in the blocks.

15 It is universally conceded that wood-block floors are ideal when properly constructed. In view of the fact that a great deal of the success depends upon the design and construction of the floor, it has become standard practice for the block manufacturers to not only furnish blocks, but to take charge of their installation as well. This not only obviates a division of responsibility, but also insures the installation of the floor under the supervision of experts. The block manufacturer should not only be permitted to study the conditions and assist in drawing the specifications, but he should be required to furnish a finished floor. This procedure will practically eliminate floor failures and will increase the popularity of creosoted wood-block floors in every locality.

No. 1775 *f*

PROCESSES AND EQUIPMENT USED IN WOOD PRESERVATION

By E. S. Park,[1] Pittsburgh, Pa.

and

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Non-Members

PROCESSES for the use of a preservative agent in the treatment of timber to prevent its decay or its destruction through the boring of insects may be classified into three groups: (1) Surface application: coating the timber with preservative by means of brush or spray; (2) open-tank treatment: immersion of the timber in preservative in open tanks; and (3) pressure treatments: immersion of the timber in closed tanks or cylinders with the application of pressure above atmospheric to force the preservative into the wood.

2 Application of the preservative by means of a brush or spray is, of course, the simplest and cheapest method of treating timber, but it is the least effective. It can be used only with preservatives of the oil class and, at best, gives but a slight penetration. This method is confined largely to the butt treatment of poles and posts and is usually employed only when the quantity of material to be treated or the location does not warrant the installation of equipment for treatment by a more efficient process.

3 In open-tank treatment several different methods are employed and varying results obtained. The mere dipping of the material in the preservative as practiced in the butt treatment of poles and posts is of course an improvement over the brush or spray application. A more efficient process consists of a prolonged bath in heated preservative, relying on the absorptive properties of the wood to secure penetration. This method is best adapted to the treatment of small quantities of timber or for use with a preserva-

[1] Pittsburgh Wood Preserving Company.

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tive that is corrosive to iron. It is a rather slow process and usually expensive as the quantity of material treated at one time is small and as the preservative must be kept hot for several hours.

4 The pressure process has as its prime object, first, the distribution of the preservative throughout the wood as uniformly as possible; second, the securing of an absorption of a sufficient quantity of the preservative to insure the results desired. The essential feature of all pressure methods is the use of pressure to force the preservative into the wood. This is accomplished in several ways, the kind of wood, the use for which it is intended or the kind of preservative used, making it necessary or advantageous to vary the methods employed. The timber is treated in closed retorts or treating cylinders built to withstand a pressure of 250 lb. per sq. in. and equipped with steam coils for heating the preservative.

5 The most widely used preservatives and those generally recognized as the most efficient are coal-tar creosote and zinc chloride, the latter being applied in the form of a 2 to 4 per cent aqueous solution. In one process employed in the treatment of cross ties a mixture of 80 per cent zinc chloride solution and 20 per cent creosote is used. In treating with creosote the amount injected ranges from 4 to 20 lb. per cu. ft., depending on the kind of timber, the process employed, and the proposed use of the treated timber. When zinc chloride is used the aim is to treat the wood to refusal, the strength of the solution being regulated to give an absorption of approximately ½ lb. of dry salt per cu. ft. of the wood treated.

6 Pressure treatments may be grouped into two classes as follows:

a Full-cell process, the object of which is to fill the intercellular spaces of the wood as completely as possible with preservative

b Empty-cell process, the object of which is to secure as thorough and deep a penetration as possible with the use of a minimum quantity of preservative.

When the preservative used is creosote the full-cell treatment is known as the Bethell process; when zinc chloride is used, as the Burnett process, and when a mixture of creosote and zinc chloride is the preservative, as the Card process. In all three processes the methods used are much the same.

7 In the full-cell process the timber is placed in the retort, a vacuum is drawn, and without breaking the vacuum, the retort is completely filled with the preservative fluid. The vacuum not only accelerates the entrance of the preservative into the retort

but also makes it possible to force the preservative into the timber more quicky and with less pressure than is the case when the preservative must displace or compress the air in the wood. After the retort is filled additional preservative is forced into the cylinder by means of pressure pumps, the pressure being gradually raised to and maintained at 125 to 180 lb. per sq. in. until the required amount of preservative has been forced into the wood. The pressure is then released, the preservative drawn from the cylinder, and as a rule, another vacuum is drawn, the object of this final vacuum being to remove the surplus preservative from the surface of the timbers and hasten the dripping of the preservative so that the timber can be removed from the cylinder as soon as possible.

8 In the empty-cell process no preliminary vacuum is employed and to enter the wood the preservative must therefore displace and, to some extent, compress the air in the wood. Such is the treatment known as the Lowry process. In the more widely used Rueping process the entrance of the preservative into the wood is further retarded by subjecting the timber to an initial air pressure of from 50 to 75 lb. per sq. in. after which the preservative is forced into the cylinder at a higher pressure. After the treating cylinder is filled the procedure in empty-cell processes is practically the same as in full-cell treatments except that the final vacuum is held longer. The results obtained, however, are quite different, as in the empty-cell treatments the release of the pressure and the removal of the preservative from the retort permits the expansion of the air compressed in the wood. The effect of the final vacuum is to cause further expansion and, to some extent, expulsion of the air in the wood and a corresponding expulsion of a portion of the preservative that was forced in during the pressure treatment. The result is a thorough penetration with a minimum amount of preservative. Empty-cell methods are used almost entirely with creosote and in treating timbers which do not require the complete filling of the interstices of the wood. The cost of such a treatment is considerably less than by full-cell processes, as from 25 to 40 per cent less preservative is required.

9 The paper continues with a description of a typical treating plant.

No. 1775 *g*

ELECTRICALLY DRIVEN SAWMILLS

By Allan E. Hall,[1] Milwaukee, Wis.
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THE complete modern plant for the manufacture of lumber includes everything required to transform the growing tree into finished lumber ready for the builder; and this plant naturally divides into forest equipment and mill equipment. The forest equipment, which includes logging machinery and railroad and water transportation of the logs to the mill, is not considered in the present paper. The mill equipment, to which attention will be given, reduces the log to lumber, dries it, finishes the surface of the lumber if desired, and so converts the log into a marketable product.

2 The mill equipment includes the sawmill proper, the dry kilns, the planing mill, the necessary storage sheds for kiln-dried lumber rough and finished, and a storage yard for part of the lumber which is stacked outdoors for air drying. The sawmill breaks down the log into rough-surfaced pieces of various sizes. Part of this lumber may be shipped "green" from the saw, but most of it is either air-dried or kiln-dried. Part of the dry lumber is also shipped rough, but the higher grades are generally surfaced in the planing mill and made into flooring, siding, ceiling, and numerous other special forms.

3 The capacity of a sawmill is regularly given in thousands of feet board measure per day of 10 hours. The power required for sawmill alone varies from 4½ to 8 hp. per 1000 ft. of lumber per day; e.g., a sawmill of 100,000 ft. daily capacity will require from 450 to 800 hp. The lower figure is for mills cutting small and medium pine logs; the higher figure for Pacific Coast mills working the heaviest timber, or mills sawing hardwood.

4 The planing mill will require from 2 to 3½ hp. per 1000 ft. on the same basis. The total power for milling is therefore from

[1] Allis-Chalmers Manufacturing Company

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$6\frac{1}{2}$ to $11\frac{1}{2}$ hp. per 1000 ft. board measure of lumber sawed per 10 hours.

5 In deciding whether to drive a sawmill by lineshaft or motors, the first consideration is the probable life of the plant. Unlike most manufacturing plants, the sawmill must nearly always be built close to the supply of raw material; for it is not commercially possible to transport sawlogs far from where they grow, except in the case of valuable timber like mahogany and other tropical hardwoods. When the supply of accessible timber is sawed, the plant must be abandoned or moved with a very small salvage value. Sawmills which have a timber supply in sight to saw for twenty years or more form a small percentage of the total number. The first cost must therefore be kept down to a figure which can be wiped out from the profits in a few years — fifteen, ten, or even eight years — without making too great an annual charge.

6 The size of the sawmill plant is important. It is found that for very small mills the first cost of the electric power plant and motors is greater than for a steam plant and belted drive. For medium-size sawmills the first cost does not differ greatly when everything is considered; and for large plants the first cost may be less for a motor-driven than for a shaft-driven mill.

7 Accessory or by-product equipment will affect the choice between the two kinds of power transmission. As previously explained, the planing mill is considered as a part of the complete installation; but beyond this every intelligent lumberman is constantly trying to make the waste wood from his sawmill into useful products. As an example, a large mill recently built for the manufacture of yellow-pine lumber makes from waste wood (*a*) kiln sticks for spacing lumber in the dry-kiln stacks, (*b*) lath, (*c*) rosin-barrel staves, (*d*) shingles, (*e*) box boards and cleats, (*f*) short stove wood, (*g*) molding strips and (*h*) ground chips for fuel. Moreover sawmills are built today for producing small wood specialties primarily from the log, the output of lumber in the form of boards or other building material being small and incidental. One large plant has been built recently for making oval wood dishes, butter boats and clothes pins; another makes all possible stock into agricultural-tool handles, barrel hoops, door knobs, teapot buttons, and a great variety of small woodturnery, while a third has given serious consideration to the question of putting in machinery for making candy sticks — those used in the manufacture of lollypops. A small sawmill followed by such a remanufacturing plant is usually

motor-driven, while the sawmill alone might not justify the investment; for the small specialty machines can be driven electrically with convenience and economy.

8 A condensed list of the machines found in the ordinary sawmill follows, the sizes of motors suitable therefor being stated in each case.

MACHINES	MOTOR SIZES
Log jack, for hauling logs into the sawmill	25 to 50 hp.
Circular log cut-off saw, for dividing long logs after being drawn into the mill	25 to 50 hp.
Drag saw, for cutting off logs too large for the circular saw above mentioned	15 to 40 hp.
Overhead log canter, for turning large logs lying on the log storage deck or on the carriage	10 to 20 hp.
Band mill, for ripping the logs which are moved back and forth in front of it on the carriage	100 to 300 hp.
Circular head saw, sometimes used instead of band mill	100 to 400 hp.
Power setting machine, for moving log forward after every cut into position for the next cut. This machine is mounted on the carriage. Two types are used, one direct steam-driven, the other power-driven. The latter is used exclusively for the heaviest work, and motor may be mounted on carriage	5 to 10 hp.
Live rolls, or power-driven rolls, for transporting lumber after being sawed. These are connected and driven together in trains of 6 to 20 or more, according to the mill design; power required per roll	0.4 to 0.6 hp.
Band resaw, for further reduction of large pieces dropped by the head band mill. It may be either vertical or horizontal. The latter is used for resawing slabs as well as splitting thick stock from the head band	75 to 200 hp.
Transfer chain tables, for moving lumber sidewise between machines. These are of such varying length and width according to the individual case, that no power requirement can be specified.	
Gang mill, for sawing entire log into boards at one passage through the machine. Two opposite slabs are taken off the log by the head band saw, giving the log two flat faces, after which it is fed to the gang	50 to 400 hp.
Edger, for ripping bark edges from the boards and squaring them	15 to 250 hp.
Slab slasher, for dividing waste stock into short lengths for lath, stovewood, or other by-products	20 to 75 hp.
Trimmer, for cutting ends off boards and making them of standard length	20 to 75 hp.
Swing cut-off saw, for hand-trimming of large timbers, etc.	10 to 25 hp.
Timber sizer, for surfacing two or four sides of timbers at one operation	50 to 60 hp.

Lath bolter, for preparing lath bolts from slabs or other waste wood..	30 to 60 hp.
Lath machine, for making lath from bolts.....................	20 to 30 hp.
Hog, a grinder with knives on a revolving disk, for chipping waste wood for fuel or other purposes....................	25 to 150 hp.
Planers, of great variety in size and type. The largest standard size requires about 75 hp.	
Exhaust fans, for transporting dust and shavings from planers to fuel-storage house..................................	25 to 150 hp.
Chain conveyors, for transporting waste wood and sawdust, the length varying with the mill design up to about 350 ft.......	5 to 30 hp.

No. 1776

STATIC ADJUSTMENT OF TRUCKS ON CURVES

BY R. EKSERGIAN,[1] PHILADELPHIA, PA.
Non-Member

In the first part of this paper the adjustment of the running gear of a locomotive on a curve and the lateral reactions developed are considered from the geometrical limitations imposed which govern the length of wheelbase, the relation between lateral play and length of wheelbase, and the lateral displacement of trucks, together with a study of lateral devices for exerting guiding reactions between truck and the frame of the rigid wheelbase.

The author then considers the effect of the major lateral reactions developed between the various parts of the running gear. The effect of the centrifugal force and the overturning moment which it produces resulting in a change of load on the non-crossed part of the equalization is next discussed in order to arrive at the proper journal loads in the design of the driving- and trailing-truck axles. The principal reactions developed in a two and four-wheel engine truck and formulæ for the flange reaction of the guiding axle are developed in terms of the lateral reaction exerted at the center pin of the truck. An analysis is made of the lateral forces developed in skewing the rigid wheelbase of a locomotive in a curve. Finally formulæ for proper guiding and maximum flange reactions for the entire running gear are developed for three types of locomotives.

The third section is a recapitulation of the previous work with the object of keeping the various points outlined in working up a preliminary design.

IN the preliminary design of a locomotive we are primarily concerned with two major factors: (1) the tractive performance of the locomotive, and (2) the layout of the running gear. In the design layout of the running gear we must consider (1) the tractive requirements together with the type of drive and the foundation braking, (2) the weight or equalization distribution, and (3) the adjustment of the running gear for both tangent track and the various curves to be encountered together with the lateral reactions thereby introduced. The weight distribution and the adjustment of the running gear on curves are of very great importance

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when a heavy locomotive is to be designed for a medium roadbed with sharp curves. The tractive performance of locomotives, both steam and electric, is gradually reaching a satisfactory goal. The proper type of locomotive drive to employ, especially for electric locomotives, is more or less unsettled, but careful studies of this problem are already under way. With the development of the track-stress work performed lately by the Joint Committees of the American Society of Civil Engineers and American Railway Engineering Association, we may expect further attention as to the proper distribution of weight of the running gear of a locomotive. As to the adjustment of the running gear on tangents and curves, and the consideration of the lateral reactions thereby induced, due to its complexity and more or less necessary assumptions and the difficulty of introducing such data in design, this very important phase has been more or less neglected by the majority of railway designing engineers.

2 The object of this paper is mainly to introduce the subject of the action of the running gear both on tangents and curves. It is intended to be more suggestive than final, and it is hoped it will raise discussion as collective observations and experience are necessary to arrive at a more accurate conclusion on the various points discussed. Further, the subject of the preliminary analysis is rather from a qualitative than a quantitative point of view. The work covers the principal elements in the design of the running gear as adapted to traversing plane curves. The final design arrived at must necessarily be a compromise between the best arrangement of the running gear for traversing curves and for securing stability on tangents at high speeds. The latter aspect is a very complex subject in itself and can be to a considerable extent differentiated from the former. It was therefore deemed advisable to consider this phase in a separate paper at a later date.

3 In the adjustment of a locomotive running gear on a curve we shall consider the following four subdivisions:

a The geometrical limitations imposed for a given curve with a proper wheelbase and the relation between lateral play and length of wheelbase, etc.

b A study of the nature of the lateral reactions induced between the various elements of the running gear for various wheel arrangements and curves

c The elementary requirements in the design, layout and arrangement of the wheelbase, together with the proper type of guiding trucks, etc.

d The effect of the lateral reactions on the running gear as to strength of axles, etc.

4 In the configuration of a locomotive running gear on a curve we have two conditions, the statical and dynamical adjustments. At low and medium speeds the effect of centrifugal force is small as compared with the friction forces developed between the driving wheels and rail and thus the locomotive takes the statical adjustment. At high speeds the centrifugal forces cause the entire wheelbase to come more or less in contact with the outer rail. The dynamical adjustment therefore requires less flange play, and a longer wheelbase can be made to traverse a sharp curve without binding than

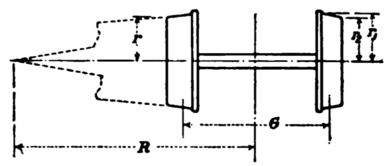

FIG. 1 SINGLE AXLE

in its statical adjustment. It is for this reason only necessary to consider the statical adjustment in a curve.

I GEOMETRICAL LIMITATIONS OF RUNNING GEAR ON A CURVE

5 *Single Axle.* For a single axle to traverse a curve radially it is necessary that the wheels should be theoretically coned. If, as shown in Fig. 1,

r_1 = radius of larger diameter of tread
r_2 = radius of smaller diameter of tread
R = radius of curve
G = gage of track
r = mean radius of wheel = $(r_1 + r_2)/2$
y = lateral displacement of center of axle from center of gage
$1/n$ = taper of coned wheel
ϕ = angle turned by axle

then
$$\frac{r_1\phi}{R+\frac{G}{2}} = \frac{r_2\phi}{R-\frac{G}{2}}, \quad \text{where} \quad r_1 = r + \frac{y}{n} \text{ and } r_2 = r - \frac{y}{n}$$

and therefore
$$R = \frac{G(r_1 + r_2)}{2(r_1 - r_2)} = \frac{G}{2}\left(\frac{rn}{y}\right) \quad \ldots\ldots\ldots [1]$$

which determines the proper radius of curvature for a given coning of the wheel or vice versa. In practice, however, due to wear it is impossible to maintain a perfectly coned wheel, and further, with rigid wheelbases the axles are constrained from assuming a purely radial direction. The coning of wheels reduces slippage and gives an ideal radial traversing on a curve for free axles, that is, axles not constrained in a rigid wheelbase. Since the lateral play of the axle is fixed and equal to the flange play, the taper of coning should theoretically increase with the degree of curve. Therefore the proper coning of a wheel can at best be but a compromise for the various curves to be encountered.

6 *Truck with Pair of Axles.* The axles now become partly constrained by the reaction of the truck frames. The displacement

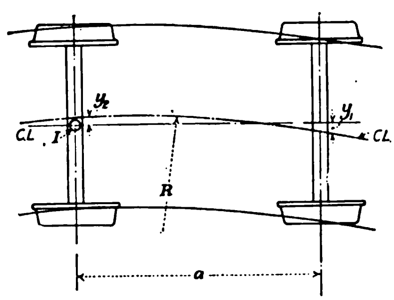

FIG. 2 TRUCK WITH PAIR OF AXLES

of a truck in a curve consists of a radial displacement of the center of gravity of the truck toward the center of curvature together with an angular displacement about the center of gravity of the truck. The combination of these two motions may be resolved into a single rotation about an instantaneous center I. (See Fig. 2.) Thus with a two-axle truck the instantaneous axle I will be assumed at the rear axle and the rear axle will be assumed radial and its normal tangent to the curve. If, then, as in Fig. 2,

a = distance between axles, i.e., length of rigid wheelbase, ft.
y_1 = outward lateral displacement of leading axle, ft.
y_2 = inward lateral displacement of second axle, ft.
R = radius of curve, ft.,

then from geometrical considerations

$$R^2 = [R - (y_1 + y_2)]^2 + a^2 \quad . \; . \; . \; . \; . \; . \; . \quad [2]$$

which simplifies approximately to

$$y_2 = \frac{a^2}{2R} - y_1 \quad \text{[3]}$$

When $y_2 = y_1$, the flange of the second axle just touches the inner rail and the maximum rigid wheelbase is determined as follows: $y = y_2 = y_1 = a^2/4R$; hence $a = 2\sqrt{Ry}$, where the value y obviously represents one-half the total lateral flange play of the given two axles.

7 *Rigid Wheelbase with Three Axles.* Assuming the instantaneous center of rotation between the second and third axles, we have, if Z = distance of I from the rear axle, and y_0 = lateral displacement of I from the center of the gage, as shown in Fig. 3,

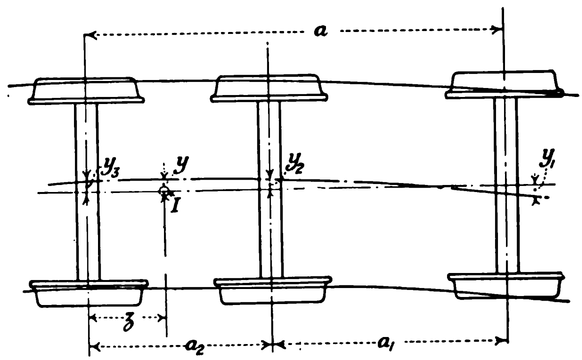

FIG. 3 TRUCK WITH THREE AXLES

$$y_0 = \frac{(a - Z)^2}{2R} - y_1 \quad \text{[4]}$$

$$y_0 = \frac{Z^2}{2R} + y_3 \quad \text{[5]}$$

If the lateral plays of the axles are assumed equal, then $y = y_1 = y_3$ and $Z = (a^2 - 4Ry)/2a$. If then Z is known and a lateral play y allowed for, we may determine the maximum allowable rigid wheelbase a. Further, since $a = Z + \sqrt{Z^2 + 4Ry}$, the greater the value Z, the longer the allowable wheelbase.

8 If a_1 = distance between first and second axles, and a_2 = distance between second and third axles, we have

$$y_0 = \frac{(a_2 - Z)^2}{2R} + y_2 \quad \text{[6]}$$

hence

$$y_2 = y_0 - \frac{(a_2 - Z)^2}{2R} = y - \frac{a_2^2 - 2a_2Z}{2R} \quad \text{[7]}$$

If the flange play of middle axle is the same as that of the front and rear axles, then $Z = \frac{1}{2}a_2$ and the flange of the middle wheel touches the inner rail. A greater value of Z requires that $y_2 > y = y_1 = y_3$. Thus a rigid wheelbase of three axles may be increased by giving greater lateral flange play of the middle driver over that of the front and rear drivers. For this reason the middle drivers for locomotives traversing sharp curves often have flangeless or cylindrical tires. It is, however, often better to secure the necessary lateral play by journal instead of additional flange play and thus retain the flanges on intermediate drivers. Crank intermediate axles are necessarily limited in journal play and therefore additional flange play must be given.

9 When the flange plays of the three axles are equal,

$$a = \frac{a_2}{2} \pm \sqrt{\frac{a_2^2}{4} + 4Ry} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

In the particular case when the intermediate driver is midway between the front and rear drivers, $a_2 = \frac{1}{2}a$, and $a = 2\sqrt{2}\sqrt{Ry}$. Thus wheelbases in which both the intermediate and rear driver flanges come in contact with the inner rail simultaneously, have an effective wheelbase $\sqrt{2}$ times greater than when only the rear-axle flange comes in contact with the inner rail.

ROY METHOD FOR THE GEOMETRICAL LAYOUT OF THE RUNNING GEAR

10 The advantage of the Roy Method is that the wheelbase and clearances can be plotted on a drawing board by the following simple graphical method: If the total flange play $= S = 2y$, (Fig. 4). then we may draw two circles of radius R and $(R - S)$, respectively, Since the flange of the inner driver just touches the inner rail, we draw a line tangent to the inner circle to the intersection of the outer circle. Obviously the length from the tangency of the inner circle to the intersection of the outer circle, represents the allowable wheelbase a. From simple geometry

$$a^2 + (R - S)^2 = R^2 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [9]$$

hence

$$a^2 = S(2R - S) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [10]$$

But since S is small compared with R and a,

$$a^2 = 2RS, \text{ or } S = \frac{a^2}{2R} . \; . \; . \; . \; . \; . \; . \quad [11]$$

11 Using now a uniform scale and S_s, a_s and R_s for S, a and R, if we make $a_s = a/n$ and $R_s = R/n^2$ in actual length, then

$$S_s = \left[\frac{\frac{a^2}{n^2}}{2\frac{R}{n^2}}\right] = \frac{a^2}{2R} \quad \ldots\ldots\ldots\ldots [12]$$

and hence $S_s = S$. We thus have the total flange play full size, which is especially convenient since the flange play is always small. Since the angles are functions of the ratios a_s/R_s, or S_s/a_s, we have

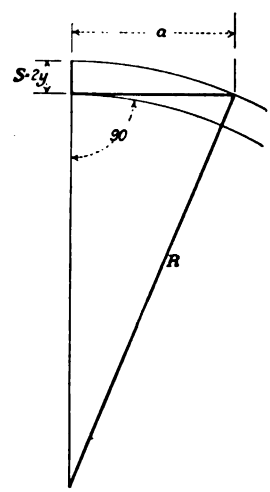

FIG. 4 ROY METHOD FOR THE GEOMETRICAL LAYOUT OF RUNNING GEAR

$$\frac{a_s}{R_s} = \left[\frac{\frac{a}{n}}{\frac{R}{n^2}}\right] = n\frac{a}{R} \quad \ldots\ldots\ldots\ldots [13]$$

and

$$\frac{S_s}{a_s} = \frac{S}{\frac{a}{n}} = n\frac{S}{a} \quad \ldots\ldots\ldots\ldots [14]$$

that is, the angles on a layout are n times their actual value. In an actual layout it is convenient usually to further reduce S_s, R_s and a_s by $\frac{1}{2}$, and for n we take a value between 8 and 12.5.

12 The total flange play S is the sum of the flange play on a tangent track plus the widening of the gage on a curve. From the Roy Method we are able to tell at a glance the configuration of the

running gear in the curve and the necessary plays that must be allowed for in order that the engine may properly traverse the curve. In the layout of the running gear in a curve by the Roy Method, when not constrained by binding in the curve the rear axle of the rigid wheelbase should be drawn as nearly radially as possible. In other words, the rear axle of the rigid wheelbase is assumed to move tangentially to the curve. Obviously, the ideal wheelbase is when the inner flange of the rear axle is brought just in contact and tangent to the inner rail. It is to be observed, however, that a much longer wheelbase will traverse the curve without complete binding. By giving sufficient journal play in the frame or flange play a very long wheelbase can be made to traverse a very sharp curve. With short wheelbases not only is the flange guiding force on the front driver of the rigid wheelbase increased, but there is introduced the possibility of violent lateral oscillations.

LATERAL SWING OF TRUCKS AND LENGTH OF RADIUS BAR

13 Both for clearance limitation and the design of the hanger, for producing the desired lateral constraint between truck and frames it is important to estimate accurately the swing of a truck for a given-degree curve. Referring to Fig. 5, let

a = length of rigid wheelbase, ft.
b = distance from front axle to center pin of truck, ft.
d = required swing of center pin of truck, ft.
Z = distance to instantaneous center of wheelbase from rear axle of rigid wheelbase, ft.
r = length of radius bar, ft.
R = radius of curvature, ft.
y = one-half total lateral flange play of axle of rigid wheelbase assumed same for each axle, ft.

Then, assuming a rigid wheelbase with an intermediate driver,

$$\frac{(a + b - Z)^2}{2R} - (d + y) = \frac{Z^2}{2R} + y \quad . \quad . \quad . \quad . \quad . \quad [15]$$

By the previous analysis

$$2y = \frac{a^2 - 2aZ}{2R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [16]$$

Substituting this value in Equation [15],

$$d = \frac{2(a - Z)b + b^2}{2R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

and from elementary geometry

$$r = \frac{dR}{(a - Z) + b} \quad \ldots\ldots\ldots \quad [18]$$

When $Z = 0$, as would occur when the inner flange of the rear driver is assumed tangent with the inner rail or in a two-axle rigid wheelbase,

$$d = \frac{2ab + b^2}{2R} = \frac{(a + b)^2 - a^2}{2R} \quad \ldots\ldots \quad [19]$$

therefore

$$r = \frac{2(a - Z)b + b^2}{2(a - Z + b)} \quad \ldots\ldots\ldots \quad [20]$$

and for its maximum value,

$$r = \frac{(a + b)^2 - a^2}{2(a + b)} \quad \ldots\ldots\ldots\ldots \quad [21]$$

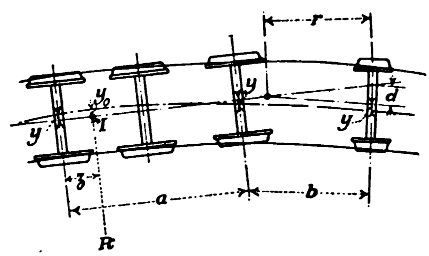

FIG. 5 DIAGRAM SHOWING LATERAL SWING OF TRUCK WITH RADIUS BAR

We thus see the length of radius bar is independent of the radius of curvature. Further, since Z depends upon the wear of the flanges and play, the ideal or theoretical radius must vary with the service of the engine. It is therefore impossible to place a radius bar that will give ideal guiding of the truck for all conditions of service, and we must compromise by experience for the proper length of bar to be used.

14 Common formulæ for the swing of a truck and the length of radius bar are based on the assumption that the wheelbase center line intersects the front and rear axles at the center line of the track. In other words, Z is assumed as equal to $\frac{1}{2}a$. This condition can only take place when the centrifugal forces on the running gear are sufficient to overcome the friction forces between the various driving wheels and rail and slide the entire wheelbase against the outer rail.

LATERAL DISPLACEMENT OF A TRAILING TRUCK

15 The swing of a trailing truck relative to the frame depends upon the location of the truck axle with respect to the intersection

of the longitudinal center line of the wheelbase and the center line of the truck. When the location of the truck axle is between the rear axle and a point beyond this intersection where the lateral displacement equals the flange play of the truck, the trailing truck floats between the rails. Beyond this point the outer flange is brought in contact with the outer rail. From the rear axle to the point of intersection there is a more or less tendency of the truck to hug the inner rail. From the following analysis it will be especially noted that ordinarily the trailing-truck outer flange is seldom brought in contact with the outer rail, since for this condition the trailing axle must be necessarily spaced a considerable distance back from the rear axle of the rigid wheelbase. Thus we would expect the flange wear on trailing wheels to be negligible. In Fig. 6, let

a = length of rigid wheelbase, ft.
c = distance from rear axle to center line of trailing axle, ft.
d = swing of center of axle relative to frame, ft.
Z = distance of instantaneous center of wheelbase from rear axle of rigid wheelbase, ft.
y = lateral flange play in axle, ft.

Then

$$\frac{(Z+c)^2}{2R} - (d+y) = \frac{(a-Z)^2}{2R} - y \quad . \quad . \quad . \quad . \quad . \quad [22]$$

from which

$$d = \frac{c^2 - a^2}{2R} + \frac{(a+c)Z}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

16 When $d + y = 0$, the point of intersection of the longitudinal center line of rigid wheelbase and the center line of track is given by

$$C = -Z \pm \sqrt{Z^2 + (a^2 - 2aZ - 2yR)} \quad . \quad . \quad . \quad . \quad . \quad [24]$$

For the outer flange to come in contact with the outer rail, the trailing axle must be placed back from the rear axle of the rigid driving wheelbase at least a distance

$$C = -Z \pm \sqrt{Z^2 - a(2Z - a)} \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

17 When $Z = 0$, as would occur with a four-wheel rigid wheelbase or with a three-axle rigid wheelbase when the intermediate driver is not brought in contact with the inner rail,

$$d = \frac{C^2 - a^2}{2R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [26]$$

18 Obviously when $d = 0$, $c = a$, and therefore for contact of the outer flange of the trailing truck with the outer rail, the trailing axle must be spaced from the rear axle of the rigid wheelbase a distance equal to the length of rigid wheelbase. In practice, due to more or less binding of the wheelbase or due to centrifugal reactions at high speeds, the rear axle of the rigid wheelbase is brought closer in contact with the outer rail. In such a case there is a tendency for the trailing-truck outer flange to run against the outer rail.

LATERAL PLAY OF INTERMEDIATE DRIVERS IN A RIGID WHEELBASE

19 A considerable part of the flange-guiding reaction on the front driver of a rigid wheelbase or the lateral force of a guiding truck is utilized in overcoming the friction due to lateral sliding of the middle or intermediate drivers. It is therefore good design to give sufficient lateral journal play in order that the outer flange

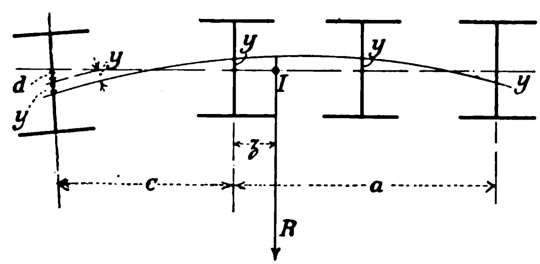

FIG. 6 DIAGRAM OF TRAILING TRUCK WITH RIGID WHEELBASE

may strike the outer rail, thus guiding the axle by its own flange reaction and relieving the lateral reaction on the frame of the rigid wheelbase which in turn reduces the lateral reaction on the leading driving-wheel flange. The necessary play may be arrived at by noting that the intermediate axle must be displaced outward from the center line of the track a distance equal to the flange play y. Now the center line for a three-axle rigid wheelbase at the intermediate axle is displaced inward from the center line of the track a distance

$$y_2 = y - \frac{a_2^2 - 2a_2Z}{2R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [27]$$

Hence the necessary journal play to relieve the lateral reaction on the flange for an intermediate driver, becomes

$$e = 2y - \frac{a_2^2 - 2a_2Z}{2R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [28]$$

LATERAL DEVICES BETWEEN ENGINE TRUCK AND RIGID-WHEELBASE FRAME

20 Due to the lateral displacement of the center pin of the truck from the center line of the rigid wheelbase while an engine traverses a curve and the necessity of a guiding force on the rigid wheelbase, some device must be introduced between the truck and frame of the rigid wheelbase that offers a lateral resistance which is a definite function of the lateral displacement of the truck. For this purpose the following devices are used (see Fig. 7):

a Ordinary swing links or two-point hangers, the lateral resistance increasing from zero with the displacement

b Inclined hangers consisting of two swing links given an initial inclination usually swung outwardly at the bottom, the lateral resistance being a differential action of the two and increasing from zero with the displacement

c The heart link or three-point hanger, the resistance increasing from an initial value with the displacement

d An inclined plane with slides, rollers or rockers rolling along the plane and giving a constant resistance quite independent of the lateral displacement of the truck.

Other types of centering devices such as springs, inverted three-point hangers, etc., have been used but the above are those most commonly found in practice.

21 Neglecting the friction at the hanger bearings we have for the lateral reaction in type (*a*), Fig. 7,

$$Q = \frac{Wd}{\sqrt{L^2 - d^2}} \quad \ldots \ldots \ldots \quad [29]$$

where

W = center-pin load, lb.
d = lateral displacement of center pin, in.
L = length of hanger, in.

22 With inclined hangers the links are usually displaced outward at the bottom a distance d_0 when in the central position. The total lateral reaction is a differential action of the two simple hangers which have initial displacements $(+ d_0)$ and $(-d_0)$, respectively. The lateral reaction becomes

$$Q = W\left[\frac{(d + d_0)}{\sqrt{L^2 - (d + d_0)^2}} - \frac{d - d_0}{\sqrt{L^2 - (d - d_0)^2}}\right] \quad \ldots \quad [30]$$

23 With a heart link or three-point hanger the turning takes place about either of the pins of the hanger at the top. If

h = perpendicular distance from a line through the centers of the upper pins to the center of the bottom pin, in.

S = distance between the upper pin centers, in.

then for a lateral displacement d,

$$Q = \frac{W\left(\frac{S}{2} + d\right)}{\sqrt{h^2 - d(S + d)}} \quad \ldots\ldots\ldots\ldots \quad [31]$$

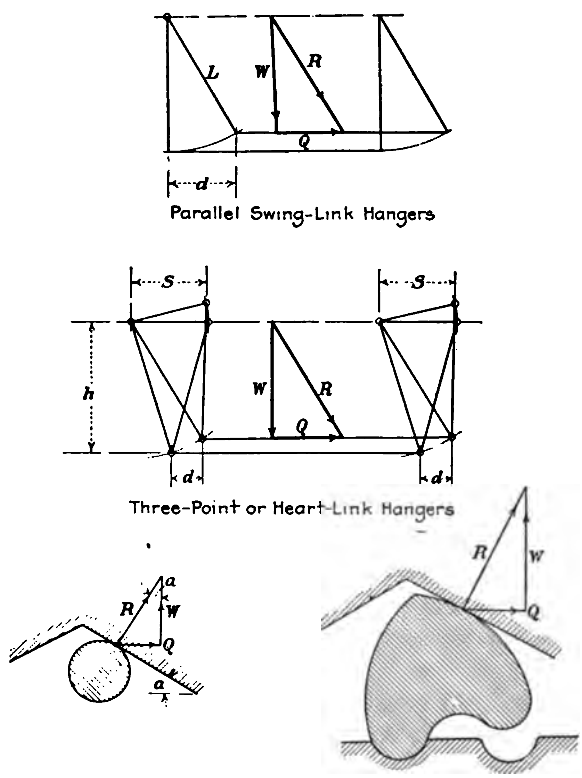

Fig. 7 Lateral Devices

and the initial lateral reaction becomes

$$Q_i = W\frac{S}{2h} \quad \ldots\ldots\ldots\ldots \quad [32]$$

24 With an inclined plane supported by rollers or rockers, neglecting friction in a first approximation, we have the resultant

reaction normal to the plane, the vertical component balancing the center-pin load W_1 while the horizontal component gives the required lateral reaction Q. Therefore the magnitude of the lateral reaction depends on the inclination of the plane with the horizontal, in which case

$$Q = W \tan a \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [33]$$

and is independent of the lateral displacement of the truck. Trucks using this device are known as "constant-resistance" trucks. The type of lateral device to be used for guiding depends upon a careful consideration of the lateral forces developed when the engine rounds a curve. Analysis should be made of the required lateral force for various degrees of curvature. In this way we can arrive at the proper function of the lateral reaction as against swing of truck or degrees of curvature.

LOCATION OF CENTER PIN OF A HINGE JOINT BETWEEN TWO ENGINE RUNNING GEARS

25 By the use of a hinge joint two running gears may be effectively connected together and the lateral reactions on either unit thereby decreased. That is, the lateral reaction at the hinge aids in guiding either unit. To do this the hinge center pin must be located in advance, i.e., in the direction of motion of the common intersection of the center line of the two running gears. Therefore it is important to arrive at the approximate intersection of the center lines of the adjacent wheelbases. In Fig. 8 let

a = length of front wheelbase, ft.
b = length of rear wheelbase, ft.
z = distance from rear axle of front unit to instantaneous center of rotation of front unit, ft.
z_1 = distance from rear axle of rear unit to instantaneous center of rotation of rear unit, ft.
c = distance from rear axle of front unit to intersection of center lines, ft.
d = distance from front axle of rear unit to intersection of center lines, ft.

Then from elementary geometrical considerations,

$$(c + z)^2 + (R - y_0)^2 = (d + b - z_1)^2 + (R - y_0')^2 \quad . \; . \quad [34]$$

Since y_0 and y_0' are sensibly equal and negligible as compared with R, we have to a close degree of approximation, $c + z = d + b - z_1$.

But we have seen that

$$z = \frac{a^2 - 4Ry}{2a} \quad \text{and} \quad z_1 = \frac{b^2 - 4Ry}{2b} \quad . \; . \; . \; . \; . \quad [35]$$

where y = common lateral flange play of the individual axles. If we now substitute these values in the previous equation,

$$c - d = \frac{b - a}{2} + 2Ry\left(\frac{1}{a} + \frac{1}{b}\right) \quad . \; . \; . \; . \; . \; . \; . \quad [36]$$

But

$$c + d = L \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [37]$$

hence

$$c = \frac{L}{2} + Ry\left(\frac{1}{a} + \frac{1}{b}\right) - \frac{a - b}{4} \quad . \; . \; . \; . \; . \; . \quad [38]$$

and

$$d = \frac{L}{2} - Ry\left(\frac{1}{a} + \frac{1}{b}\right) + \frac{a - b}{4} \quad . \; . \; . \; . \; . \; . \quad [39]$$

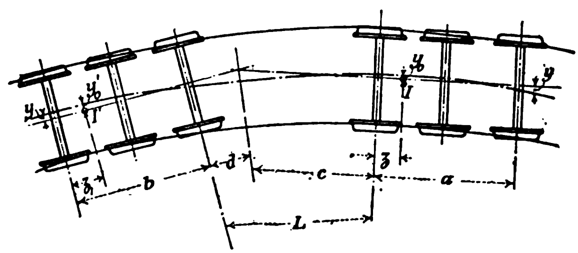

FIG. 8 DIAGRAM SHOWING LOCATION OF HINGE JOINT

the latter values being of sufficient approximation for ordinary design. If the center pin of the hinge joint between the two units is located so that $c < \frac{1}{2}(L + b)$ and $d > \frac{1}{2}(L - b)$, we have the proper guiding effect on the two units. In a two-way locomotive it is desirable to locate the center pin midway between the two units. Since in general it is in advance of the direction of running either way of the common intersection of the center lines of the two adjacent wheelbases, we have effective guiding in running either direction. In the location of the center pin of the hinge and in the determination of the limiting distance between the front axle of the rear rigid wheelbase and the rear axle of the front wheelbase, care must be taken that neither unit binds on the maximum curvature.

26 If we make $y = 0$ in Equations [38] and [39] for c and d, there will be sufficient guiding for either wheelbase and

$$c = \frac{L}{2} - \frac{a - b}{4} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [40]$$

$$d = \frac{L}{2} + \frac{a - b}{4} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [41]$$

Obviously the best location of the hinge is between these and the former values.

II LATERAL REACTIONS IN THE RUNNING GEAR

27 The configuration of the running gear consists of a rigid wheelbase with guiding trucks. The lateral reaction of a guiding truck on the main frame assists in the angular rotation of the wheelbase about the friction center of the rigid wheelbase. The proper guiding can only be ascertained from analysis of the various reactions brought on the rigid wheelbase together with the friction moments between the driving wheels and rails. The proportioning of the flange reactions for the guiding truck and leading driver of a rigid wheelbase is a matter of especial importance in that it determines the wear on the corresponding flanges. The effect of centrifugal force with low-degree curves and ordinary speeds is small as compared with the lateral reaction required to skew the wheelbase against the friction between the wheels and rail, and therefore for a first approximation for slow-speed engines we may neglect the effect of centrifugal force. At high speed with sharp curves the centrifugal force of the running gear and cab must be considered together with the various other forces acting.

CENTRIFUGAL FORCE ON A CURVE

28 In Fig. 9 let

D = degree of curve
R = radius of curve, ft., where $R = 5730/D$, approx.
w = total weight, lb.
C = centrifugal force, lb.
G = gage of track, in.
e = elevation of outer rail, in.
g = acceleration due to gravity, ft. per sec. per sec.

Then

$$C = \frac{w}{g}\frac{v^2}{R} \qquad [42]$$

where

$$v = \frac{\text{m.p.h.} \times 5280}{3600} \text{ ft. per sec.}$$

and

$$R = \sqrt{C^2 + w^2}$$

Dividing R into two components S_y and S_z parallel and normal to the track, we have,

$$S_y = R \sin (B - a) = \sqrt{C^2 + w^2} \sin (B - a) \quad . \; . \; . \quad [43]$$

$$S_z = R \cos (B - a) = \sqrt{C^2 + w^2} \cos (B - a) \quad . \; . \quad [44]$$

where

$$a = \tan^{-1} \frac{e}{G} \text{ and } B = \tan^{-1} \frac{C}{w}$$

hence

$$S_y = w \sin (B - a) \sqrt{1 + \frac{v^4}{g^2R^2}} \quad . \; . \; . \; . \; . \quad [45]$$

$$S_z = w \cos (B - a) \sqrt{1 + \frac{v^4}{g^2R^2}} \quad . \; . \; . \; . \; . \quad [46]$$

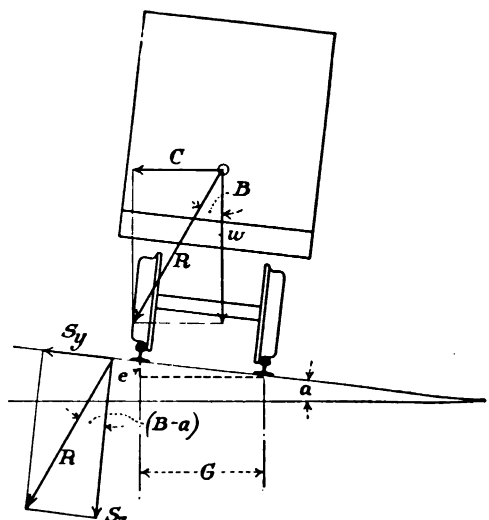

FIG. 9 DIAGRAM OF CENTRIFUGAL COMPONENTS

When $S_y = 0$, we have the theoretical banking of the track, so that the normal reaction S_z gives a component in the horizontal plane that balances the centrifugal effect. When $S_y = 0$, $\sin (B - a) = 0$ and therefore $e/G = C/w$, and $v^2 = Rge/G$ and the value of v becomes

$$v = \sqrt{\frac{32.16\ Re}{G}} = \sqrt{\frac{32.16 \times 5730\ e}{GD}} \text{ ft. per sec.} \quad . \; . \quad [47]$$

For standard gage $G = 56.5$ in., hence $v = 57\sqrt{\frac{e}{D}}$ ft. per sec. or in miles per hour,

$$v = 38.8\sqrt{\frac{e}{D}}\ \text{m.p.h.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [48]$$

The centrifugal effect may be neglected when the actual speed is close to the theoretical speed.

29 When a locomotive enters a curve we have a complicated dynamic action. Due to the flange, journal and box plays the adjustment to the curve is not immediate, the several plays necessarily having to be taken up. We thus have more or less of an impact lateral action and the magnitude of the lateral force is a function of the speed on entering the curve and depends as well upon the distribution of masses of the engine. The lateral reaction at the rail is decreased with a high center of gravity of the total masses of the engine, since greater transverse angular motion is possible. On the other hand, large lateral oscillations must be limited, consistent with the transverse stability of the locomotive. The general introduction of spiral curves has greatly reduced the impact effect on entering a curve.

30 It is of interest, however, to consider the transition of a long cab with center pins on either end, as in an electric locomotive, from a tangent directly into a plane curve, with the lateral plays neglected. The center-pin lateral reaction thus obtained can then be multiplied by a suitable impact factor for its maximum lateral reaction. On entering a curve let us assume a long cab with center pins at a distance L apart. (See Fig. 10.) Then the angle turned by cab $= y/L = \theta$. But $y = R(1 - \cos\phi)$, hence

$$\theta = \frac{y}{L} = \frac{R}{L}(1 - \cos\phi) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [49]$$

and for the angular velocity

$$\frac{d\theta}{dt} = \frac{R}{L}\sin\phi\frac{d\phi}{dt} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [50]$$

and for the angular acceleration

$$\frac{d^2\theta}{dt^2} = \frac{R}{L}\cos\phi\left(\frac{d\phi}{dt}\right)^2 - \frac{R}{L}\sin\phi\frac{d^2\phi}{dt^2} \quad . \quad . \quad . \quad . \quad . \quad [51]$$

But $\cos\phi = 1$ (approx.) and $\frac{d^2\phi}{dt^2} = 0$ (approx.); further, let $\omega = \frac{d\phi}{dt}$; then

$$\frac{d^2\theta}{dt^2} = \frac{R}{L}\omega^2 = \frac{v^2}{RL} \text{ (approx.) radians per sec.}^2 \quad . \; . \; . \quad [52]$$

31 Assuming a cab of uniformly distributed weight of dimensions a and b (see Fig. 10), we have for the moment of inertia about a vertical axis through the center of gravity

$$I_G = \frac{w\,(a^2 + b^2)}{12} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [53]$$

and about the rear center pin

$$I = \frac{w(a^2 + b^2)}{12} + \frac{wL^2}{4} = \frac{w}{4}\left(\frac{a^2 + b^2}{3} + L^2\right) \quad . \; . \; . \; . \quad [54]$$

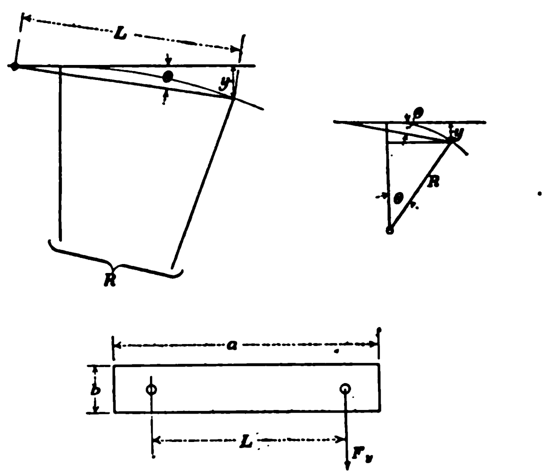

FIG. 10 DIAGRAM SHOWING CENTRIFUGAL EFFECT OF LONG CAB ON ENTERING CURVE

Since the mass distribution in a cab is seldom uniform but bunched in the center,

$$I = k\,\frac{w}{4}\left(\frac{a^2 + b^2}{3} + L^2\right) \quad . \; . \; . \; . \; . \; . \; . \; . \quad [55]$$

where $k = 0.6$ to 0.8, roughly. Taking moments about the rear center pin, we have, for the lateral reaction on the front center pin,

$$F_v = \frac{kw}{4g}\left(\frac{a^2 + b^2}{3} + L^2\right)\frac{v^2}{RL^2}, \text{ lb.} \quad . \; . \; . \; . \; . \; . \; . \quad [56]$$

This equation includes both the effect of centrifugal and the angular inertia of the cab on entering a curve. It may be put in another and perhaps more convenient form, as follows:

32 If λ_y = radius of gyration about the center of gravity, λ = radius of gyration about center pin, then, for cab length L,

$$I = w\left(\lambda_y^2 + \frac{L^2}{4}\right) = w\lambda^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [57]$$

and

$$F_y = \frac{w}{g}\left(\lambda_y^2 + \frac{L^2}{4}\right)\frac{v^2}{RL^2} = \frac{w}{g}\lambda^2\frac{v^2}{RL^2} \quad . \quad . \quad . \quad . \quad . \quad [58]$$

Now $\frac{\lambda}{L}$ = 0.7 (approx.) for a properly designed cab, hence

$$F_y = \frac{1}{2}\frac{w}{g}\frac{v^2}{R} \text{ (approx.) lb.}$$

To allow for impact the value F_y should be increased to 1.25 F_y which is also the value of the lateral center-pin reaction after the cab has entered the curve.

LOCATION OF THE PRINCIPAL MASSES OF A LOCOMOTIVE

33 Let

W = total weight of engine or weight of one running gear and half of cab as in electric locomotive with cab supported on two running gear, lb.

W_s = weight of spring-supported parts, lb.

W_t = weight of trucks and drivers, lb.

Z_g = height of center of gravity of total locomotive above rail, ft.

Z_{gs} = height of center of spring-borne parts above rail, ft.

Z_{gt} = height of center of gravity of the trucks and axles of the running gear above the rail.

Then $W - W_t = W_s$, and the value of Z_{gt} may be readily obtained by estimating the height of the various driving axles and the center of gravity of the component guiding trucks, or

$$Z_{gt} = \frac{\Sigma W_t Z_t}{W_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [59]$$

The center of gravity of the total locomotive may be assumed to be some given distance above the rail, the value depending of course on the type of locomotive under consideration. The center of gravity of the spring-borne parts may then be determined from the equation

$$Z_s = \frac{ZW - Z_{gt}W_t}{W_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [60]$$

34 In estimating the guiding forces on the rigid wheelbase it is of importance to locate the center of gravity of the rigid wheelbase

and parts attached thereto. In a steam locomotive we may include the total mass of the engine minus the trucks since the latter, do not form a rigid *lateral* system with the rigid wheelbase. In an electric locomotive, which has a heavy cab supported on two engine gears, at either end of cab, the cab does not form a rigid *lateral* system with respect to either rigid wheelbase. Therefore we have external lateral reactions exerted by the guiding trucks and cab, respectively. It is for this reason important to locate the center of gravity of the running gear separately from the trucks and cab, respectively.

35 Let

W = total weight of engine minus guiding trucks for steam locomotive or total weight of half engine unit minus trucks and cab load as in an electric locomotive, lb. = rigid wheelbase mass.
C = center-pin load or resultant load of cab on running gear
T = weight of a guiding truck
N = normal rail axle load.

Also let e, d, d' and L equal the moment arms to the center of gravity of W, C, T and N from the front axle of the leading truck, respectively. Then

$$\Sigma NL - Cd - \Sigma Td' - We = 0 \quad \ldots\ldots \quad [61]$$

and

$$W = \Sigma N - \Sigma T - C \quad \ldots\ldots \quad [62]$$

hence

$$e = \frac{\Sigma NL - Cd - \Sigma Td'}{W} \quad \ldots\ldots \quad [63]$$

Thus, knowing the rail loads and their distribution, the resultant cab load and its line of action and the weights and moment arms to the center of gravity of the several guiding trucks, we may readily estimate the location of the center of gravity of the rigid wheelbase mass.

EFFECT OF CENTRIFUGAL FORCE ON THE RUNNING GEAR

36 The spring-borne parts of a locomotive are constrained laterally by the box journal reactions of the driving axles and the lateral reactions of the guiding trucks. Assuming given lateral guiding reactions of the trucks Q_1, Q_2, etc., we have for the required total lateral reaction of the driving boxes,

$$\Sigma Q_a = S_y - \Sigma Q = \frac{W_2}{g}\frac{V^2}{R} - \Sigma Q \text{ (approx.)} \quad \ldots \quad [64]$$

The lateral reactions ΣQ_a and ΣQ together with S_y form a couple acting on the spring-borne parts and increase the equalizer load on one side over that on the other. The balancing couple due to a change in load on the equalizer system on one side over that on the other takes place entirely on the equalizer system *not* cross-equalized. Let

d_q and d_a = vertical distance between the center of gravity of the spring-borne parts and the lateral reactions Q and Q_a, respectively

Φ = resultant reaction on one side of the non-crossed equalizer system

L_Φ = lateral distance between the equalizer system not cross equalized

$\triangle \Phi$ = resultant change in load of the equilization.

Then

$$\triangle \Phi L_\Phi = \Sigma Q_a d_a + \Sigma Q d_q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [65]$$

or

$$\triangle \Phi = \frac{\Sigma Q_a d_a + \Sigma Q d_q}{L_\Phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [66]$$

where $\Delta \Phi$ represents the resultant change in load of the equalization. To obtain the change in load on a given axle, we have

$$\Phi = \Sigma W_e + \Sigma k W_e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [67]$$

where the resultant equalizer reaction is divided into the various journal reactions W_e and kW_e. If an equalizer beam leading from a main journal has a ratio from fulcrum $= a/b$, then $k = a/b$. Hence for a change of load on a given axle, we have

$$\triangle(kW_e) = \frac{kW_e}{\Phi} \triangle \Phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [68]$$

We are thus able to estimate the effect of the centrifugal force of the spring-supported parts on the various axle loads.

37 In a guiding truck the lateral reaction Q together with the lateral reaction of the journals form a couple increasing the journal load on one side over that on the other. Strictly speaking, we should also include the centrifugal effect of the truck itself in causing an additional overturning. Knowing the lateral reactions brought on the journals and flanges, and the change in the journal loads of the axle due to the centrifugal effect of the spring-borne parts, we may readily estimate the normal rail pressures and the lateral friction forces between base of wheel and rail.

NATURE OF THE GUIDING REACTION BETWEEN WHEEL FLANGE AND RAIL

38 The reaction between wheel and rail on a tangent truck may be divided into a normal and tangential component, the latter being negligible unless the wheel is braked or the wheel is a driver. The tangential component in no case can exceed nN, where n is the coefficient of friction and N the normal component. Where an axle enters a curve the wheel takes an oblique position with respect to the rail. Due to this obliquity, the flange contact is moved slightly ahead of the contact of wheelbase or wheel tread and rail. The reaction between rail and wheel is then divided into two separate reactions, a flange reaction and a tread reaction at the base of wheel and top rail. If it were not for friction, the flange reaction would be normal to the surface of contact between flange and rail. In Fig. 11, let

Y = lateral component of the flange reaction
X_1 = nosing or longitudinal component of the flange reaction
N_1 = vertical component of the flange reaction
P_1 = resultant reaction of the rail on flange
N_2 = vertical component of the tread reaction
X_2 = tangential component of the tread reaction.

From Fig. 11, if $d = \tan^{-1}n$, $n = 0.15$ to 0.25. The resultant reaction, P_1, makes an angle d with the normal. Since the relative motion of flange with respect to rail is downward, the effect of friction must cause P_1 to incline above the normal to the surface of flange contact and rail. If the normal is inclined with an angle B to the vertical, then, since X is always small,

$$Y = P_1 \sin (B - d) \text{ approx.} \quad [69]$$

and

$$N_1 = P_1 \cos (B - d) \text{ approx.} \quad [70]$$

hence

$$N_1 = Y \cot (B - d) \text{ approx.} \quad [71]$$

and further

$$X_1 = Y \sin a \text{ (approx.)} \quad [72]$$

where a = striking angle or angle of obliquity between wheel and rail.

39 The condition of the required weight on an axle for maximum guiding may be considered as follows. If N is the total vertical load on a wheel and the rail, then $N = N_1 + N_2$. Now as Y increases, N_1 and B increase as well. Since $\tan (B - d) = Y/N_1$, the angle B will remain constant so long as with an increase in Y, N_1

does not exceed N the total weight on rail. This will be the ordinary contact of flange and rail and takes place toward the bottom of the flange where B is of smaller value. As Y increases N_1 increases and B remains constant until $N_1 = N$. Further increase in Y causes B to increase to its maximum value. Hence for a given lateral reaction Y_0, the load of wheel at rail must be $N \geqq Y_0 \times \cot (\phi - d)$, or otherwise the wheel will leave the rail.

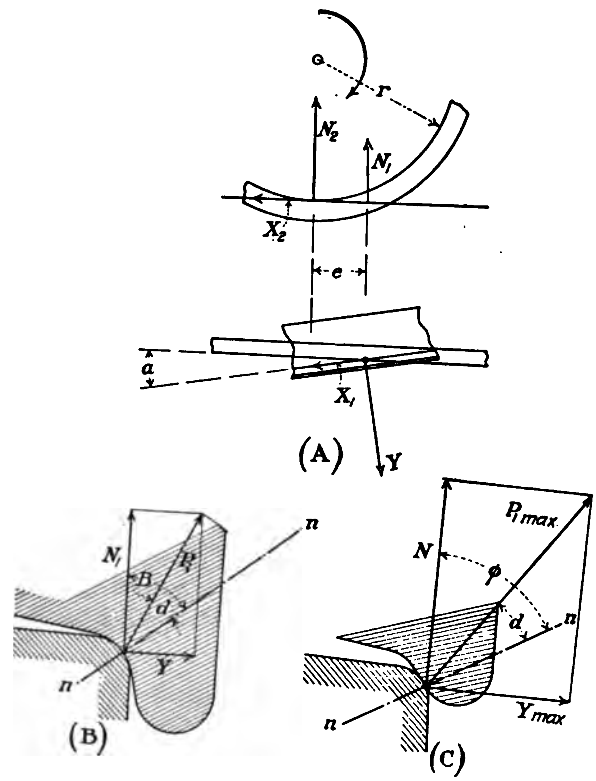

FIG 11 DIAGRAM OF GUIDING REACTION BETWEEN FLANGE AND RAIL

40 In practice the rail load should be considerably greater than this value. The angle ϕ of course varies for different conditions of wear between flange and rail, and a careful study of this point is of utmost importance when a minimum weight is to be placed on a truck for guiding. The stability from derailment depends greatly upon the contact point between flange and rail. The lower or inner portion of the flange is concave inward to the rail while the upper or outer portion is concave outward. The point of

inflection is obviously the point of contact between flange and rail where we obtain the maximum lateral reaction for a given weight on the wheel. In the concave region an increase in the lateral reaction Y results in the contact point approaching the point of inflection of the flange contour. Since with a given weight on the wheel the lateral reaction increases to this point, the concave part to the point of inflection is the stable portion of the flange, relatively speaking. From the point of inflection on the flange contour of the concave outward part, an increase in the lateral reaction Y results in a vertical component greater than the weight on the wheel, with consequent derailment. Therefore the concave outward part is the unstable portion of the contour of the flange.

41 The contact point of flange and rail depends upon the magnitude of the obliquity of the wheel with respect to the rail, that is, the angle between a skewed axle and the perpendicular to the rails. It is obvious that the greater this obliquity the nearer the contact point approaches the top edge of the flange. When the contact point is behind the point of inflection of the flange contour on its concave outward part, derailment is likely to result if the lateral reaction is appreciable. With lateral oscillations of the truck the lateral reaction may reach a very high value. It would appear important, therefore, to limit the striking angle or angle of obliquity between wheel and rail so that the contact point between flange and rail is well within the concave portion of the flange contour.

42 Further, since the area of flange and rail contact decreases with the angle of obliquity, the bearing pressure increases as some function of the angle of obliquity. The allowable maximum angle of obliquity between wheel and rail is therefore additionally limited by the allowable maximum bearing pressure. Continental practice limits the angle of obliquity or striking angle to 4 deg. The component X_1 offers a direct resistance to the motion of the wheel, and due to the flange contact being ahead of the tread of wheel and rail contact a distance e, we have a resisting couple $N_1 e$. This in turn causes a tangential component at the base of wheel tread and rail $= N_1 \frac{e}{r} - X_1$ where r = radius of the wheel. The total resistance to forward motion is therefore roughly

$$X = X_1 + (N_1 \frac{e}{r} - X_1) = Y \frac{e}{r} \cot (B - d) \text{ lb.} \quad . \quad . \quad . \quad [73]$$

Since d decreases and e increases with the obliquity of the wheel with respect to the rail, the resistance to forward motion is thereby increased with the wheel obliquity.

43 In considering the reactions on the running gear and axles, the point of contact of flange and rail will be assumed at the contact of wheel tread and rail. Further, the nosing reaction X_1 will be neglected unless the striking angle is very large. Hence at the wheel contact with rail we have simply two components, the normal component N and the lateral or transverse component Y.

LEADING- AND TRAILING-TRUCK ELEMENTS

44 For guiding the rigid wheelbase on a curve two types of leading trucks are in common use — (1) two-wheel or single-axle guiding trucks, and (2) four-wheel or two-axle guiding trucks. Trailing trucks are almost universally of a two-wheel type. Two-wheel trucks have radius bars reacting from a vertical pin to the axle. The reactions brought on a two-wheel truck are the center-pin load of the front equalizer and a lateral reaction of the center-pin bearing in the frame. The latter is caused by the swing of hangers or inclined plane, etc. In addition we must consider a small lateral reaction of the center pin of the radius bar due to the skewing of the truck on a curve. These reactions are balanced by the side thrust of the rail together with the friction at the rail due to skewing of the truck.

45 The approximate reactions on a two-wheel leading truck are shown in Fig. 12-A, in which

W = center-pin load
w = weight of truck
Q = lateral reaction due to the swing of the hangers
Q' = lateral reaction of the radius-bar pin
nN_2 = friction force between inner rail and wheel $n = 0.25$
G = gage of track
d = perpendicular distance from radius pin to axle.

The force Q' causes a skewing of the truck. Taking moments about the outer rail contract with flange, we have

$$Q'd - nN_2G = 0 \qquad [74]$$

and

$$Q + Q' = Y \qquad [75]$$

If the reaction Q is applied at a distance h_q above the rail, and Q' a distance h_q', then

$$N_2 = \frac{W + w}{2} - \frac{Qh_q + Q'h_q'}{G} \quad . \; . \; . \; . \; . \; . \; . \quad [76]$$

Substituting Equations [75] and [76] in [74] we have,

$$(Y - Q)d - n\frac{(W + w)}{2}G + n[Qh_q + (Y - Q)h_q'] = 0 \quad . \quad [77]$$

hence

$$Y = \frac{Q[d - n(h_q - h_q')] + n\frac{(W + w)}{2}G}{d + nh_q'} \text{ lb.} \quad . \; . \; . \; . \quad [78]$$

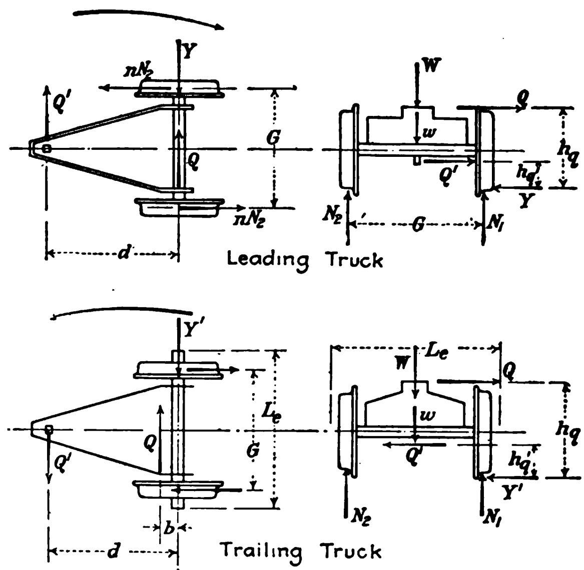

FIG. 12 DIAGRAM OF REACTIONS ON LEADING AND TRAILING TWO-WHEEL TRUCKS

As a first approximation the above value of Y reduces to

$$Y = Q + n\frac{(W + w)}{2}\frac{G}{d} \text{ lb.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [79]$$

With a radius bar of ideal length, the friction nN_2 would be along the axle at the inner wheel and the force $Q' = 0$. With this condition we have

$$Y - nN_2 = Q \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [80]$$

and since

$$N_2 = \frac{W + w}{2} - \frac{Qh_q}{G} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [81]$$

we have

$$Y = n\frac{(W+w)}{2} + Q\left(1 - n\frac{h_q}{G}\right) \text{ lb.} \quad \ldots\ldots \quad [82]$$

$$= n\frac{(W+w)}{2} + Q \text{ (approx.) lb.} \quad \ldots\ldots \quad [83]$$

which is roughly the same value as Equation [79].

46 Let us next consider the reactions on a trailing truck, assuming the rear part of the rigid wheelbase in contact with the outer rail as would occur at high speeds or when a long wheelbase passes a sharp curve. The lateral reaction on a trailing truck is applied from the equalizer by means of friction pads or swing links. In the former the lateral reaction is constant, depending only on the load on truck and the coefficient of friction of pad, while in the latter it depends upon the swing of the links. In traversing a curve the equalizer loads brought on the truck are unequal on either side. Since, however, one side is increased as much as the other side is decreased, the total lateral reaction remains the same. In Fig. 12-B let

Q = total lateral reaction, lb.
b = perpendicular distance of Q from axle, in.
Q' = lateral reaction of radius pin, lb.
d = perpendicular distance from radius pin to axle
nN^2 = friction force between inner wheel and rail
ΔW = increment change of equalizer load brought on truck on either side, lb.
L_e = distance between equalization.

Then

$$Q'd - Qb - nN_2G = 0 \quad \ldots\ldots\ldots \quad [84]$$

and

$$Y = Q - Q' \quad \ldots\ldots\ldots\ldots \quad [85]$$

To determine N_2, we have

$$N_2 = \frac{W+w}{2} - \frac{Qh_q - Q'h_q' + \Delta WL_e}{G} \quad \ldots\ldots \quad [86]$$

Substituting Equations [85] and [86] in [84] we have

$$(Q-Y)d - Qb - n\frac{(W+w)}{2}G + n(Qh_q - (Q-Y)h_q' + \Delta WL_e) = 0 \quad [87]$$

from which we obtain

$$Y = \frac{Q[d - b + n(h_q - h_q')] - n\left[\frac{W+w}{2}G - \Delta WL_e\right]}{d - nh_q'} \text{ lb.} \quad [88]$$

As a first approximation, the above formula reduces to,

$$Y = Q\left(\frac{d-b}{d}\right) + n\triangle\frac{WL_e}{d} - n\left(\frac{W+w}{2}\right)\left(\frac{G}{d}\right) \text{ lb.} \quad . \; . \; . \quad [89]$$

47 The lateral reactions on a four-wheel guiding truck consist of the center-pin reaction, the centrifugal force of the truck which is usually small as compared with the center-pin reaction, and the flange and friction forces at the base of the wheels. The center-pin reaction being applied at a height h above the rails, causes an over-

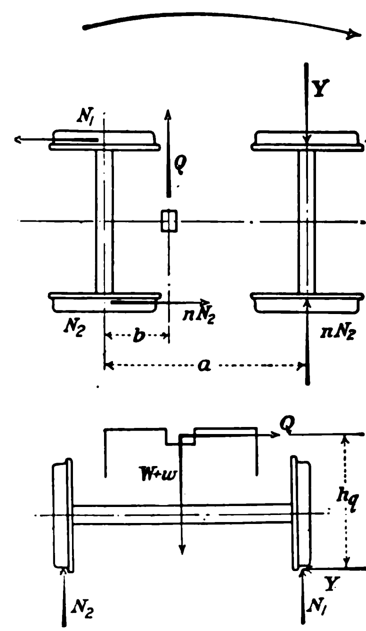

Fig. 13 Diagram of Reactions on Four-Wheel Trucks

turning couple which increases the rail load on the outer rail over that on the inner. In Fig. 13, let

Q = the center-pin lateral reaction, lb.
C_t = centrifugal force of the truck, lb.
h = height of center pin above rail, in.
h_c = height of center of gravity of truck above rail, in.
w = weight of truck, lb.
W = center-pin load, lb.
a = distance between axles, in.
b = distance of center pin from rear axle, in.

Then

$$(Y - nN_2)a - Qb - C_t\frac{a}{2} - nN_2G = 0 \quad . \; . \; . \; . \quad [90]$$

where

$$N = \frac{W + w}{4}; \quad N_2 = \frac{W + w}{4} - \frac{Qh}{2G} - C_t\frac{h_e}{2G} \quad . \; . \; . \quad [91]$$

and

$$C_t = \frac{Wv^2}{gR} \text{ lb. (approx.)} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [92]$$

Neglecting the effect of centrifugal force of the truck itself and the small change in rail load,

$$Y = nN\left(1 + \frac{G}{a}\right) + Q\frac{b}{a} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [93]$$

where $N = (W + w)/4$. When $b = \frac{1}{2}a$, as occurs with the majority of simple four-wheel trucks,

$$Y = \frac{Q}{2} + nN\left(1 + \frac{G}{a}\right) \text{ lb.} \quad . \; . \; . \; . \; . \; . \; . \quad [94]$$

48 We have seen that very roughly the lateral reaction on a two-wheel truck is $Y = Q + nNG/d$; hence for a given guiding force Q the lateral reaction of the front flange is reduced considerably with a four-wheel truck over that on two-wheel truck, though that part of the reaction Y required to skew the truck about a curve is greater with a four-wheel truck than with a two-wheel truck. It is for this reason that when a large guiding force Q is required a four-wheel truck should always be used rather than a two-wheel truck.

49 In the foregoing analysis the effect of centrifugal force of the trucks themselves was considered negligible as compared with the lateral reaction brought on to the truck. The effect of various radii of curvature on the friction forces developed between the wheels is small. On very small-degree curves, that is, curves of large radius, the friction forces become greatly reduced, since the larger radius of the outer wheels tend to assist in the skewing of the truck which is now small. Thus the inherent forces developed in a truck are not greatly influenced by the radius of curvature though the lateral reactions brought on to a truck at the center pin usually increase with the degree of curve.

LATERAL REACTIONS ON A RIGID WHEELBASE

50 The lateral reactions of the cab or boiler are exerted on the frame of the rigid wheelbase. These reactions are balanced by the

reactions of the guiding trucks and the flange and friction reactions exerted by the rails on the wheels of the rigid wheelbase. At low speeds the lateral reactions of the cab or boiler are small as compared with the various friction forces at the base of the wheels of the rigid wheelbase and therefore the moment of the guiding reaction of the trucks together with the flange reaction of the front driver are exerted to overcome the friction torque required to skew the rigid wheelbase about its friction center. We would expect, therefore, at low speeds the lateral reactions to be quite independent of the

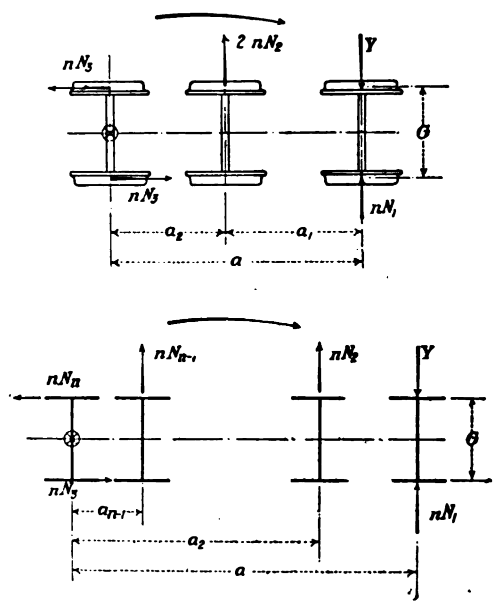

FIG. 14 DIAGRAMS OF REACTIONS ON RIGID FOUR-WHEEL TRUCKS

radius of curvature, while at high speeds the lateral reactions to be greatly dependent on the degree of curvature.

51 Let us consider the lateral reactions brought on a rigid wheelbase with an intermediate driver. Neglecting the effect of centrifugal force, we have the flange reaction utilized in overcoming the friction torque due to the skewing of the rigid wheelbase. Further, it will be assumed that the friction center is at the middle of the rear axle. Then if the wheel loads for the various axles are N_1, N_2, and N_3 (see Fig. 14.),

$$(Y - nN_1)a + 2nN_2a_2 + nN_3G = 0 \quad . \quad . \quad . \quad . \quad . \quad [95]$$

hence,

$$Y = nN_1 + 2nN_2\frac{a_2}{a} + nN_3\frac{G}{a}\text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad [96]$$

For equal driver-wheel loads and the middle driver approximately midway between the outer drivers,

$$Y = nN\left(2 + \frac{G}{a}\right)\text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [97]$$

In general, for a rigid wheelbase of n axles, assuming the rotation of the wheelbase about the rear axles,

$$(Y - nN_1)a + 2nN_2a_2 + 2nN_3a_3 \quad . \quad . \quad 2nN_{n-1}a_{n-1} + nN_nG = 0. \quad . \quad [98]$$

hence

$$Y = nN_1 + 2nN_2\frac{a_2}{a} + 2nN_3\frac{a_3}{a} \quad . \quad . \quad . \quad 2nN_{n-1}\frac{a_{n-1}}{a} + nN_n\frac{G}{a}\text{ lb.} \quad [99]$$

and for n' equal driver loads with axles equally spaced, then,

$$Y = nN\left\{1 + \frac{2}{n'-1}\left[(n'-2) + (n-3) \quad . \quad . \quad . \quad n' - (n'-1)\right]\right\} + \frac{G}{a}\text{ lb.} \quad . \quad . \quad . \quad . \quad [100]$$

From Equations [96] to [100] it is evident that with a given adhesion weight the longer the wheelbase a the smaller the necessary guiding reaction Y. Thus a short wheelbase, of a given number of drivers, offers a greater lateral rail thrust than a long wheelbase. Therefore, as long a wheelbase as will fit in a given curve should be utilized, from the point of view of flange wear. It is further important to note that the closer the intermediate drivers are bunched to the rear driver the smaller the lateral reaction Y. Thus with three pairs of axles the intermediate driver should be placed as close the rear driver as possible.

52 Further, from the previous equation it will be noted that a considerable amount of the reaction Y on the front flange of the leading driver is utilized in overcoming the friction of the middle or intermediate drivers. If the intermediate drivers are given sufficient play in their boxes so that their flanges can come in contact with the outer rail, the lateral journal reaction on the frame of the rigid wheelbase is eliminated and we thus decrease the flange reaction Y on the front driver. Thus in laying out a wheelbase there is considerable advantage in maintaining flanges on the intermediate drivers.

53 When a rigid wheelbase is partly guided by guiding trucks, we have several arrangements:

a Rigid wheelbase of four wheels guided by a four-wheel truck 4–4–0 class American-type running gear

b Rigid wheelbase of six wheels guided by a two-wheel truck 2–6–0 class Mogul-type running gear

c Rigid wheelbase of six wheels guided by a four-wheel truck with a two-wheel trailer truck 4–6–2 class Pacific-type running gear.

In the analysis of such running gears we have an interaction between the truck and rigid wheel system of a definite value depending on the swing of the links, which in turn is a function of the lateral displacement of the truck. Constant-resistance guiding trucks have been extensively used. In this case the interaction takes place between slides or rollers and a plane of constant inclination. Finally, the lateral reaction may be simply due to friction. In either of the two latter cases the reaction is independent of the lateral displacement of the truck from the center line of the frame.

54 The effect of the lateral reaction between the truck and rigid wheel system with a leading truck is to decrease the lateral reaction of the rail on the flange of the front driver. On the other hand, the lateral center-pin reaction of the truck causes a lateral reaction of the rail on the front outer flange of the truck. In the design of the running gear the ratio of the truck-flange guiding reaction to the flange reaction of the first driver of the rigid wheelbase is a matter of considerable importance as the principal flange and consequent rail wear may be attributed to these reactions, and therefore the minimum wear depends upon the proper ratio used.

55 With a trailing truck when the speed is high or with a long rigid wheelbase on a sharp curve the truck is brought in contact with the outer rail and the lateral reaction between the truck and rigid wheelbase is to increase the side rail thrust on the flange of the front driver. Therefore, in a trailing truck it is of considerable importance to decrease the lateral reaction to a minimum. When a guiding truck functions both as a leading and trailing truck, as when a locomotive runs in opposite directions, we must compromise between the amount of lateral guiding when leading and the amount of lateral resistance when trailing. When leading, if a = length of rigid wheelbase and b = distance from front driver to center of truck, then for a lateral reaction Q,

a When the truck acts as a leading truck, the lateral reaction Y on the front driver is reduced by $Q\left(\frac{a+b}{a}\right)$ lb.

b When the truck acts as a trailing truck, with a long rigid wheelbase or at high speeds, the lateral reaction Y on the front driver is increased by $Q\left(\frac{b}{a}\right)$ lb.

56 In a two-way single-wheelbase engine with guiding trucks on either end, the effective reduction in the flange reaction Y of the leading front driver becomes approximately Q lb. When the rigid wheelbase is relatively short, that is, free from binding and with low speeds, the lateral reaction of the trailing truck is negligible. In the derivation of the following elementary formulæ the effects of centrifugal reactions are neglected.

57 In the Pacific-type running gear, 4–6–2 class, the rigid wheelbase is guided by a four-wheel leading truck with a two-wheel trailing truck which offers lateral resistance. We have now the guiding lateral reaction of the front truck and the resisting lateral reaction of the rear trailing truck in addition to the radius-pin reaction of the rear truck which aids in the turning of the rigid wheelbase. Referring to Fig. 15, let

Q_1 = lateral guiding reaction of front truck, lb.
Q_2 = lateral resisting reaction of trailing truck, lb.
Q_2' = lateral reaction of radius-bar pin of trailing truck, lb.
Y = front flange reaction on four-wheel truck
Y' = front flange reaction of rigid wheelbase
Y'' = flange reaction on trailing truck
a = length of rigid wheelbase
a_1 and a_2 = distance between front and intermediate driver and intermediate and rear driver of rigid wheelbase, in.
a' = distance between axles of the front truck
b = distance from front driver of rigid wheelbase to center pin of truck, in.
c = distance from rear driver of rigid wheelbase to axle of trailing truck, in.
d = length of radius bar of trailing truck, in.
e = distance from trailing axle to line of action of Q_2, in.

For the rigid wheelbase, we have (Fig. 15)

$$(Y' - nN_3)a + Q_1(a + b) - 2nN_4a_2 - nN_5G + Q_2'(c - d) - Q_2(c - e) = 0 \quad . \quad . \quad . \quad . \quad . \quad [101]$$

Further,

$$Q_1 = 2Y - 2nN_1\left(1 + \frac{G}{a'}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad [102]$$

$$Q_2' = \frac{e}{d}Q_2 + nN_6\frac{G}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [103]$$

58 The lateral reaction Q_2 is fixed and depends upon the weight brought on to the trailing truck by the equalization and the coefficient of friction of the supporting pads. When a centralizing device is introduced its reaction also must be included. Therefore $Q_2 =$

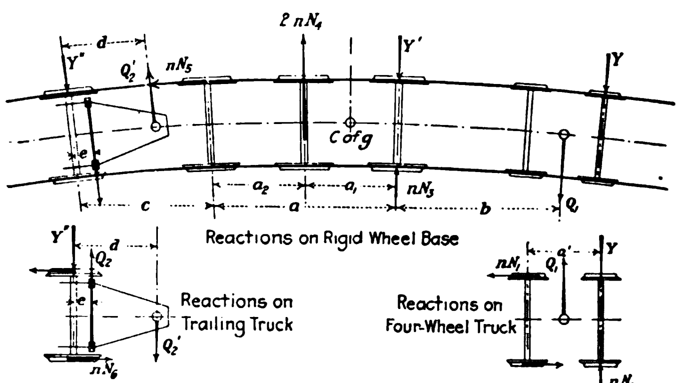

FIG. 15 PACIFIC-TYPE RUNNING GEAR

nkW_e, where kW_e is that part of the equalization load brought on to the trailing truck. Substituting the above values in the previous equations, we have

$$Y = \frac{1}{(2 + J)a + 2b}\left[nN_3a + 2nN_1\left(1 + \frac{G}{a'}\right)(a + b) + 2nN_4a_2 + nN_5G + Q_2\left\{\frac{e}{d}(c - d) - (c - e)\right\} + nN_6\frac{G}{d}(c - d)\right] \quad . \quad . \quad [104]$$

where

$$J = Y'/Y; \quad J = 0.4 \text{ to } 0.9 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [105]$$

59 In all the above equations it will be noted that a large element of the required lateral guiding force Y is utilized in overcoming the friction of intermediate drivers. When these drivers have flanges and sufficient journal and box play can be given, then we are able to greatly reduce the value of Y and other guiding forces.

LATERAL REACTION AT HINGE JOINT BETWEEN TWO ENGINE RUNNING GEARS

60 When two adjacent wheelbases are connected together by a hinge joint, the maximum lateral reaction on the hinge pin may be estimated by a consideration of the guiding reaction required at the hinge to skew the rigid wheelbase of the rear engine unit in a curve, since the force which will completely skew the rear unit can only partly aid in the skewing of the front unit due to the shorter moment arm to the front engine friction center as compared with that to the rear friction center. If

K = the lateral reaction of the hinge, lb.
a = length of the front rigid wheelbase, ft.
b = length of the rear rigid wheelbase, ft.
c = distance from rear axle of front unit to center pin of hinge, ft.
d = distance from front axle of rear unit to center pin of hinge, ft.
d_1 = distance from rear axle of rear unit to its intermediate driver
N_1, N_2 = wheel loads of several axles
G = gage of track, ft.
n = coefficient of rail friction,

then, for a two-axle rigid wheelbase,

$$K = \left(\frac{n}{b+d}\right)(2bN_1 + GN_2) \text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{106}]$$

and for a three-axle rigid wheelbase,

$$K = \left(\frac{n}{b+d}\right)(2bN_1 + 2b_1N_2 + GN_3) \text{ lb.} \quad . \quad . \quad . \quad [\mathbf{107}]$$

The guiding reaction on the leading driver flange of the front rigid wheelbase is reduced by Kc/a lb. We thus see that complete guiding for the rear unit will result only in partial guiding for the front unit.

EFFECT OF CENTRIFUGAL FORCE ON THE RUNNING GEAR

61 When the effect of centrifugal force on the running gear is large, all the drivers of the wheelbase are brought so far as possible in contact with the outer rail. Further, the overturning couple due to the centrifugal action of the running gear increases the normal reaction on the outer rail of the rear driver. The tendency to pivot,

therefore, is at the contact of rear driver and outer rail, rather than at the center of the axle. However, in the assumption of either friction center the value of the lateral reactions are not greatly affected. Further, unless the speed is very great and with a very sharp curve, the effect of the centrifugal reactions is relatively small as compared with the friction forces developed between the rail and drivers, and therefore the pivot point for rotation of the rigid wheelbase is not changed appreciably from the center of the rear axle and the previous equations with a modification for the various centrifugal reactions are applicable.

III ELEMENTARY REQUIREMENTS IN THE DESIGN LAYOUT OF THE RUNNING GEAR

62 In the design layout of the running gear we must compromise between various requirements and limitations. Thus the later reactions induced on rounding a curve require a certain arrangement of wheelbase and guiding trucks, but, on the other hand, we must not lose sight of the load distribution on the rails, the requirements in adhesion, and the best arrangement and location of the drive. The best design is the best compromise giving heed to the basic requirements of these several aspects.

63 *Rigid Wheelbase.* The number of axles in a rigid wheelbase is determined primarily on the allowable axle load and the adhesive requirements. In a steam locomotive we have, neglecting the Mallet type, usually one rigid wheelbase, where in an electric locomotive we may utilize two or more rigid wheelbases below the center pins of either end of cab. Knowing the number of driving axles, we may determine the length of wheelbase to traverse a given curve. When the rigid wheelbase is guided by its own leading driver it has been shown for minimum guiding requirements and consequent minimum flange wear that with a given number of axles, the length of wheelbase should be made as long as possible consistent with traversing the curve of minimum radius. To increase the length of wheelbase we may give greater flange play to the intermediate drivers by decreasing the flange or eliminating it entirely as is often done when very sharp curves must be traversed. On the other hand, eliminating the flanges of intermediate drivers requires the guiding to be done by the frame with consequent increased lateral reaction on the flange of the front driver or guiding truck. In general, giving greater flange play to intermediate drivers in order that they may traverse the sharpest curves requires much

additional journal play for bringing the outer flange against the outer rail, which, if not done, increases the friction torque of the rigid wheelbase and consequent front guiding reaction. It is for this reason then when the wheelbase can be reduced to comfortably fit the sharpest curve it should be done in preference to augmenting the flange play of the intermediate driver.

64 When the guiding of the rigid wheelbase is conducted mainly by the guiding truck, a short rigid wheelbase offers less resistance to skewing than a long rigid wheelbase. On the other hand even when guided by a truck the reaction of the guiding truck is not sufficient to dampen out the lateral oscillations of the rigid wheelbase. Thus, even with guiding trucks a rigid wheelbase acts more or less freely so far as lateral oscillations are concerned. The flange reaction on the leading driver due to lateral oscillation is decreased with the lengthening of the rigid wheelbase. It is for this reason often desirable to maintain as long a rigid wheelbase as will properly traverse the curve of minimum radius.

65 Knowing the flange clearance, the length of wheelbase is readily determined by the formula, $a = \sqrt{2Rs}$ ft. With an intermediate driver, assuming the middle and rear drivers to come in contact with the inner rail, we have $a = 2\sqrt{Rs}$ ft. If we take the standard flange clearance = $\frac{3}{4}$ in. for standard gage, then for a 16-deg. curve assuming a maximum widening of the gage = $\frac{3}{4}$ in.,

$$s = \frac{1.5}{12} = 0.125 \text{ ft.}$$

$$R = \frac{5730}{16} = 358 \text{ ft. (approx.)}$$

hence

$$a = \sqrt{2 \times 358 \times 0.125} = 9.47 \text{ ft.}$$

Assuming the middle driver of a six-wheel rigid wheelbase to come in contact with the inner rail, we have $a = 13.4$ ft. By giving the middle driver greater play the length of wheelbase may be increased. Constructively, it is especially difficult on steam locomotives to give intermediate axles, especially crank axles, sufficient journal play. However, this difficulty has been more or less overcome by continental engineers.

66 The preceding formula for length of wheelbase should not be regarded as giving the maximum limiting value, since a much longer wheelbase can be made to traverse a curve without undue

binding. The value above is the ideal maximum value wherein no binding can take place and no journal play is required. In order to decrease the lateral friction of intermediate driving wheels and thus reduce the flange cutting on the front driver, sufficient journal and box lateral play should be given the intermediate driver in order that its outer flange may come in contact with the outer rail. The axle then acts free and exerts no journal resistance on the frame. When only a small journal and box lateral play is possible, as when the intermediate driver is a main driver, the flange cutting may be reduced by bringing this axle closer to the rear axle. In general the spacing of intermediate drivers close to the rear axle reduces the flange cutting when the length of wheelbase is fixed. On the other hand, decreasing the distance between intermediate and rear axles of a rigid wheelbase, limits the wheelbase to a somewhat shorter length, which in turn may more than offset the reduction in flange cutting due to the closer spacing of the intermediate and rear drivers.

67 *Leading and Trailing Trucks.* Leading trucks are usually either a two-wheel or four-wheel type. From the previous analysis it is shown that the greater guiding effect is produced by placing the truck as far as possible away from the rigid wheelbase. When the entire guiding of the rigid wheelbase is to be effected by the truck it is very important to space the center pin of the truck a considerable distance from the front axle of the rigid wheelbase. On the other hand, we are limited constructively in the allowable swing of a truck, which in turn gives a limitation in the distance between front axle and truck center pin. In considering the type of truck to be used, we should be guided by the relative magnitude of the lateral reaction required of the truck and its center pin. When the lateral reaction is relatively large, it has been shown that to reduce the flange wear a four-wheel truck should be used in preference to a two-wheel truck. In fact, a four-wheel truck is always preferable to a two-wheel truck for guiding considerations, but, on the other hand, with a limited-weight locomotive with maximum adhesion requirements, the greater weight must be thrown on the drivers with a limited weight for the truck. When the lateral reaction required is relatively small, the greater friction required to skew a four-wheel truck on a curve over that of a two-wheel truck brings the difference between the guiding-flange reactions of the two types of trucks smaller. With a given flange reaction we must have a given maximum limiting weight on the axle. Therefore, with relatively small lateral reactions greater minimum weight is required for a four-

wheel truck than for a two-wheel truck. It is for this reason, even from a guiding point of view, a two-wheel truck may prove more advantageous than a four-wheel truck. Further, from constructive layout limitations we are often compelled to use a two-wheel truck in preference to a four-wheel truck. In the final choice of the type of truck to be used, the effect of vertical loads and their distribution in the bending stresses developed in the rail is a matter of vital consideration as well. Therefore, in the final decision of the type of truck to be used — whether two-wheel or four-wheel — a careful analysis should be made as to the relative magnitude of the lateral reactions at the center pin, the guiding-flange reactions, the limiting minimum required weight on the truck and the allowable distribution of weight on wheelbase and trucks. In general, other considerations being the same, a four-wheel truck is always preferable to a two-wheel truck for guiding purposes.

68 To reduce the flange guiding reaction on the front axle of a four-wheel guiding truck, (*a*) the wheelbase should be made as long as possible consistent with the maximum limiting values for passing through a curve, and (*b*) the center pin can be placed closer to the rear axle of the truck, thus giving greater leverage of the front-axle guiding-flange reaction. Short-wheelbase four-wheel trucks with small lateral reactions required have little advantage over two-wheel trucks.

69 To reduce the center-pin lateral reaction on the truck, (*a*) the center pin should be spaced as far as possible from the front axle of the rigid wheelbase; (*b*) intermediate driving axles should be spaced as close as possible to the rear driving axle of the rigid wheelbase; (*c*) journal play in intermediate drivers sufficient to allow their flange to come in contact with outer rail should be given; and (*d*) when the greater part of the guiding is to be effected by the guiding truck, a short rigid wheelbase is of advantage. Care should be taken, however, that the striking angle of the front driver of the rigid wheelbase is not too great, otherwise violent reactions due to lateral oscillations may be included on the flange of the front driving axle of the rigid wheel base when a short rigid wheelbase is used.

70 The lateral reaction of trailing trucks when used in conjunction with long rigid wheel bases or on high-speed locomotives, tends to augment the required guiding forces on the front of the engine. Therefore the lateral reaction on a trailing truck should be made as small as possible The effect of the lateral reaction may

be greatly reduced by bringing the trailing axle as close as possible to the rear driving axle. With relatively short rigid wheelbases the lateral reaction of the trailer may be neglected. In a two-wheel double-end-operation engine with trucks on either end, the front truck having a greater moment about the friction center of the wheelbase than that of the trailer, we always have in general a reduction in flange cutting due to the use of such trucks. The flange reaction on the leading driver of the rigid wheelbase is reduced directly by the lateral reaction at the center pin of the leading truck.

WEIGHT ON GUIDING TRUCK

71 With a given lateral reaction sufficient weights must be brought on the guiding wheel in order that the contact between flange and rail shall always remain in the tread contour of the flange. If ϕ = maximum angle the normal of contact makes with the vertical and d = friction angle, then with a given lateral Y_1 the rail wheel load must be such that $N \geqq Y \cot (\phi - d)$, or otherwise the wheel will leave the rail. If we assume $\phi - d = 45$ deg., then $N \geqq Y$. Since the rail wheel load on one side is greater than on the other, with an axle load twice the lateral reaction Y, we have $N > Y$. Hence, roughly, the axle load should be somewhat greater than twice the maximum lateral reaction expected.

STRIKING ANGLE

72 In the layout of the rigid wheelbase it is important to limit the angle of obliquity or striking angle to a small value. When the angle of obliquity is large the contact between flange and rail is brought to the upper part of the flange and we have danger of derailment when the lateral reaction is appreciable. Further, the area of contact is decreased and the bearing pressure greatly increased. It is therefore desirable to limit the striking angle to 4 deg.

HINGE JOINT BETWEEN TWO ENGINE UNITS

73 By the use of a hinge joint between two separate running gears the lateral guiding of either unit is aided. The hinge joint should be located in advance of the common intersection of the center line of the two running gears when running forward. The location of the pin should be such that $c < \frac{1}{2}(L + b)$ and $d > \frac{1}{2}(L - b)$, where c = distance from rear axle of front unit to intersection of

center line, d = distance from front of rear unit to intersection of center lines, $L = c + d$, and b = length of rear unit rigid wheelbase.

LOCATION OF THE PRINCIPAL MASSES OF A LOCOMOTIVE

74 The effect of a high center of gravity above the rail of the spring-borne parts causes a greater unbalancing of the vertical load distribution on the non-crossed equalization, which in turn results in a change in the journal and normal rail loads with a consequent slight modification of the several friction forces developed between wheels and rail. If we neglect the small change in these friction forces, the lateral reactions throughout the engine are quite independent of the vertical distribution of mass, when the engine is once in a curve.

75 On entering a curve, however, due to journal and box play we have more or less of an impulse lateral action between rail and flange. A mass concentric with the axle has a lateral velocity equal to that of the axle imparted to it. If the center of gravity of the mass is located considerably above the axle, we have a combined translation and rotation about the center of gravity of the mass, since the initial righting moment of the non-crossed equalization is nil. The translating velocity imparted to the center of gravity of the higher mass is therefore considerably less than the velocity imparted to the axle. Therefore with a high center of gravity the impulsive lateral reaction is considerably reduced. The same advantage of a high center of gravity applies in a consideration of lateral oscillations.

76 The rolling angular moment imparted causes a considerable increase in the unbalancing of the load distribution on either side of the equalization over that of the unbalancing caused by the centrifugal effect when in a curve. Thus a high center of gravity though it reduces lateral reactions on entering a curve or in lateral oscillation in a tangent, may on the other hand cause excessive loads over their static values on the springs of the equalization. Once in a curve the location of the position of the center of gravity merely alters the load distribution on the equalization with a minor effect in the change in the lateral reactions throughout the engine.

77 So far as the longitudinal masses are concerned, the masses rigidly attached to the rigid wheelbase should be bunched as closely as possible to the center of gravity of the rigid wheelbase mass consistent with the proper weight distribution on the equalization.

Then the lateral reaction on the front driving wheel is reduced on entering a curve and in the lateral oscillations on the wheelbase.

IV STRENGTH OF VARIOUS PARTS OF THE RUNNING GEAR AS AFFECTED BY THE LATERAL FORCES

78 In a preliminary design the principal parts that should be considered in regard to lateral reactions are (1) Strength of axles (2) Transverse stiffness of frame.

DESIGN OF RAILWAY AXLES

79 In the design of railway axles it has been customary to base the dimensions of the axle on more or less arbitrary fibre stresses, the limiting allowable working stress being gradually reduced on the repetition of axle failures. The method of calculating the strength of axles on all except driving axles has been by the Reuleaux method, whereas driving axles are usually calculated on the basis of maximum piston load and very often by the Reuleaux method as well. It is customary, therefore, to base the limiting fibre stress on either of these two methods.

80 Axle failures due to over stresses are caused usually by lack of consideration given to bending due to excessive lateral reactions at the rail. Further, the journal loads may be greatly increased over their static value due to rolling oscillations and centrifugal effect of the equalized supported parts, and since either the front or rear equalizer system is usually crossed equalized, the balancing moment must be taken up entirely on the other equalizer system, thus augmenting the journal loads.

81 The Reuleaux method for the design of railway axles assumes the static journal load increased by 26 per cent to allow for vertical oscillation and rolling and a lateral reaction applied through the center of gravity of the car 72 inches above the rail, practically sufficient to overturn the axle about the contact of wheel and outer rail.

82 The lateral reaction, applied 72 inches above the rail for standard gage is taken at 0.4 the static journal load increased by 26 per cent. It is obvious that if a lower height above the rail is assumed for the center of gravity of the car, a greater lateral reaction is required to just overturn the axle.

83 With ordinary car axles, the loads apportioned to the journals are equally divided among the various axles. The effect

of centrifugal force and rolling and vertical oscillation is also equally divided among the various axles, and since the mean height of the center of gravity of the average car is approximately 72 inches above

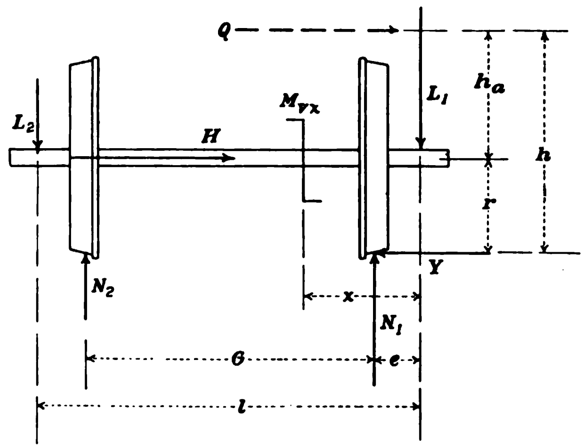

FIG. 16 REACTION ON TRUCK AXLES, OUTSIDE JOURNALS

the rail, we are completely justified in using the Reuleaux method as a maximum possible loading to be brought upon an axle.

84 In the running gear of a locomotive, however, we have leading and trailing trucks, which are designed to give definite lateral

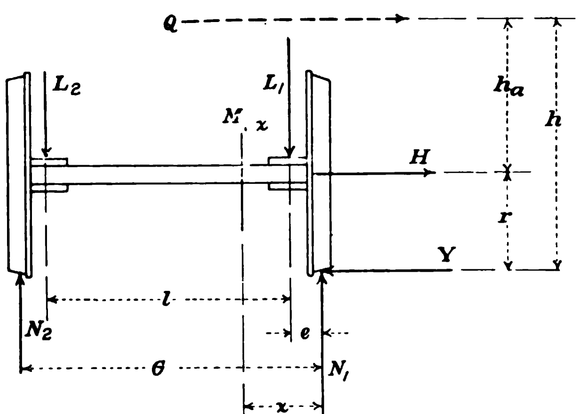

FIG. 17 REACTION ON TRUCK AXLES, INSIDE JOURNALS

reactions for guiding the rigid wheelbase, and these lateral reactions applied to the trucks are quite independent of the various other lateral forces, such as centrifugal forces, etc., and the various journal loads as well. We have, thus, very often large lateral reactions applied at a small height above the rail and these in turn induce large flange reactions, causing bending moments considerably in excess of the bending moments due to journal loads. In the design of such axles, we are not justified in using the Reuleaux method which assumes the overturning lateral reaction 72 inches above the rail.

85 *Truck axles.* We will consider the strength of an axle of a two-wheel and four-wheel truck respectively. (See Figs. 16 and 17.)

Let L = distance between center line of journals, in.

W = total load on truck including its weight, lb.

L_1 and L_2 = journal loads, outer and inner, respectively, lb.

Y and N_1 = the lateral and vertical component of the reaction of the outer rail, lb.

N_2 = the vertical reaction of the inner rail, lb.

H = the horizontal thrust along the axle against the hub of the outer wheel for inside journal supports or against outer hub of inner wheel for outside journal boxes.

86 Then for a two-wheel truck with inside journal boxes, in vertical plane, we have,

$$L_1 = \frac{W}{2} + \frac{Qh_a}{L}; \qquad L_2 = \frac{W}{2} - \frac{Qh_a}{L} \quad . \; . \; . \; . \quad [108]$$

$$N_1 = \frac{W}{2} + \frac{Qh}{G}; \qquad N_2 = \frac{W}{2} - \frac{Qh}{G} \quad . \; . \; . \; . \; . \quad [109]$$

where h_a and h are the heights of center pin of truck from axle and rail respectively. Evidently $h - h_a = R$, the radius of the wheel.

87 Further,

$$Y = Q + nN_2\ G/d$$

$$= Q + \frac{nW}{2}\frac{G}{d} \text{ approx.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [110]$$

where d = length of radius bar.

88 Measuring from the center line along the axle, we have, for the equation of bending moment in a vertical plane,

$$M_{vx} = Yr - N_1x$$

to the center line of the journal box bearing

$$M_{vx} = Yr - N_1x + L_1(x - e) \quad . \; . \; . \; . \; . \; . \quad [111]$$

beyond the center of journal.

89 In a horizontal plane we have a very small bending moment nN_2e and torsion nN_2R due to the skewing of the truck.

90 These may be safely neglected as compared with the vertical bending. With outside journals, the equation of bending in a vertical plane becomes

$$M_{vx} = L_{1x}$$

to the center line of rail

$$M_{vx} = L_1x + Yr - N_1(x - e) \quad . \; . \; . \; . \; . \; . \quad [112]$$

beyond the center line of rail.

91 The maximum bending moment with inside journals,

$$M = Yr \text{ (in lb.)} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [113]$$

while with outside journals

$$M = Yr + L_1e \text{ (in lb.)}. \quad . \; . \; . \; . \; . \; . \quad [114]$$

Hence axles with outside journal boxes must have larger axles than axles with inside journals, due to the additional bending moment of the journal load.

92 With a four-wheel truck the greatest stress comes on the front axle. The vertical reaction at the journal and rails respectively are:

$$L_1 = \frac{W}{4} + \frac{Qh_a}{2L}; \qquad L_2 = \frac{W}{4} - \frac{Qh_a}{2L} \text{ (lb.)} \; . \; . \quad [115]$$

$$N_1 = \frac{W}{4} + \frac{Qh}{2G}; \qquad N_2 = \frac{W}{4} - \frac{Qh}{2G} \text{ (lb.)} \; . \; . \quad [116]$$

For a four-wheel truck, we have for the flange reaction

$$Y = Q/2 + nN_2 \; (1 + G/a) \quad . \; . \; . \; . \; . \; . \quad [117]$$

where a = the distance between the axles of the truck.
The bending moment equation, for inside journal boxes, becomes

$$M_x = Yr - N_{1x}$$

to the center line of the journal.

$$M_x = Yr - N_{1x} + L_1 \; (x - e) \quad . \; . \; . \; . \; . \; . \quad [118]$$

beyond the center of journal, and for outside journal boxes,

$$M_x = L_1x$$

to the center line of rail,

$$M_x = L_1x + Yr - N_1 \; (x - e) \quad . \; . \; . \; . \; . \; . \quad [119]$$

beyond the center line of rail.

93 In the above equations for bending moment, the importance of the lateral reaction is clearly seen. Thus axles for four-

wheel trucks may be made lighter than for two-wheel trucks. To allow for vertical and lateral oscillations, the load W and consequently Q, since the latter is usually proportional to W, should be increased by 25 per cent.

94 *Driving Wheel Axles.* The axles of driving wheels are usually designed on the basis of maximum steam thrust with steam locomotives and maximum adhesion torque with electric locomotives. Very often a separate estimation is made of the strength of an axle, on a curve by the Reuleaux method as well. It is, however, important to consider the loadings simultaneously. We have two cases to consider, (1) starting on curve and (2) the reaction on the axle at high speeds on the maximum curvature. In the former we have the maximum steam thrust bending in a horizontal plane together with bending in a vertical plane due to the lateral reaction at the wheel flange required to overcome the friction moment of the rigid wheelbase. In the latter the lateral reaction at the wheel flange is increased, but on the other hand the main rod reaction on the wrist pin is decreased.

95 *Steam Locomotive Driving Axles.* Let

V = lateral flange reaction on the driver, lb.
R = lateral reaction of journal box in frame, lb.
N = rail load under driving wheel
P = total piston load = $0.785\ d^2\ p$
p = full boiler pressure, lb. per sq. in.
W = total journal load, lb.
r = radius of driver, in.
S = piston stroke, in.
a = the distance from crank pin centers to center line of box bearing in frame measured in direction of axle, in.
b = the distance from wheel tread center line to journals box center line measured in direction of axle, in.
c = distance between center lines of frame, in.
G = gauge of tread, in.
H_1 and H_2 = horizontal journal box reactions, lb.
V_1 and V_2 = vertical journal box reactions, lb.

96 The horizontal journal reactions become, for both outside and inside frames,

$$H_1 = P_r\left(\frac{a+c}{c}\right) \pm P_l\frac{a}{c} \pm nN \quad \ldots\ldots \quad [120]$$

$$H_2 = P_l\left(\frac{a+c}{c}\right) \pm P_r\frac{a}{c} \pm nN \quad \ldots\ldots \quad [121]$$

and the vertical journal box reactions are

$$V_1 = \frac{W}{2}\left(1 + \frac{\Delta\phi}{\phi}\right) \text{ (lb.)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [122]$$

$$V_2 = \frac{W}{2}\left(1 - \frac{\Delta\phi}{\phi}\right) \text{ (lb.)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [123]$$

$$\Delta\phi = \frac{\Sigma Q_g d_g + \Sigma Q d_q \text{ (lb.)}}{c} \quad . \quad . \quad . \quad . \quad . [124]$$

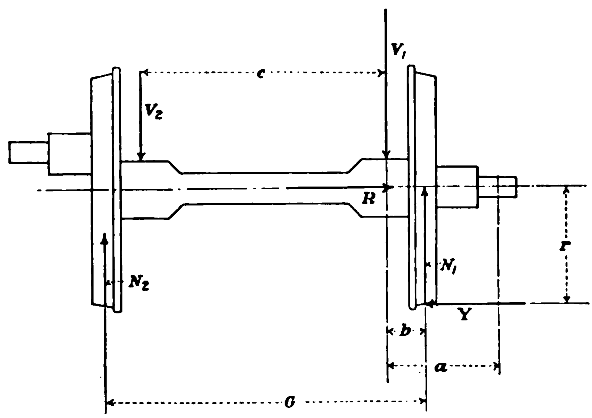

FIG. 18 STEAM LOCOMOTIVE DRIVING AXLE REACTIONS, VERTICAL PLANE

where ϕ = the resultant reaction on one side of the non-crossed equalizer system

ΣQ_g = total lateral reaction at the driving boxes

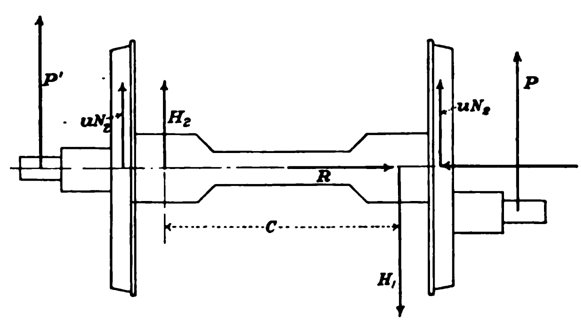

FIG. 19 STEAM LOCOMOTIVE DIVIDING AXLE REACTIONS, HORIZONTAL PLANE

ΣQ = total guiding truck reactions

d_g = distance from center of gravity spring-borne parts to center line of driving axles

d_q = distance from center of gravity of spring-borne parts to truck center pins.

97 Neglecting the weight of the driving axles as a first approximation, we have

$$N_1 = \frac{W}{2}\left(1 + \frac{\Delta\phi}{\phi}\frac{c}{G}\right) + (Y - nN)\frac{r}{G}\text{ (lb.)} \quad . \; . \; . \; . \quad [125]$$

$$N_2 = \frac{W}{2}\left(1 - \frac{\Delta\phi}{\phi}\frac{c}{G}\right) - (Y - nN)\frac{r}{G}\text{ (lb.)} \quad . \; . \; . \; . \quad [126]$$

98 For coupling driving axles, the coupling rod reaction should be taken when the crank is at 45 deg. Then $P = 0.94\,W\frac{r}{s}$ where r = radius of driver and s - length of stroke. When the main driver is not a leading axle, the lateral reaction is less than Y, its value being nN and $N_1 = \frac{W}{2}\left(1 - n\frac{R}{G}\right)$ and $N_2 = \frac{W}{2}\left(1 + n\frac{R}{G}\right)$. On starting we may generally neglect the term $\frac{W}{2}\,\frac{\Delta\phi}{\phi}$.

99 Then if M_h, M_v and T are the horizontal and vertical bending moments and torsion, we have at the journal box bearings,

Inside Frames	*Outside Frames*
$M_h = Pa + nN_1b$	$M_h = Pa$
$M_v = Yr - N_1b$	$M_v = P\frac{sa}{2l}$
$T = nN_1r$	$T = P\frac{s}{2}$

100 When the lateral reaction is relatively large as compared with the piston load with outside frames the maximum bending is likely to occur at the wheel hub. In this case we have a decrease in the horizontal bending but a larger vertical bending moment introduced. We will have

$$M_h = P(a + b) - H_1b$$
$$M_v = Yr$$
$$T = nN_1r$$

The equivalent bending moment at the section considered, then becomes,

$$M_e = \frac{I}{2}\left(M + \sqrt{M^2 + T^2}\right)\text{ (in lb.)} \quad . \; . \; . \; . \quad [127]$$

where $\qquad M = \sqrt{M^2_h + M^2_v}$ (in lb.)

101 *Electric Driving Axles, Geared Axles.* With geared axles we have a part of the motor frame suspended directly on the axle. The reactions on the axle consist of the bearing reactions of the motor frame together with the gear, journal box and lateral reaction of the flange.

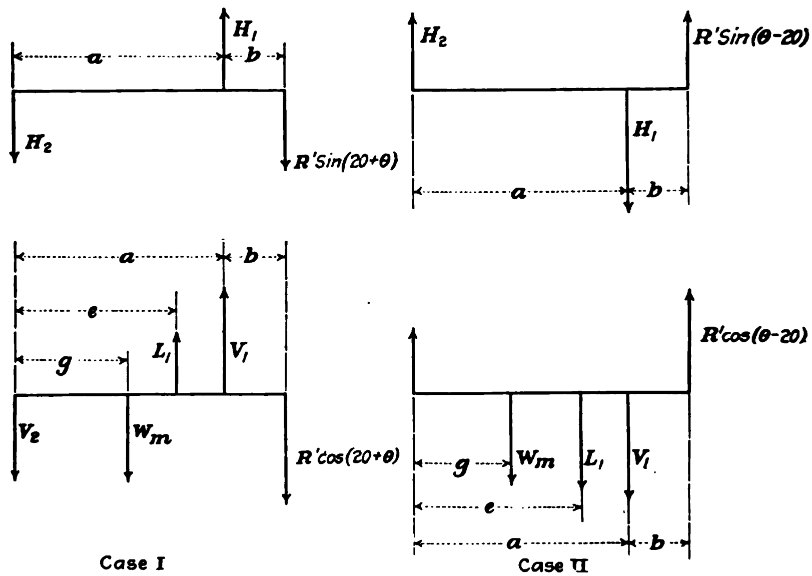

FIG. 20 ELECTRICAL DRIVING AXLES

R = gears and pinion reaction, lb.
N_1 and N_2 = normal rail reactions at wheelbase, lb.
d_p = diam. of pinion to pitch circle, in.
d_g = diam. of gear wheel to pitch circle, in.
d_o = perpendicular distance from center of axle to nose support
a = distance between motor bearings on axle, in.
b = distance from gear and motor-bearing to center line of gear, in.
c = distance from gear center line to rail center line
d = distance from rail center line to journal box center line
D = diam. of wheel, in.

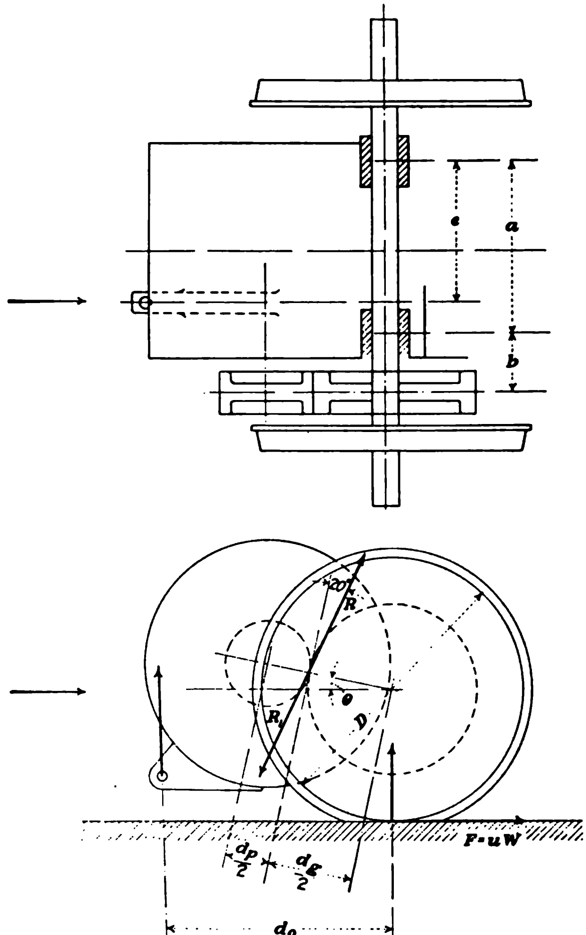

Fig. 20 Electrical Driving Axles

θ = inclination of motor center with horizontal
L_1 = nose reaction, lb.
W_1 = total journal load, lb.
w_1 = weight of one axle and its two wheels, lb.
W_m = weight of motor housing, lb.
W = total rail load, lb.

For the gear reaction, we have,

$$R = \frac{W}{3 \cos 20} \frac{D}{d_g} \text{ lb.} \qquad [128]$$

Assuming 0.6 of the motor weight to be carried directly on the axle, we have for the nose reaction,

$$L_1 = \frac{W}{6}\frac{D}{d_o} \pm 0.4\ W_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [129]$$

+ (when reaction acts up on nose, Case I

− (when reaction acts down on nose, Case II)

For the horizontal reactions on motor housing

$$H_1' = R\frac{(a+b)}{a}\sin(20+\theta)$$

$$= \frac{W}{3}\frac{D}{d_o}\left(\frac{\sin(\theta \pm 20)}{\cos 20}\right)\frac{(a+b)}{a}\ \text{(lb.)} \quad . \quad . \quad . \quad . \quad . \quad [130]$$

$$H_2' = \frac{Rb}{a}\sin(20+\theta)$$

$$= \frac{W}{3}\frac{D}{d}\left(\frac{\sin(\theta \pm 20)}{\cos 20}\right)\left(\frac{b}{a}\right)\ \text{(lb.)} \quad . \quad . \quad . \quad . \quad . \quad . \quad [131]$$

and for the vertical components on motor housing,

$$V_1 = \frac{R\cos(\theta \pm 20)(a+b) - L_1 e \pm W_m g}{a}\ \text{(lb.)} \quad . \quad . \quad . \quad . \quad . \quad [132]$$

using + for Case I and − for Case II,

$$V_2 = \frac{R\cos(\theta \pm 20)b + L_1(a-e) \pm W_m(a-g)}{a}\ \text{(lb.)} \quad . \quad . \quad [133]$$

using − for Case I and + for Case II.
These reactions reversed are the reactions exerted by the motor housing and gear on the axle.

102 The journal box reactions become

$$H_1 = H_2 = \frac{W}{6}\ \text{(lb.)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [134]$$

The total rail load equals

$$2N = N_1 + N_2$$
$$= W_1 + W_m + w_1 \pm L_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [135]$$

− when reaction on nose acts up and + when reaction on nose acts down. Hence,

$$2N = N_1 + N_2$$
$$= W_1 + w_1 + 0.6W_m \mp \frac{ND}{3d_o} \quad . \quad . \quad . \quad . \quad . \quad [136]$$

therefore for the mean rail load, we have

$$N = \frac{\frac{W_1 + w_1}{2} + 0.3W_m}{1 \mp \frac{D}{6d_o}} \quad \ldots\ldots\ldots\ldots \quad [137]$$

103 But on the other hand under traction, the journal load W_1 does not retain its static value. It would be necessary therefore to modify W_1 slightly. Further, W_1 is a distributed load but of small value as compared with the loads brought on the axle. If, therefore, we assume the mean rail load and journal load the same we greatly simplify the calculations and the results are amply accurate considering the other variables involved. Hence,

$$N = W/2\left(1 \pm \frac{D}{3d_o}\right)$$

when W = static journal load.

The rail loads become,

$$N_1 = N + \frac{Qh}{2G} \quad \ldots\ldots\ldots\ldots \quad [138]$$

$$N_2 = N - \frac{Qh}{2G} \quad \ldots\ldots\ldots\ldots \quad [139]$$

where G = gauge of track.

For trucks

$$V_1 = \frac{W}{2} + \frac{Qh_a}{2L} \quad \ldots\ldots\ldots\ldots \quad [140]$$

$$V_2 = \frac{W}{2} - \frac{Qh_a}{2L} \quad \ldots\ldots\ldots\ldots \quad [141]$$

where h_a and h are the heights of center pin of trucks from axle and rail respectively, Q is the lateral reaction at the center pin, and the distance between ground boxes.

104 When the motors are seated in a rigid wheelbase with side equalization, the reactions on the journals and wheelbase are:

$$V_1 = \frac{W}{2}\left(1 + \frac{\Delta\phi}{\phi}\right) \quad \ldots\ldots\ldots\ldots \quad [142]$$

$$V_2 = \frac{W}{2}\left(1 - \frac{\Delta\phi}{\phi}\right) \quad \ldots\ldots\ldots\ldots \quad [143]$$

$$N_1 = N\left(1 + \frac{\Delta\phi}{\phi}\,\frac{L}{G}\right) + (Y - nN)\,\frac{D}{2G} \quad \ldots\ldots \quad [144]$$

and

$$N_2 = N\left(1 - \frac{\Delta\phi}{\phi}\,\frac{L}{G}\right) - (Y - nN)\,\frac{D}{2G} \quad \ldots\ldots \quad [145]$$

105 In addition we have the lateral reaction of the flange which causes a bending moment of considerable value at the wheel hub. The magnitude of this reaction has been evaluated in previous reactions.

106 From the reactions on the axle we may readily plot the bending moment curve. We have

At the wheel hub:

$$M_h = \frac{W}{6} d$$

$$M_v = V_1 d + Y \frac{D}{2}$$

$$T = \frac{W}{12} D$$

At the gear center:

$$M_h = \frac{W}{6} (d + c)$$

$$M_v = V_1(d + c) + Y \frac{D}{2} - N_1 c$$

$$T = \frac{W}{12} D$$

STRENGTH OF A LOCOMOTIVE FRAME

107 The exact determination of the stresses in a locomotive frame is impossible since the reactions brought on to the frame are indeterminate to a considerable degree. Thus on a steam locomotive we have indeterminable reaction of the cylinder saddle and the various boiler supports. With an electric locomotive the major reactions brought on to the frame are more definite in character Even after a careful computation of the stresses at various sections, we always have large secondary stresses due to the structural form of the frame together with very large internal stresses due to shrinkage, etc. However it is believed that even a crude approximation of the various stresses in a locomotive frame is of considerable value if for nothing more than a logical relative comparison of different types of frames with their loadings. In the following only the lateral reactions and their consequent stresses will be considered.

108 The external reactions brought on a locomotive frame on steam locomotives consist of lateral reactions at the cylinder saddle and the various boiler pads. In addition, due to the high center of gravity of the boiler, a large transverse bending moment is introduced at the saddle which is balanced by a couple due to the transverse change in loading of the rear equalizer system.

109 Therefore, a considerable part of the locomotive frame is in torsion. The magnitude of this torsion depends upon the magnitude of the centrifugal force of the boiler, and its height above the frame.

The lateral reactions excited by the boiler on the frame are balanced by the journal box reactions of the several drivers and the lateral reactions of the guiding trucks. It is important to note, however, that very often the centrifugal forces of the boiler are negligible as compared with the lateral reactions at the journals due to the large magnitude of the friction torque of the rigid wheelbase. In an electric locomotive the lateral reaction of the cab is brought on the frame principally at the center pin, though we have small lateral forces due to friction at the various cab spring supports. Further, the overturning effect due to centrifugal force of the cab causes a crosswise bending which is taken up by side supports usually at the center pin. The center of the rear equalization is not greatly to the rear of the center pin, hence usually a smaller part of the frame is subjected to torsion than with a steam locomotive. Again usually the lateral reaction of the center pin of the cab is small as compared with the friction forces at the wheelbase.

110 The maximum lateral reaction brought on the frame is that due to either the guiding truck lateral reaction acting at its center pin or the journal box reaction of the leading driver. In an electric locomotive the guiding reaction of the front truck causes lateral bending in the front part of the frame. The reaction of the leading driver on the frame = $(Y - nN_1)$ lb. when Y = the flange reaction and nN_1 = the friction lateral force of the inner wheel. Such a reaction causes secondary stresses between cross-beam supports, sometimes of considerable value since the resistance of the section through the boxes is relatively small as compared with vertical bending. The determination of the stresses in cross transom is complicated in a steam locomotive by the cross-bracing of the boiler pads, whereas in an electric locomotive these are used doubly for carrying motor or cab loads and if strong enough for these loads they are usually amply strong laterally.

111 In conclusion, the author trusts that careful experimental work will shortly be undertaken, the results of which will confirm the foregoing theoretical presentation and furnish further data which will prove of special value in locomotive design. Since the paper deals only with the aspect of traversing a plane curve, it is hoped that a future paper on lateral oscillations and the dynamics of the running gear on entering a curve, with a consideration of spirals and the effect of banking, will be shortly published.

References pertaining to Section I and Roy diagram.

(a) Die Lokomotiven der Gegenwart, Das Eisenbahn-Maschinenwesen, Von Borries. (b) Handbuch zum Entwerfen Regelspuriger Dampflokomotiven, Von Lotter.

DISCUSSION

Lawford H. Fry. The paper opens up a subject of considerable importance, but it hardly does more than open the subject. The equations developed are all presented in general form, so that their practical significance is not to be seen without applying them to particular conditions. It is to be hoped that in the subsequent papers promised by the author the equations will be brought into direct relation with actual designs. In this connection the writer would like to take issue with the author on the statement in Par. 1 to the effect that the important question of the lateral reactions of the running gear on curves has been neglected by the majority of locomotive designing engineers. It may be true that no complete mathematical study of the question has been published in English, although French and German writers have dealt with it. Nevertheless the practical side of the question has received the attention of every designer who has taken part in the development of a new design. With the introduction of the 4-4-0 or American type of locomotive, the problems of a four-wheel leading truck had to be solved. The Mogul and the Consolidated types to be successful required the development of the two-wheel front truck, while the Atlantic type brought the trailer truck under consideration. In some details of locomotive design, such as for example the boiler proportions, it is possible for the designer to err and to have the error overlooked. In the arrangement of the running gear, however, failure to conform to the fundamental laws forces itself to attention by derailment or excessive flange wear. The fact that the designer is closely held by natural laws and has little freedom of choice is shown by the high degree of uniformity in design. The number of types of running gear which have survived is strictly limited, and in each type the variations in design and dimensions are slight.

In 1864 Zerah Colburn said that the best locomotives presented no more than a judicious compromise between two series of principles, one requiring all parts to be enlarged, the other requiring all parts to be diminished. The locomotive, he said, owed nothing to mathematical investigation. In its early form it had been designed by unlettered men and it had grown into what it was solely through practical experiment.

The fact that the practical designer has secured solutions of certain definite problems does not detract from the value of the mathematical analyses presented by the author. Such work helps to establish the empiric solutions on a rational basis and are necessary in the development of new combinations such as are required for Mallet and electric locomotives.

Having noted the value of the work, the writer would like to call attention to a few points which might be developed more fully to make the conclusions more readily available for the working designer. In Par. 6 the analysis of the action of a four-wheel truck is based on the assumption that the rear axle runs radially and that the instantaneous center of rotation falls on this axle. To one coming new to the work a demonstration of the accuracy of these assumptions is necessary before the equations based on them can be accepted with confidence. Similarly Equation [3] is stated to be a satisfactorily approximate equivalent of Equation [2], but the degree of accuracy is not shown. The assumptions and approximations are undoubtedly justified, but if they were accompanied by the evidence on which they are based, the final conclusions would be immediately available, without it being necessary to go through the complete analysis to ascertain the stability of their foundations. This applies to all of the calculations in which assumptions as to wheelbase positions are made or in which approximations are used.

In Par. 15 an analysis of the positions of a trailing axle is made, but no mention is made of the length of radius bar. Does this mean that the position is independent of this length?

In his study of wheelbase adjustment the author has omitted one important factor which he is probably reserving for the future papers he promises, that is, the effect of the tractive effort. This has a considerable influence on the position taken by the driven wheels, so that the driving wheelbase will not have the same position when developing tractive effort as it will when the engine is coasting. The point requires consideration to make the study complete.

In conclusion the writer echoes the author's hope that experimental work on the subject will be undertaken. This might well be done under the auspices of the Research Committee of the Railroad Section. The author is to be congratulated on having tackled an abstruse problem. It is hoped that in the near future he will clear up some of the still doubtful questions and present a set of formulæ which will be directly available for the busy practical designer.

William E. Woodard. The whole matter of tracking of locomotives and locomotive trucks is of vital interest to the locomotive designer. The author is to be congratulated on the work he has done, which as he puts it, is "suggestive rather than final." These comments are offered with this thought in mind, and with the hope that others will do the same and thus contribute to a better understanding of the subject, and possibly to a more complete solution of the problems under discussion.

Locomotive designers must study this question with care and thought to keep abreast of the experiments and studies on the track structure which are now being made. For the most part these give the accumulative effect of the locomotive wheels on the track and from their nature do not include an analysis of the action of the various elements of the locomotive itself which produce these effects. This is a province of the locomotive designing engineer.

Quoting from Par. 2, the author says "the final design arrived at must necessarily be a compromise between the best arrangement of running gear for traversing curves and for securing stability on tangents at high speed." These requirements appear to the writer to be reversed. Stability on tangent track is the vital factor in arrangement of trucks and truck resistance because by far the greater part of locomotive running is done upon tangent track and the requirement of stability at speeds is, in the writer's opinion, the primary problem. The wheelbase and truck designs should be so arranged as to sacrifice as little as possible of stability to the provisions which have to be made for taking curves. In other words, trucks should be primarily designed for straight-track running with a possibility of deflection on curves. A common way has been to design trucks to curve, and stability on tangent track has been somewhat of a secondary consideration.

Referring to trailer trucks, Par. 14, the position of the trailer truck axle is almost entirely determined by the factors of weight distribution and firebox and ashpan clearance, and to a great extent the latter factor also determines the length of radius bar. A large number of trailing trucks are operating with radius bars considerably longer than called for by theoretical considerations, and they appear to be giving very good service. In this connection, the length of the trailer truck radius bar does not appear to be so vital a factor as the length of the leading truck radius bar. This is probably because the former is being dragged by the locomotive, while the leading truck is being pushed.

Par. 24 reads as follows: "The type of lateral device to be used for guiding depends upon a careful consideration of the lateral forces developed when the engine rounds a curve. Analysis should be made of the required lateral force for various degrees of curvature. In this way we can arrive at the proper function of the lateral reaction as against swing of truck or degrees of curvature." As pointed out in the comments on Par. 2, the writer believes that stability on tangent track is of primary and vital consideration. Any truck which will hold an engine against "nosing" on a tangent track will certainly take care of the same condition on a curve where the rigid wheelbase lies against one rail.

The problem, therefore, is to get a truck which is stable at speeds on a tangent track and yet has capacity for lateral movement on curves without running the side resistance up to a point where the truck is in danger of overturning. With long heavy locomotives now in use, this is not an easy problem, but the condition has been successfully met by certain designs. The parallel swing link hangers and the three-point or Hart link hangers shown in Fig. 7 do not fully meet the requirements for heavy and long locomotives. With both these forms of centering arrangement either the side resistance for normal tangent track running is too small or at the maximum deflection the resistance is too great. On modern locomotives, truck swings of 6 and 7 in. each side of the center are sometimes encountered, and it is especially with these large deflections that trouble is found with either the parallel swing link hangers or the three-point swing link hangers.

In Par. 46, the author says: "The lateral reaction on trailer truck is applied from the equalizer by means of friction pads or swing links." There appears to be considerable question whether stabilizing the trailer truck by friction is the best way to accomplish the desired result. The writer has always felt that the same type of resistance which is used on guiding trucks is the proper form to use on trailing trucks. Controlling the lateral resistance by friction has the disadvantage that friction retards the return of the truck to normal position by the same amount that it opposes the displacement of the truck. Moreover, friction is variable and hard to estimate accurately. Locomotive engine runners have a rather effective method of relieving the lateral motion at the back end of a trailer truck locomotive having frictional resistance over the trailer truck when the side motion becomes excessive. It is a very simple expedient and easily applied. A few handfuls of ballast thrown on the friction plate accomplishes a great deal. This is mentioned to show how indeterminate a factor friction is when used for such purposes as stabilizing engine and trailer trucks. Any trailer truck centering device should return the truck to the true central position. This is one of the defects of the frictional type of resistance, which tends to allow the truck to run to one side or the other and does not center it.

There has recently been brought out a type of trailing truck with gravity centering arrangement which gives promise of successfully meeting some of the difficulties experienced with these trucks.

In reference to Par. 47, Fig. 13, this diagram shows the engine truck center pin set back near the second axle of a four-wheel truck. The same matter is also referred to in Par. 68, (*b*). Although this appears to be theoretically correct, the writer believes practical considerations will always control the location of the center pin of the four-wheel truck. The question of weight distribution, clearance of truck wheels from adjacent parts, and the mechanical construction of the truck itself almost always limits the position of the center pin to the present accepted location, approximately midway between the axles. There is also the consideration of backing to provide for.

At the close of Par. 51, the author makes this recommendation: "thus with three pair of axles the intermediate driver should be placed as close to the rear driver as possible." It is obvious that this should be limited to locomotives operating in one direction only. As a matter of fact, practically all six-coupled locomotives without trucks are switchers which operate in both directions. In such cases it is plain that the intermediate axle should be midway between the front and back axle. The same applies to any six-coupled locomotive operating in both directions.

The matter of trailing truck resistance is again referred to in Par. 55 as follows: "Therefore, in a trailing truck it is of considerable importance to decrease the lateral reaction to a minimum." As pointed out before, the lateral resistance of a trailing truck must be proportioned for stability on tangent track running. The writer believes that not enough attention has been given to this detail in the past, with the result that some trailer truck engines swing badly at high speeds.

It is rather difficult to pass, without comment, a sentence in Par. 65 reading as follows: "By giving the middle driver greater play, the length of the wheelbase may be increased. Constructively, it is especially difficult on steam locomotives, to give intermediate axles, especially crank axles, sufficient journal play. However, this difficulty has been more or less overcome by continental engineers."

At the risk of mentioning a personal matter, the writer might say that this problem has been met in a somewhat different way in this country by allowing lateral motion on either the front or back driving axle of a long wheelbase locomotive. This accomplishes the same result as lateral motion on the intermediate wheel. There are a large number of locomotives now running in this country with very long driving wheelbases and having a special provision for the lateral motion on the front driving axle to accommodate curving. This arrangement has long passed the experimental stage, and appears to very fully meet the need of a flexible driving wheelbase on long wheelbase locomotives. So far as the writer is aware, continental engineers had no hand in this development.

In closing, the writer heartily seconds the wish of the author that experiments will be undertaken to verify the various theories expressed in this paper. There are a great many things to be known about tracking of locomotive wheelbases, locomotive trucks and the lateral resistance of the same that experiments alone can fully answer.

L. K. Sillcox. For the past six months we have been diligently investigating the question of flange wear in locomotive service especially that experienced on engine truck wheels. Tire wear is produced by four causes: first, that due to brake application causing abrasion between brake shoes and face of tread and face of flange, when producing a stop and sometimes when released; second, slipping which is generally experienced in starting trains resulting in not only the loosening of tire, but grave indications of wear where it is experienced to any great extent; third, rolling friction which all classes of wheels have to resist caused by abrasion between wheel face and rail, resulting in flow of metal; fourth, flange wear resulting from pressure of the flange on tire against the rail in curving and on straight track.

For obvious reasons and principally to maintain safe service, close gaging of wheel flanges and treads is a matter of daily routine in the handling of railroad equipment. The four stages of wear above mentioned unite to create danger if not carefully watched, as worn treads are hard on frogs and crossings, and worn flanges are serious for two reasons; first, if the flange wears vertically the wheel may climb a defective joint or open a split switch; second, difficulty is experienced through interference with angle bars, especially if the side face of ball of rail is worn.

Accepting the fact that there are differences in the amount of tire wear, even with the same material and upon engines of the same class as well as that occasioned by different handling of the power, it can be shown that the question of flange lubrication as well as running gear design plays a very great and most important part in this matter. The subject would deserve careful scrutiny even if we were to disregard the question of delay to engines held out of service while returning or replacing the worn tires from the standpoint of cost of replaced tires alone.

When turning a tire to obtain a normal flange, it is necessary to remove a large quantity of valuable wearing metal and oftentimes as much as one-half to three-quarters of an inch thickness of the total rim has to be removed in order to return the wearing face to normal contour.

If flange wear on truck wheels is important, it is more so on driving wheels. This is also true from the standpoint of expense alone because we have to recognize the fact that all drivers must be turned to a size within a given limit of the smallest, whether all their flanges be worn or only one. The drivers being larger, the tendency to climb bad joints or a switch or a frog point is greater in proportion and troublesome derailments can be traced to driving wheels with sharp flanges, particularly when on the forward wheel. In order to save the drivers, the truck must do the principal part of the guiding and if these wheels wear they are much more easily and cheaply replaced than the drivers. The latter, however, can do a portion of the work without much detriment, especially if properly arranged to require attention simultaneously with driving boxes, so as to mimimize the number of times wheels have to be dropped.

The use of bald tires for locomotives has been practically abandoned except in extreme cases. Modern locomotives are generally provided with flanged tires on all wheels, those of the front and rear being slightly closer together than the middle ones; that is to say, one-eighth to one-fourth of an inch.

The slipping of tires can be avoided through proper discipline and care in handling, and difficulty from tread-worn tires is avoidable if proper limits of thickness are maintained. The matter of brake-shoe wear as affecting tires is something that can be easily controlled through proper design and maintenance.

Returning again to the matter of flange wear, which is by far the most important item to consider, it is well to observe the action which takes place. The flange comes in contact with the side of the rail head when passing through a curve, the force depending largely upon the degree of curvature, and the rotation of the wheel at the same time produces a very heavy friction and wear which can be easily observed by examining the inside of an outer rail which has been in use for some time and comparing it with that on straight track. In normal operation this wear is at least 20 per cent greater on the inside rail as compared to the outer, as the tendency is to throw more friction upon the outer wheel so that the inner one must slip to an amount equal to the difference of the lengths of the outside and inside rails, the latter becoming worn entirely upon the top.

To sum up the tendencies in operation, the conditions preceding, and inducing nosing, that is, conditions tending to combine the forces and elements which develop it, are so interlinked and dependent upon so many circumstances, that a mathematical analysis of the phenomena does not yield practical results. The nosing oscillations of locomotives are of the most serious moment as the action of the very great weights affected can produce forces of great magnitude, and yet may readily tilt the rails or spread them at the point of impact, numerous derailments and accidents having been caused in this manner.

In general practice, however, this has been avoided through the use of independent four-wheel trucks to guide engines in high-speed service. In order to prevent the transference of shocks, vibrations, etc., occasioned by track irregularities from the wheel to the rest of the running gear, springs perpendicularly flexible are in general practice interposed between the axles and frames connected together with equal beams so as to stabilize the loading. Relative to the effect of the interaction of springs and equalizers on rail and road bed, it is quite evident that the locomotive will run the safest when the pressure of the wheel on the track is kept most constant, this being especially true for those employed in guiding.

The leading axle of a locomotive when running over irregularity in track is either dropped or raised, with the result that the tension in springs is correspondingly altered and the front end of the locomotive seeks to follow this rise or fall of the axle, resulting in a change in the increased loading of the remainder of the axles.

The ordinary leaf spring has, owing to the separate leaves being restricted from moving longitudinally when deflected, a certain internal resistance which is the greater the larger the number of leaves. This friction tends to dampen the oscillation of the springs and avoids their accumulation. It has, therefore, been experienced that leaf springs are preferable to the coil type for this purpose.

The adequacy or safety of the means employed to lead a locomotive through a curve is dependent upon the effective pressure on the flange of the leading wheel which is the resultant of the lateral pressure and the loading placed upon the wheel. The leading of locomotives on curves as well as on tangents by trucks is by far the safest method to employ, since the oscillations of the locomotive itself have little, if any, effect on the particular direction of motion of the truck; hence, the tendency to set up varying flange pressures is absent. The truck, whose weight is comparatively small compared to the rest of the locomotive, runs more steadily against the outside rail and follows every irregularity of the track without building up oscillations.

The following are the outstanding differences at high speeds without and with the truck:

In the first case, the leading rigid axle in the main frame must follow the movement of the mass of the locomotive and in so doing its action in affecting the leading is influenced so that varying flange pressures of regular periods are set up between the flange of the leading wheel and the rail.

In the second case, the truck runs independently of any such oscillations, follows the track irregularities very closely, and in consequence, is not affected thereby.

Two-wheel leading trucks with radius bar adjust themselves on curves so that they roll on different tread circles, while on tangent, the wheel seeks to roll on equal tread circles. They are, therefore, not led by the locomotive, but seek to find their own way and hence follow all track irregularities, as, for example, variations in alignment which result in a more flexible movement of the front end of the locomotive than obtains when the locomotive is led by rigid axles or one that can only be displaced laterally. At very great speeds, however, if the centering forces are insufficient, pronounced nosing may result. Leading trucks of this order are not, therefore, well adapted for high-speed locomotives. There have been many instances in the past where locomotives with leading trucks of this design have been attempted. The Mogul and Prairie types of locomotive have been displaced by the ten-wheel and Pacific types, and if present-day proof were required as to the action of two-wheel radial trucks maintained in fast service, it would only be necessary to attempt running Pacific type locomotives backwards, that is, with the trailing truck in leading position, which aside from the objections already mentioned, would be affected by the restraint of the four-wheel engine truck in trailing position. The four-wheel leading truck is now generally used for high-speed service, and sometimes we find the center pin eight or ten inches back of the center of the truck wheel base in order to relieve the front truck wheel of some of the wear.

Opinions differ as to the best adjustment that can be obtained. If the leading driving wheels wear their flanges sharp, it is quite evident that the engine truck is not doing its share of the work in curving; on the other hand, if the truck wheels develop sharp flanges, they do too much of the guiding. However, the differences in action

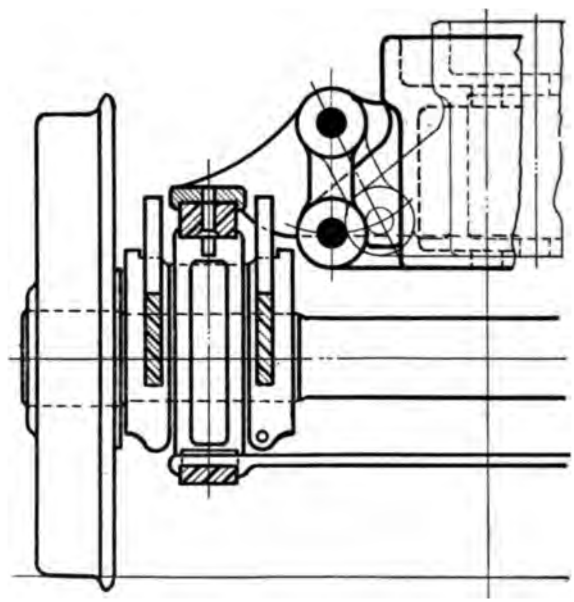

FIG. 21

are easily controlled through careful design of rockers or swing hangers. Most roads at this time are employing heart-shaped hangers or rockers which prevent of parallel motion and at the same time produce quite a horizontal pull to return the locomotive (or truck) to a central position, the measure of which can be judged by the angle which the center line of the hanger makes with the vertical.

Illustrated below are five classes of installation. In Fig. 21, the action between both hangers is the same, but is quite small, showing that the guiding power of the truck is not great. In Fig. 22 and 23 the stress is placed in one of the hangers at a time, depending

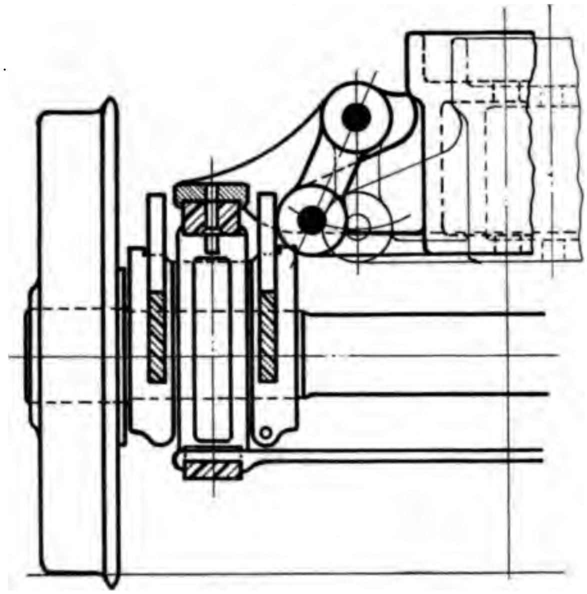

FIG. 22

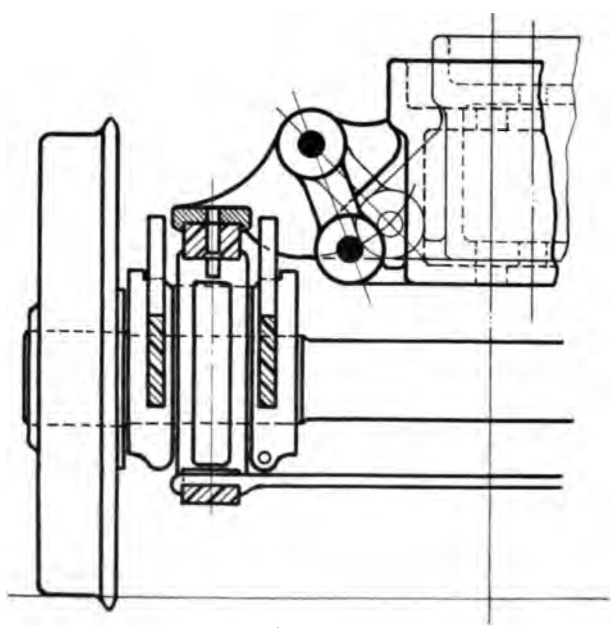

FIG. 23

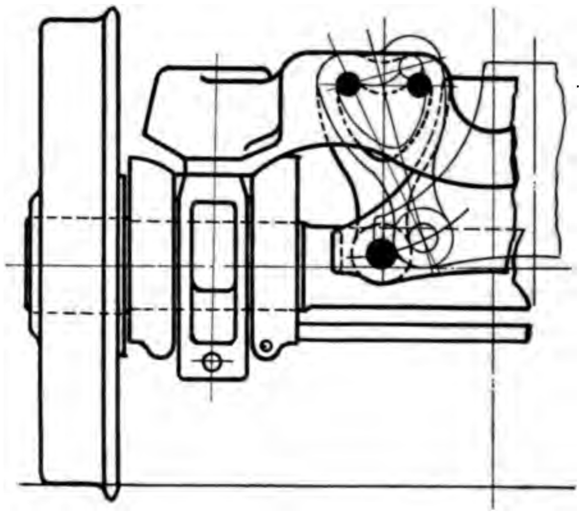

FIG. 24

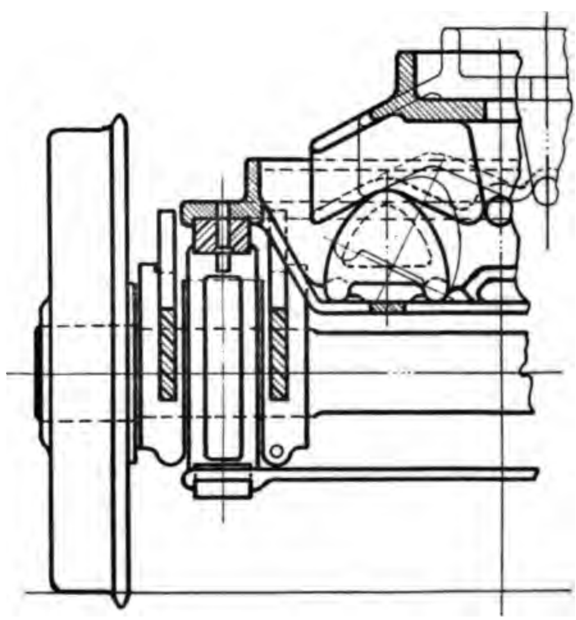

FIG. 25

on which side is to be guided. In Fig. 24, both hangers exert a similar force which can be increased by spreading the top centers so that the motion at all times is parallel. In Fig. 25, the action is similar to the fourth, except supports are reversed so that by spreading the base pivot points, increased stability can be obtained and therefore greater resistance. The difference in types of Figs. 24 and 25 is that they do not raise the front end of the engine at a rapid rate, and therefore provide for greater stability and avoid nosing or swinging back and forth when running at high speeds.

C. R. Henderson. In Par. 33 appears the remark that "the center of gravity of the total locomotive may be assumed to be some given distance above the rail, depending on the type of locomotive under consideration." This " assumption " is rather difficult to make in the case of a steam locomotive, where the various weights and heights form a complicated structure, in consideration of the vertical location of the center of gravity. In an electric locomotive, an estimate can probably be made in less time and with greater accuracy. If a similar locomotive is in existence, the vertical location of center of gravity can be determined experimentally in a few minutes, by first weighing the engine upon track scales, and then raising the rail on the narrow side of the scale platform by means of blocks, laid like ties on the side wall and the dead rail, so as to leave the other low rail alone in contact with the weighing mechanism, reweighing the engine on the low side and measuring the elevation of the raised rail, when the increased load on the low rail will indicate the horizontal displacement of the center of gravity, and the angle of inclination will give the point of intersection of the gravity line and the center of the engine, thus locating the center of gravity vertically. (This is fully explained in the writer's Locomotive Operation, p. 18.) This method was actually used to determine the height of the center of gravity of three different classes of steam locomotives, and the results were eminently satisfactory and the time required about one hour per locomotive.

In Par. 71, the eccentric wheel loading of trucks by centrifugal force is referred to, and the relation of this force to axle load is considered. It is important that the wheel on the inside rail should never lift or leave the rail, and the safety against such a condition may be assured by a simple geometrical diagram. If we remember that the line of force must pass through a direction parallel with the angle of swing made by the truck hangers, then the maximum swing of these hangers can be determined for the sharpest curve which must be traversed, and by drawing a line from the center of application of the truck load (bearing of center pin) parallel to the hangers when at their maximum swing, until it meets the level of the rails, the point of intersection of this line and the rail level will indicate the proportion of load on the outside rail. If the line should pass through the rail, then there will be no load on the other wheel or it will lift clear of the inside rail and a dangerous condition will result. The geometrical construction to determine this point is so simple that it should always be made when designing a new type of locomotive.

J. A. Hance.[1] The writer would call particular attention to Par. 2 which reads in part as follows: "The object of this paper is mainly to introduce the subject of the action of the running gear both on tangents and curves. It is intended to be more suggestive than final, and it is hoped it will raise discussion as collective observations and experience are necessary to arrive at a more accurate conclusion on the various points discussed."

The writer would like to emphasize this particular paragraph as the design of the running gear of the locomotive is frequently dependent on the conditions that are imposed upon the manufacturer. In very few instances do the designers have an absolutely free hand in the location of a number of parts, particularly the center pin connections between the two units of articulated locomotives.

Frequently the desired location of radius-bar pin center comes in conflict with other necessary parts of a locomotive and it is often necessary to make compromises in order to get all the parts properly placed.

The total wheelbase is usually governed by the length of the turn tables, transfer tables, roundhouse tracks, etc. Height and width are also limited by the allowable clearances on the roadway. This, of course, requires a great number of parts to be placed in the small space allowed.

Par. 4 reads in part as follows: "In the configuration of a locomotive running gear on a curve, we have two conditions, the statical and dynamical adjustments. At low and medium speeds the effect of centrifugal force is small as compared with the friction forces developed between the driving wheels and rail and thus the locomotive takes the statical adjustments. At high speeds the centrifugal forces

[1] Engineer of Tests, Baldwin Locomotive Works, Philadelphia, Pa.

cause the entire wheelbase to come more or less in contact with the outer rail, etc."

The writer believes that the wheels will run to the outer rail when the engine is in motion, due to the centrifugal force without having to overcome the friction between the wheel and the rail, unless the outer rail elevation is such that the force of gravity exceeds the centrifugal force or unless the truck has an inclination to keep some of the wheels away from the rail on account of the compromised position of the axles.

Par. 5 showing the curve that a single axle will probably tread is theoretically correct but if this formula were followed rigorously it would be necessary to have a different taper on the wheel for each curve that is passed over.

Par. 6 giving the formula for the radius of the curve which a truck with a pair of axles will turn is probably correct, but it gives a larger value than would be expected to be obtained from purely geometrical considerations; for example, a truck with one pair of axles located 6½ ft. apart would turn a curve of about 704 ft. radius if y is taken at ¾ of an inch which is one-half of the sum of the lateral play of the two axles, while from geometric considerations taking into account the gage of the road, we would expect this same truck to turn a curve of 350 ft. radius or about one-half that given by the formula. A curve of this small radius would not be used, of course, on a road where the vehicles were supposed to pass over it at customary speeds, but could only be used on switches or turnouts. Also, a truck will take a shorter curve than shown by either of these calculations on account of the rails spreading under the load.

Regarding Pars. 7 to 20, the writer considers it essential after the calculations are made that an actual drawing be made to scale, showing all of the wheels in their calculated position. He does not believe that it is possible by mathematical calculation, where so many equations are involved, to keep track of the relative position of all the parts.

Attention is called to the location of the radius-bar pin which cannot always be located in theoretical position on account of interferences with other parts of the mechanism.

Par. 38 and succeeding paragraphs referring to the striking angle of the wheel on curves and the influence of the diameter of the wheels on same are well taken and show very clearly that the diameter of the wheel should be considered.

The last part of Par. 51 regarding the use of the longest wheelbase possible should be very carefully considered, for the reason that the track is seldom perfectly circular and the allowances will have to be made for irregularities.

A Pacific type engine is shown in Fig. 15, but the action of the draw bar is not shown in this cut. This would very materially change the reactions when the engine is on a curve, as the drawbar on this type of engine is usually located quite a distance back of the rigid wheelbase.

The writer does not entirely agree with Par. 69. It is true that a great distance between the front engine truck and the rigid wheelbase will require less force to guide the engine around a curve. If this distance is very great the lateral deflection of the track will cause the center pin hangers to assume very great angles in rounding a curve, furthermore, the position of the center pin on a four-wheel truck is generally governed by the position of the cylinder saddle as the weight of the cylinders and part of the engine frame and boiler are carried on this truck.

Mention has been made of the advantages of lateral play in the boxes. The lateral play of six- and eight-wheel connected steam locomotives is very limited as this causes them to run out of line with the connecting rods.

George M. Eaton. In Par. 1 the author says: "The tractive performance of locomotives, both steam and electric is gradually reaching a satisfactory goal." It is important to realize the radical difference of tendency to be expected in further developments. In steam practice there is great incentive to multiply the number of axles in a rigid wheelbase, in order to concentrate more power under a single throttle lever, and with attendant complication of the problems involved in curving.

The desired fundamental of more power per train can be secured in electric locomotives on the multiple unit basis, and we can look with confidence toward an ultimate adoption of short, simple wheelbases with easy curving characteristics.

Referring to Par. 7, a value for Z is derived from Equation [5] together with the assumption that $y = y_1 = y_3$. It is rather important, in considering the mathematical superstructure erected on these assumptions, to realize that the assumption $y = y_1 = y_3$ is entirely arbitrary and probably involves a very considerable error.

The Roy diagram referred to in Par. 12 is a very valuable assistance in analyzing locomotive performance. The actual personal laying out of Roy diagrams gives one a clearer conception of the positions assumed by vehicles on curves, than can be obtained in any other way, short of detail observations of locomotives in service.

It is not clear why, as stated in Par. 12, "the ideal wheelbase is when the inner flange of the rear axle is brought just on contact and tangent to the inner rail." In this position, if any wheel coning exists, the slip of inner driver is increased, by virtue of the greatest wheel diameter being forced to travel on the shortest rail. Also the angle of approach of leading wheel to outer rail, and consequent flange wear, is greater than would be true if the rear inner wheel flange were materially clear of the rail.

It would be very interesting to know whether data are available checking the accuracy of the theory that "with short wheelbases, the flange guiding force on the front driver is increased." In the writer's practical experience, long wheelbase and rapid flange wear go hand in hand, but he has no case in mind where two vehicles have been approximately similar in all main features except wheelbase.

It may be of interest to note the precise reason why violent lateral oscillations may occur with extremely short rigid wheelbases. Some track irregularity, as for example, a bad rail joint, cocks the truck so that its leading axle runs toward the right-hand rail. When the flange throat reaches the rail, a slight climbing occurs. The shorter the wheelbase, the greater the angle of approach and the greater the chance of climbing. This climbing brings into action a larger effective wheel diameter and the right-hand wheel tends to run ahead, thus setting the leading axle to run toward the left-hand rail. As the right-hand wheel drops from throat to tread rail contact, a further impulse is given toward left-hand rail, due partly to the drop, but probably much more to the elastic recoil of the rail. If the speed and various natural time elements happen to be critical, a resonant condition will be established. In general, this can occur only when a short wheelbase and a comparatively great moment of inertia about a vertical axis exist concurrently. When true resonance occurs, unless power is cut off, derailment must result.

In Par. 20, a list of four structures is given under the heading of devices that offer a lateral resistance, which is a definite function of the lateral displacement of the truck.

This is somewhat misleading in connection with item (*d*) which is independent of truck displacement, and it is suggested that a separate heading be introduced between (*c*) and (*d*) calling attention to this fact.

The differential equations would be mathematically somewhat more satisfying if the exact value " $\tan \theta$ " were used in place of " θ " and the equations carried through on this basis.

In Par. 31 it is essential to realize that the force F_y is the force on the center pin and not the force on the flange of the leading wheel of a double truck. The leading wheel, being in advance of the center pin, enters the curve before the center pin, and starts with a mechanical advantage to swing the cab into the curve, reacting against the transverse adhesion of the trailing axle of the leading four-wheel truck. The action is quite complex and is further complicated, when as is usual, a banked spiral is encountered. This could be profitably analyzed by the author in a future paper.

The detail views of flange and rail contact, shown in Fig. 11 are worthy of very considerable expansion. The critical conditions on any given locomotive often occur with a worn flange instead of with a new flange.

The subject of gyrostatic action *vs.* flange pressure arises periodically and a brief analysis of its influence would be interesting in the future papers referred to. The writer hopes this line of study will be followed further in this country. It has been carried much further in France and Germany, whereas with our sharp curves and heavy adhesive weights and drawbar pulls, we have made less search for the fundamental relations, and have accepted comparatively short mileage between tire turnings as necessary evils.

THE AUTHOR entirely agrees with Mr. Fry that the development of the locomotive has been in greater part along entirely practical and empirical lines, but this may be said to be true of other lines in engineering, which now require the highest technical applications. Locomotive design, as in other branches of mechanical engineering design, is passing through a transition where the necessity of technical analysis and rational design is becoming indispensable. This is true especially in the proportioning and standardization of locomotive details, in the design of very large locomotives for minimum destruction to track and rigid requirements in performance, and in the development of electric locomotives, as Mr. Fry brought out.

The assumption as to the location of the instantaneous or friction center is basic in the development of this paper.

From a mathematical analysis as well as by tests on models by H. Uebelacker, the friction or instantaneous center has been found to be always located towards the rear axle of a rigid wheelbase. In the case of a four-wheel truck the rear axle runs approximately radially, whereas with several axles the instantaneous center is usually between the next to the last and last axle, unless the wheelbase becomes entirely geometrically constrained within the curve.

The location of the friction center is obtained from a consideration of the slippage of the wheels on the rails and the equation of motion of the rigid wheelbase. Neglecting the coning of the wheels, Let

G = the gage of track
x = the distance to axle from friction center I
uN = the resultant friction force at base of wheel A and B
F_y' = the lateral friction component at outer wheelbase A
F_y = the lateral friction component at inner wheelbase B
F_x' = the tangential friction component at outer wheelbase A
F_x = the tangential friction component at inner wheelbase B.

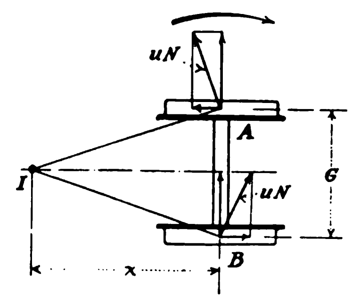

FIG. 26 WITHOUT TRACTION

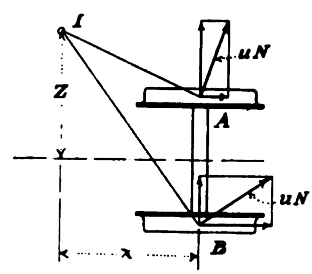

FIG. 27 WITH TRACTION

The effect of traction is to shift the instantaneous center to the side of the center line of track as in Fig. 27, the magnitude of the offset depending upon the amount of the tractive effort.

Without traction, the frictional components become,

$$F_y' = F_y = \frac{uNx}{\sqrt{x^2 + (G/2)^2}}$$

$$F_x' = F_x = \frac{uNG/2}{\sqrt{x^2 + (G/2)^2}}$$

With traction, assuming an offset z of the friction center from the center line of track, the friction components become,

$$F_y' = \frac{uNx}{\sqrt{x^2 + (z - G/2)^2}}$$

$$F_z' = \frac{uN(z_1 - G/2)}{\sqrt{x^2 + (z - G/2)^2}}$$

$$F_y = \frac{uNx}{\sqrt{x^2 + (z + G/2)^2}}$$

$$F_z = \frac{uN(z_1 + G/2)}{\sqrt{x^2 + (z + G/2)^2}}$$

Consider a rigid wheelbase of four pairs of drivers as in a consolidated locomotive, under a tractive effort Z. To rotate the rigid wheelbase about the friction center a guiding reaction Y on the front outer driving wheel is required. Further let the distance from the first to the rear axles be d_2, d_3 and d_4. The equations of motion are: For moments about the front axle,

$$(F'_{y2} + F_{y2})d_2 + (F'_{y3} + F_{y3})d_3 + (F'_{y4} + F_{y4})d_4 + (F'_{z1} - F_{z1})\, G/2 + (F'_{z4} - F_{z4})G/2 = 0 \qquad [146]$$

and further

$$F_y' + (F'_{y2} + F_{y2}) + (F'_{y3} + F_{y3}) + (F'_{y4} + F_{y4}) - Y = 0 \qquad [147]$$

$$(F'_{z1} + F_{z1}) + \dots\dots (F'_{z4} + F_{z4}) - z = 0 \qquad [148]$$

With a given tractive effort Z and substituting for the friction components, the unknowns become x, z, and Y and therefore the distance to the friction center x may be located. If x_0 = the distance from the front axle to the friction center, knowing the spacing of the axles, the distance x to any axle becomes a function of x_0. To solve directly for x_0 would be extremely complicated, hence the substitution of a trial value is the most satisfactory procedure.

We thus see that the effect of traction is to modify the friction center, but since the lateral reactions computed under traction are less than without traction, the effect of tractive force in modifying the friction center has been neglected in this paper. The position of the friction center is very slightly affected by variation of rail pressure on one side over the other for any given axle, but this latter can be entirely neglected practically.

The general solution for the friction center and corresponding guiding reactions for a rigid wheelbase is given by the following equations:

$$\Sigma F_y + C - Y = 0. \qquad [149]$$

$$\Sigma F_z - Z = 0 \qquad [150]$$

$$\Sigma(F_y d_x) + \Sigma(F_z G/2) + Cc = 0 \qquad [151]$$

where d_x = distance from front axle to any intermediate axle
C = centrifugal force
c = moment about center of front axle.

$$F_{x1} = \frac{uN_1G/2}{\sqrt{(x_0 - d_{x1})^2 + (z \pm G/2)^2}}$$

$$F_y{}'_n = \frac{uN_nG/2}{\sqrt{(x_0 - d_{nx})^2 + (Z \pm G/2)^2}}$$

$$F_y = \frac{uN_1(x_0 - d_{x1})}{\sqrt{(x_0 - d_{x1})^2 + (Z \pm G/2)^2}}$$

$$F_{yn} = \frac{uN_n(x_0 - d_{xn})}{\sqrt{(x_0 - d_{xn})^2 + (Z \pm G/2)^2}}$$

The unknowns are x_0, Z and Y, and therefore a solution is possible.

When the friction or instantaneous center is determined geometrically, we have from Equations [4] and [5],

$$Z = \frac{d_n}{2} - \frac{2R_y}{d_n} = d_n - x_0$$

where $d_n = a$ = length of rigid wheelbase, therefore

$$x_o = \frac{d_n}{2} + \frac{2R_y}{d_n}$$

In this case however, a reaction Y_0 is exerted by the inner rail on the rear driving axle and the equations become,

$$\Sigma F_y + C + Y_0 - Y = 0 \quad \text{[152]}$$
$$\Sigma F_x - Z = 0 \quad \text{[153]}$$
$$\Sigma(F_y d_x) + \Sigma(F_x G/2) + Cc + Y_0 d_n = 0 \quad \text{[154]}$$

where the unknowns now become Y_0, Y and Z, the instantaneous or friction center being determined geometrically. From these equations a complete determination of the lateral reactions in a curve is possible. The effect of unequal flange play, journal play and the various lateral oscillations and impact on entering a curve are not considered. The latter may considerably increase these reactions.

The position of the friction center can be determined experimentally by the following method. From Equation [17] we have

$$Z = a - \frac{Rd}{b} + \frac{b}{2} = x_0 - a,$$

hence

$$x_0 = 2a - \frac{Rd}{b} + \frac{b}{2}.$$

Therefore, provided the flange plays of the driving wheelbase and guiding truck are the same and a correction made for journal play, by noting the swing of the front truck d, we may determine the position of the friction center.

The length of radius bar is given by Equation [20]. When $Z = \frac{a}{2}$, we have $r = \frac{(a + b)b}{a + 2b}$, which is the ordinary formula used in locomotive construction. To show the deviation in length, we will take a consolidated locomotive with a wheelbase of 186 in. and the distance from front driver to front truck 98 in. By the ordinary formula $r = 73$ in. The maximum length from Equation [21] becomes $r = 81$ in. Thus we see the required length of radius bar may vary 10 per cent or over, according to the position of the friction center of the rigid wheelbase. If the flange plays for the various axles differ, the length of radius bar is again modified.

In regard to the length of radius bar for a trailing truck, since the lateral swing of this truck is relatively small compared with that of a leading truck and the swing is further of a magnitude comparable with the more or less variable and indeterminate plays, due to wear, etc., its length is not of importance.

Mr. Woodward's emphasis of the importance of stability on a tangent track cannot be over-estimated. This consideration was purposely avoided since it is a separate and more complicated phase on the subject of lateral reactions. Of course the final layout of a wheelbase is a compromise between

(1) General layout construction limitations, type of drive, etc.
(2) Arrangement for proper adhesion and equalization
(3) Distribution of wheel loads for minimum track stress
(4) Lateral oscillations and stability on tangent track, and finally
(5) Static adjustment on curves.

However stability on tangent track and proper curving are more or less inter-related. The oscillation of a locomotive takes place about a vertical axis through the center of gravity of the total locomotive minus the trucks. The danger of oscillation increases with the polar moment of inertia and the possibility of the natural frequency agreeing with the impressed frequency, as due to rail joints or other impressed periodic causes. The lateral elastic reaction of the rail decreases with an increase of its moment arm. Since the trucks are external to the rigid wheelbase system a *constant* lateral reaction (i.e. independent of the displacement and therefore *not* elastic) acts as a very effective damper to these oscillations. As for curving, except for very high speeds, it was pointed out that the guiding requirements amounted in greater part to turning the rigid wheelbase about its friction center and therefore was more or less independent of the radius of curvature or truck swing. Therefore a constant lateral reaction undoubtedly approaches an ideal condition and the author quite agrees with Mr. Woodward in regard to this requirement.

Due to the small moment arm as well as swing of the trailer truck, its effect on curving the rigid wheelbase is at best small and the particular type of lateral device is relatively unimportant from the point of view of curving. With lateral oscillation, however, the moment arm is greater, being now about the center of gravity rather than the friction center, and the initial reaction is important. Therefore, a centering device of a constant resistance type (a non-elastic reaction and, therefore, corresponding to a frictional damper) approaches the ideal requirement.

Giving a definite lateral motion and reaction to either the front or rear driver of a very long wheelbase, as mentioned by Mr. Woodward, is excellent. It reduces the rigid wheelbase length with additional guiding.

Mr. Sillcox has brought out the importance of the contour of the flange and tire, and the effect of several types of wear, as well as the necessity of maintaining, by proper equalization, the normal loads constant.

Since the maximum guiding capacity of an engine truck depends upon the maximum lateral reaction that can be exerted by the lateral device of the truck and this in turn depends upon the maximum lateral rail reaction capable on the forward outer truck wheel, we are, therefore, immediately concerned with the maximum ratio of this reaction to that of the weight on the truck or wheel. The maximum lateral reaction capable of any guiding wheel is a function of the normal wheel weight and the contour of the wheel flange and tread. Variation of this contour changes the ratio of lateral reaction to weight on wheel, as well as the magnitude of wear on the wheel tread. Variation of the wheel weight causes a proportional change in the allowable lateral guiding force. Hence the importance of maintaining through proper equalization a constant load on guiding wheels.

The relation of the guiding reaction to the weight and contour of a given wheel is given by Equation [71], which may be stated as follows:

$$Y = N \tan (\phi - d) = N\left[\frac{\tan \phi - f}{1 + f \tan \phi}\right]$$

when $f = \tan d =$ the cœfficient of friction.

Values of the ratio of N for different cœfficients of friction and the slope of the flange contact are given in the following table:

f	$Y/N = 45°$	$Y/N = 60°$
0	1.000	1.732
0.1	0.818	1.392
0.2	0.666	1.145
0.3	0.538	0.944

where N = wheel load

Y = guiding reaction

The contour of the standard A.M.C.B. wheel gives a slope approximately equal 60 degrees. Hence the minimum safe axle load is 2.12 times the maximum guiding reaction. It is well to maintain axle loads from 2.5 to 3.0 times the maximum guiding reaction to prevent any possibility of derailment.

It is to be noted that when the obliquity of the wheel is small, the flange contact takes place at the tread of the wheel. In this case supposing the radius of the arc of contact of wheel flange and rail to be r' and the angle to any differential area of contact along the radius r', with respect to the horizontal to be ϕ, and the radius of the wheel r, the total resistance to forward motion, becomes,

$$X = \int_0^{\frac{\pi}{2}} \frac{up\, r'\, d\phi \cos\phi\, r' (1 - \sin\phi)}{r},$$

where

$$p = \frac{Y}{r'}$$

u = coefficient of friction,

since the projected bearing pressure in the direction of Y must always equal the actual bearing pressure.

Therefore

$$X = \frac{uYr'}{r}\int_0^{\frac{\pi}{2}} \cos \phi(1 - \sin \phi)d\phi$$

$$= \frac{uYr'}{r}\left| \sin \phi + \frac{\cos 2\phi}{4} \right|_0^{\frac{\pi}{2}} = \frac{uYr'}{2r}.$$

The work due to flange friction for a velocity v ft. per sec. becomes,

$$Xv = \frac{uYr'v}{2r} \text{ ft-lb. per sec.}$$

In regard to maintaining constant wheel loads for axles subjected to large guiding reactions, much depends upon the proper design of the equalization. Since the variable loads are primarily due to impact, such as passing over rail joints, etc., it is very important that the spring system be designed for proportionate deflections and corresponding changes in load. That is the flexibility (i.e. deflection per unit load) for the various springs should be equated or made proportionate, according to the design of the spring system.

Mr. Hance places special emphasis on the various construction limitations that frequently offset the possibility of realizing ideal wheel spacing and the proper placing of the center pin of articulated locomotives. Though it is true the design must be a fair compromise, the location of the center pin of articulated locomotives is of unusual importance and takes precedence over other considerations, since a wrong location will cause negative guiding. In general the location of the center pin must be forward of the intersection of the center lines connecting wheel bases, that is $c > \frac{1}{2}(L + b)$ and $d > \frac{1}{2}(L - b)$, see Par. 25 and 73.

The author does not agree with Mr. Hance in regard to the wheelbase running to the outer rail and the independence of friction. The centrifugal force and gravity component are merely additional forces in the various equations for the adjustment of the wheelbase. The centrifugal force changes the position of the friction center as well as modifies the guiding reaction, but it is always entirely comparable with any of the various friction forces and unless the speed is high the effect of the friction forces is by far the greater.

The radius of curvature, as given in the discussion in Par. 5 and by the Roy method, should not be regarded as the minimum value, since a very much shorter radius is entirely possible without undue binding. When the friction center is assumed at the rear axle, the rear axle runs radially and the longest wheelbase in which the guiding is entirely effected by the outer wheel of the front axle is that when the inner wheel of the rear axle is just brought in contact with the inner rail. With a longer wheelbase the inner rail reacts outward on the rear inner wheel and the position of the wheelbase is then determined geometrically. It is the minimum radius by which a curve can be traversed without binding, that is, free to turn about its friction center. It is to be noted that although the friction center was assumed located approximately at the rear axle, its position may be considerably modified by the magnitude and position of the lateral and centrifugal forces brought on the truck. In general, unless complete binding occurs, when the inner rail is brought in contact with the rear axle, the guiding reaction on the front axle and friction moment is decreased and, therefore, at medium speeds, the maximum guiding is required on the curves of larger radius, rather than the smaller or sharper curves.

The diagram of the lateral reactions on a Pacific type gear represents an approximation of the lateral forces when the engine is coasting. Under traction the guiding reactions are in general reduced and it is doubtful whether the small lateral component of the drawbar pull will be sufficient to compensate for this reduction, or cause an increase of the guiding reaction. Of course, if the drawbar is at a considerable distance to the rear from the rear axle of the rigid wheelbase and on a large degree curve, the guiding reaction might be considerably increased.

Mr. Eaton has brought out the trend of development of wheelbase arrangement in steam and electric practice and the more favorable possibilities of an articulated running gear for electric locomotives. Unquestionably for slow speed operation on very sharp curves, an articulated running gear with short rigid wheelbases gives an ideal type with guiding reactions and flange wear. On the other hand the nosing of this type of running gear is bound to be excessive at high and medium speeds. A long rigid wheelbase is considerably more stable and greatly reduces the lateral guiding reactions due to lateral oscillations on tangent track. Therefore, for ordinary service for medium or high speeds, a medium length rigid wheelbase (sufficient for the sharpest curves without undue binding) with proper guiding trucks seem preferable to an articulated locomotive. On the other hand for slow speeds and sharp curves, an articulated type is preferable. With a very powerful locomotive on main line service, with many driving axles required, a partial articulation with medium length wheelbases would seem the best compromise.

In Par. 7, the rigid wheelbase is geometrically determined by the length of wheelbase a and the flange play y. With the same flange play for all axles, $Z = (a^2 - 4Ry)/2a$. When $a = 2\sqrt{Ry}$, we have $Z = 0$, and the rear axle runs radially as in Par. 6. In this case the wheelbase turns about the true friction center as compared with the geometrical center I in Par. 7. It is true that considerable error may be involved in the assumption $y = y_1 = y_3$.

The term "ideal wheelbase" in Par. 12 is misleading. The position of the rigid wheel is determined either by the friction and equilibrium equations of the rigid wheelbase or geometrically as in

Par. 7, depending whether the inner rail exerts a lateral reaction. When the wheelbase is subjected only to a front guiding force, there being no reaction on the rear axle, the friction center varies with the position and magnitude of the external forces exerted on the rigid wheelbase. In general the friction center will be found close to the rear axle and for design purposes we have assumed it at the rear axle. Therefore the rear axle is assumed to run radially when no reaction is exerted on the rear axle, that is from no contact to just contact of the inner rail with the rear axle. A longer wheelbase will cause a geometrically determined position as in Par. 7. The term "ideal wheelbase" refers to the free wheelbase as against the constrained wheelbase in Par. 7. Since often the friction center is considerably forward from the rear axle, the rear axle will not be brought in contact with the inner rail for wheelbase lengths considerably greater than $a = 2\sqrt{Ry}$.

The guiding of a rigid wheelbase may be classified into two methods; the guiding by its own leading driver or the guiding by engine trucks, external to the rigid wheelbase. In the former, lengthening the wheelbase decreases the flange guiding force, as shown by Equation [97]. On the other hand when the guiding is entirely by the lateral reaction of engine trucks, the lengthening of a wheelbase increases the guiding reaction.

In conclusion the author wishes to express his appreciation for the particular interest and kindly advices given in the preparation of this paper by Mr. George Eaton and Mr. Frederick Urban. Mr. Urban has extended the use of the Roy diagram to a very comprehensive layout of the wheelbase for various conditions of journal and flange wear on the new Chicago, Milwaukee and St. Paul electric locomotives.

No. 1777

THE ARMOR-PLATE AND GUN-FORGING PLANT OF THE U. S. NAVY DEPARTMENT AT SOUTH CHARLESTON, W. VA.

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The magnitude of such a project as the establishment of the Navy-owned, built and operated Naval Ordnance Plant at South Charleston, W. Va., has made it impossible for the author to give details on all points of the development of the plant. He describes briefly the establishment of the projectile plant and preliminary steps leading up to the actual construction of the armor-plate and gun-forging plant, a general description of which is the main object of the paper. The general layout of the plant and transportation facilities are covered and then the author passes to a more detailed discussion of the unusual features of design and construction. Among these are the arrangement of the stockyard alongside the open-hearth building, the 30-ton electric furnaces for duplexing, the electric traveling cranes to carry 400,000-lb. ingots, the original "H"-type of forge shop and treatment plant combined, the 14,000-ton steam-intensified hydraulic press, the twenty 15 ft. wide Carbottom furnaces, the machine shop with three 106 ft. wide aisles, and the gun treatment building, the high portion of which contains a 75-ton crane of 104 ft. span on rails 165 ft. above the floor level which serves a 10 by 105 ft. high electric-treatment furnace placed in a concrete pit 55 ft. deep. The 23 electric traveling cranes, twenty of which are 75-ton or larger and three of 250 tons capacity, are discussed. The 25,000-kw. electric substations, the 6500-hp. boiler plant and the 25,000,000-gal. reservoir are also discussed briefly.

The paper concludes by summarizing the various steps in the development, outlining the organization of the project, which was handled entirely within the Navy Department, giving in round numbers construction costs, and reviewing briefly the progress of construction of the plant which is now substantially completed.

SEVERAL years ago the papers were again full of the historic discussion as to whether the Navy Department should attempt to manufacture armor plate for itself or continue to go to the three non-competing firms who were able to make armor in this country, and pay the price asked. With the strong backing of Secretary

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Daniels and Senator Tillman, and despite much opposition by the manufacturers, the public at large, and certain Navy circles, Congress directed that the plant be built. Shortly thereafter the war came on and the project was generally forgotten in the rush of ordnance manufacture for the nations of Europe and for this country, when we finally entered.

2 On August 30, 1917, ground was broken by the Secretary of the Navy for the construction of a projectile plant as the first unit of the U. S. Naval Ordnance Plant at South Charleston, W. Va. The location had been determined by a Naval Board appointed for that purpose, after a very thorough investigation of possible sites. The specifications of the War College had required that it be built 200 miles inland, which ruled out the Atlantic Seaboard, the Great Lakes District, and Pittsburgh. The main advantages of South Charleston lay in its being in the center of the rich coal districts of West Virginia, in the proximity of iron mines and blast furnaces, and in the availability of natural gas and oil. The reservation lies between the Chesapeake & Ohio Railroad and the Great Kanawha River, which is navigable from its junction with the Ohio River to a point several miles above Charleston.

3 The projectile plant was completed and put into operation in the spring of 1918, since when it has operated satisfactorily, producing principally 6-in. gun forgings, counter-recoil cylinders, 16-in. armor-piercing projectiles, air-flasks for torpedoes, steel ingots up to 18 tons, and at the present time, the entire demand of the Navy Department for armor bolts.

4 In June 1918 it was decided to vigorously push the construction of the main plant for the production of armor plate and major-caliber gun forgings. The writer was summoned from Erie, Pa., where he had supervised the construction of the Erie Forge and Steel Company and other steel plants for the Navy, and placed in charge of the design and construction. A designing and drafting force was built up around his engineering assistants from Erie and his previous work, and half a dozen draftsmen who had worked on the projectile plant. During the summer of 1918 the layout and the general design for the main buildings were completed, and by November 1, approved by the bureaus concerned.

5 It was decided that due to existing conditions in the material and labor markets, the impossibility of writing specifications and getting lump-sum bids without great delay, and on account of the general disrepute into which cost-plus contracts for Govern-

ment plant construction had fallen, the work should be done directly by the Navy without resorting to a general contractor. A construction division was organized and construction work was under way by October 1, 1918.

6 The armor-plate and gun-forging plant has been designed to produce armor plate of the heaviest type, completely finished, ready to attach to battleships; and major-caliber gun forgings, up to 20-in., 50-caliber in size, rough-machined, which will be sent to the Washington Navy Yard for finish machining. It will employ about 3000 when completed and in full operation.

PURPOSE OF THE PAPER

7 The purpose of this paper is to give simply a general description of the armor and gun-forging plant as designed and built, inviting attention to unusual features in design and construction. It cannot pretend to be a treatise on the manufacture of armor plate in general, nor can it describe the specific methods which will be followed at South Charleston. These can later be described by those in active charge of operation. The magnitude of the project precludes the possibility of going into details.

REASONS FOR PLANT

8 The United States Naval Ordnance Plant is not a "war plant" in the modern usage of the term, but a permanent Navy institution. Despite previous feelings to the contrary, which were doubtless shared by the engineers of the country at large, the writer has become convinced during the progress of the work that the project is abundantly justified, and for the following reasons, although it is not the purpose of this paper to invite discussion on the Government's policy in establishing the plant.

9 First, additional capacity for the manufacture of armor plate and gun forgings has been provided. The capacity of the three previously existing plants amounts to 28,000 tons of armor per year. The present capacity of the armor plant is 10,000 tons of finished armor plate per year, which can readily be expanded to 20,000 tons per year at little additional cost. In addition to this is the forging capacity for 10,000 tons per year, or 24 sets of major-caliber gun forgings, rough-turned, and the capacity of the projectile plant for minor-caliber gun forgings, large projectiles, and various special forgings and castings. The entire armor and gun-forging

plant, when completely equipped, will represent a value of about $20,000,000. As the Navy has been paying approximately $500 a ton for armor plate and on certain occasions nearly twice this amount and as a modern battleship requires somewhat more than 10,000 tons of armor plate, the plant has cost only the equivalent of the armor plate alone for but three or four battleships.

10 Second, cost of armor manufactured by the Naval Ordnance Plant should be less to the Navy than the price now paid to commercial manufacturers and will afford a definite means of determining what the fair price for armor and gun steel to be paid steel manufacturers by the Navy should be. With the experienced operating personnel which is being assembled, armor plate and gun forgings will unquestionably be produced in a satisfactory manner and it is firmly believed at not greater manufacturing cost than by commercial concerns. The overhead cost of the armor plant should be no greater than the overhead of the outside contractor who, in addition, has to maintain his sales department, pay bonuses, etc. The Government will undoubtedly save the profit of the commercial manufacturer on steel made at South Charleston.

11 Third, the Naval Ordnance Plant will be a large-scale experimental and research laboratory which can continually strive to improve the product and simplify manufacturing methods. Commercial manufacturers are interested primarily in tonnage, although with but few exceptions the manufacturers have done everything to aid in developing and producing the best armor possible. The Navy can afford to spend considerable money to experiment on a large scale in the development of new processes and improved product and can write new specifications which it will know through experience can be met. In this connection it is interesting to note that the first group of 16-in. armor-piercing projectiles made at the Naval Ordnance Plant passed the test at Indian Head and that the plant is successfully making air flasks for torpedoes. These are two products requiring the highest skill, and several of the large steel manufacturers who have been unable to meet the tests with their products have already sent committees to South Charleston to study the practice of the plant.

12 Fourth, a very great additional advantage in the Government-owned and operated plant is the fact that improvements in product or methods can, when desired, remain the property of the United States.

GENERAL LAYOUT

13 The reservation consists of somewhat over 200 acres of land located between the Chesapeake & Ohio Railroad and the Great Kanawha River about five miles below Charleston, at South Charleston, W. Va. It is divided into a north unit of approximately 40 acres and a south unit of approximately 160 acres, by Eighth Avenue, which is the main road and which carries the trolley line between Charleston and St. Albans. The projectile plant occupies the north unit, and was getting into operation about the time active work was started on the designs for the armor and gun-forging plant in the south unit. It cost somewhat over two million dollars and consists principally of a forge and foundry, approximately 130 ft. by 560 ft., containing three 6-ton Heroult electric furnaces, two 60-in. cupolas, a small brass foundry, a 3000-ton press and a 500-ton press, eleven forge and five regenerative Carbottom annealing furnaces; a machine shop, 140 ft. by 400 ft., completely equipped for machining minor-caliber gun forgings, large projectiles, and general machine-shop work, and a heat-treatment shop 92 ft. by 153 ft.

14 The general layout of the south unit was determined first by the requirements of manufacture in modern straight-line methods, and secondly by the topography of the site. Fig. 3 shows the general layout of the armor and gun-forging plant. The four main buildings, namely, the open-hearth, the forge and furnace, the machine shop, and the gun-treatment building, are arranged in parallel on a shuttle track which runs at right angles to the major axes of the buildings and connects them all together. This track will be the backbone of the manufacturing processes. All main buildings have been placed so that future expansion to double the capacity of the plant may be readily made.

15 Scrap and pig iron will be received from the Chesapeake & Ohio Railroad, at the southeast corner of the reservation, sorted from other incoming material in the classification yard, and stored in the stock yard alongside the open-hearth building. Steel will be melted in the open-hearth furnaces, refined in the electric furnaces, and cast into ingots in the pouring pit in the open-hearth building. From there the ingots will be transferred to the forge and furnace building by way of the shuttle track for heating, preparatory to forging into armor plate or guns under the presses located in the center of the forge and furnace building. The armor-plate forgings

FIG. 1 THE UNITED STATES NAVAL ORDNANCE PLANT AT SOUTH CHARLESTON, W. VA. VIEW LOOKING IN A GENERAL EASTWARDLY DIRECTION, AUGUST 2, 1920.

Building No. 1, Machine Shop   Building No. 2, Gun-Treatment Building

In the background toward the north may be seen the projectile plant, with the barracks for the construction forces in the foreground. Immediately in front of the machine shop is the oxy-hydrogen building and gasometers in process of construction. To the right of the machine shop is the concreting plant for the gun-treatment pit, only a portion of the steelwork having been erected. The steelwork for the high portion, when completed, will reach a point three times the height of the adjacent parapet wall of the machine shop, i. e., somewhat over 200 feet, and somewhat higher than the twin chimneys for the boiler plant.

FIG. 1A THE UNITED STATES NAVAL ORDNANCE PLANT AT SOUTH CHARLESTON W. VA. PHOTOGRAPH IS RIGHT HALF OF PANORAMA ON OPPOSITE PAGE

Building No. 3, Forge and Furnace Building. Building No. 4, Open-Hearth Building. Building No. 6, Boiler House. Building No. 7, Foundations for Service Building.

The fill for the shuttle track is shown leading from the forge shop to the south side of the machine shop. Note the open monitors on the low and high portions of the forge and furnace building.

will then be carbonized, annealed, tempered, hardened, and ultimately sent over to the machine shop on the shuttle track where the armor plate will be machined and finished, ready for attaching to battleships. Gun forgings will be green-annealed in the south aisle of the forge shop, sent to the north aisle of the machine shop for rough-machining, and thence to the gun-treatment building for heat treatment prior to shipment to the Navy Yard in Washington for finish-machining and assembling.

16 The service building, which comprises the main electric substation, the air compressors, the water-supply distribution pumps, and the boiler plant, which provides steam for the steam-intensified hydraulic presses and for heating the machine shop, are located as nearly as possible to the center of gravity of their respective loads.

17 Twin buildings, one of which will be used for general stores and the other for the blacksmith shop, pattern and templet making and storage, have been located in the southeastern portion of the south unit accessible to the railroad system and the forge shop.

18 The maintenance shops in which the electricians, pipefitters, riggers and carpenters are to be located, and the locomotive repair shop and roundhouse, will be combined in one building at the approximate center of the reservation on the main roadway and adjacent to the Railroad "Y."

19 The western portion of the south unit as divided by the main railroad track running north and south, contains a large storage yard and the "skull cracker."

20 The reservoir and settling basin (Fig. 2) for the industrial water-supply system has been made by throwing two earth-filled dams across the western end of the gully which cuts diagonally across the reservation.

21 An administration building of sufficient size to house the executives for the entire ordnance plant was built at the same time as the projectile plant and located in the south unit on Eighth Avenue. The balance of the frontage on Eighth Avenue is occupied with temporary buildings for construction offices, stores, the garage, barracks to house about 1000 men on construction work, and Bungalow Park, in which 32 bungalows were erected for the occupancy of the supervisors of the construction division.

TYPE OF STRUCTURE

22 In general, the buildings consist of a structural-steel framework unusually heavy on account of the size of the cranes, on con-

crete foundations some of which contain upward of 200 cu. yd. of concrete enclosed by walls of specially designed hollow red tile building block, 5 by 12 by 8 in. Fifty per cent or more of the surface area of the walls is steel sash. The roof decks are of gypsum composition, cast in place and covered by waterproof roofing. In round numbers about 50,000 cu. yd. of concrete has been placed in the various foundations, 25,000 tons of structural steel erected, and in excavation and grading, upward of 500,000 cu. yd. of earth moved.

Fig. 2 Reservoir Looking West from Machine-Shop Roof, July 14, 1920

Shops appear at left. At extreme right is glimpse of bungalows erected for construction supervisors

The total area under roof is approximately 700,000 sq. ft. for the main buildings alone.

TOPOGRAPHY

23 The topography of the site is generally level excepting for two gullies which cut diagonally across the property to a depth of 25 ft. below the general elevation. This runs from about 595 to 600 ft. above sea level. The floor elevation of all main buildings, excepting the machine shop and gun-treatment building, which are

3 ft. lower, is at 598. The soil consists of sand and clay, ground water being at about elevation 562. As a result of various tests made, a soil pressure of 6000 lb. per sq. ft. for dead loading and 8000 lb. per sq. ft. for live loading was allowed for all foundations.

TRANSPORTATION FACILITIES

24 It was evident very early in the design period that one of the most important features of the entire plant would be the arrangements for efficiently handling and moving unusually heavy and awkward loads. For example, one armor-plate ingot which will be made about six times a year, will weigh upward of 400,000 lb. and tube forgings for the major-caliber guns will weigh up to possibly 100,000 lb. and run in length up to nearly 100 ft. An armor plate in the making must be handled something like fifty separate times and as the average weight of the finished plates is in the neighborhood of 50 tons, very special attention was required in the design of the electric traveling cranes and the railroad track system.

25 The railroad system, which totals something over seven miles in length, is outlined on Fig. 3. A classification yard of ten tracks including incoming and departure tracks has been provided. Provisions have been made for tracks entering into either end of practically every crane aisle. The radii of curves have been made as great as possible, and in no case of major importance, less than 350 ft. A complete loop within loop system has been developed so that a wreck or stoppage on any main track will not hold up the operation of the plant or of any individual major building. The shuttle track already referred to is connected to the main track system north of the forge shop, as shown. A three-way reinforced-concrete approach trestle will lead to the charging floor level of the open-hearth building.

26 The major specifications for the twenty-three overhead electric traveling cranes are given in Fig. 4. Cranes were made by three separate manufacturers and their total cost was slightly over one million dollars. The crane motors and controls were purchased separately from the cranes in order that the entire electrical equipment might be the product of one manufacturer, to simplify operation and the carrying of spares. Only seven different sizes of motors are used. Two 200-hp. motors are required for the main hoist of the high treatment crane. All braking is electrodynamic, mag-

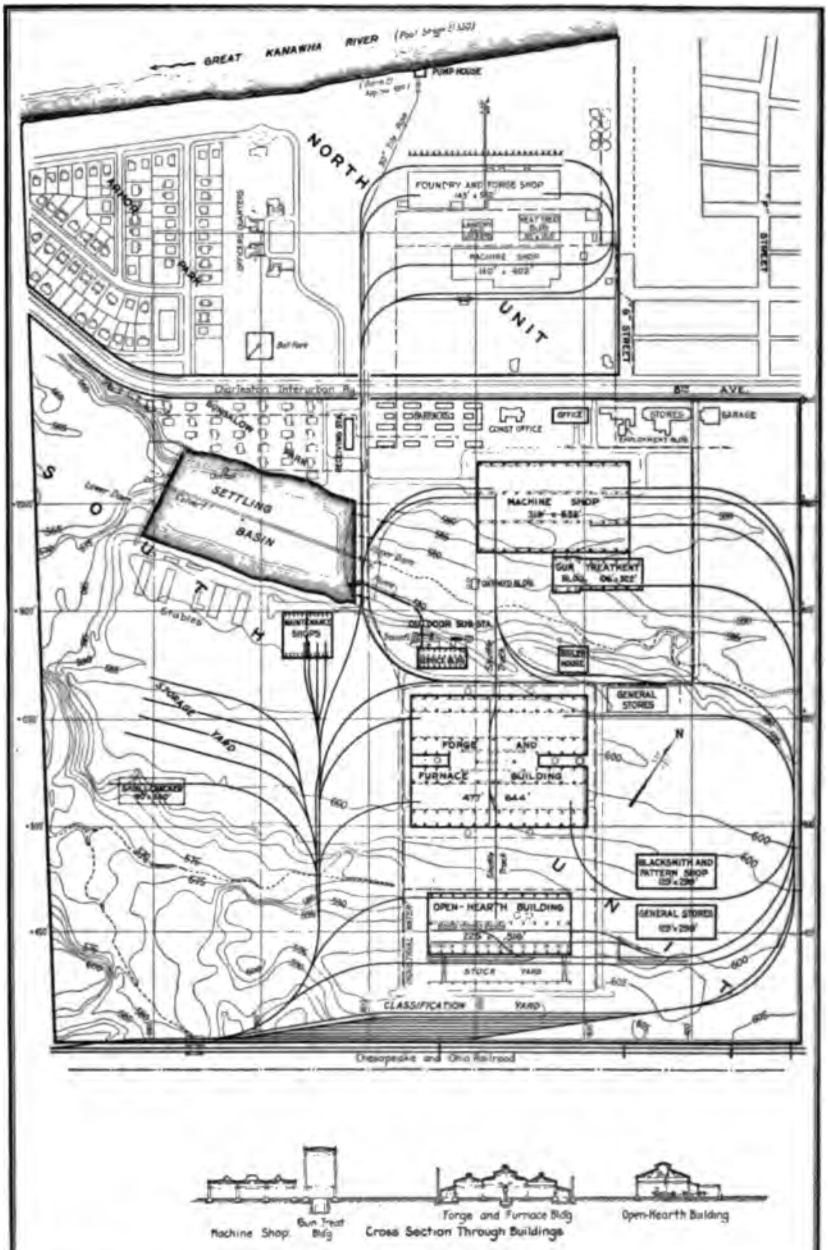


Fig. 3. General Layout of Plant Showing Track Plan

netically controlled. Each main aisle of the four main buildings is provided with erecting girders for the cranes located in the roof trusses.

27 While by far the larger portion of the traffic at the plant will be by rail, a system of concrete roads 20 ft. in width has been developed in order to provide auto-truck service to all main buildings, and for employees. The main entrances to both the north and south units will be from Eighth Avenue, where the plant railway crosses. A central receiving building with a special siding has been built near this point as shown on Fig. 3, for the unloading of less-than-carload shipments and to permit distribution by auto truck.

GENERAL DIMENSIONS OF PRODUCT

28 As a matter of general interest there is a total of nearly 300 plates in a modern battleship, the total weight being somewhat over 10,000 tons. Over 80 tons of bolts and nuts are required. The plates vary in thickness from $4\frac{1}{2}$ in. to 18 in. and in weight from a few tons up to 70 tons. The largest barbette plate on the new ships is 28 ft. long when finished, and approximately 30 ft. 6 in. in length as sprayed. The 18-in. turret port plate weighs nearly 120 tons as sprayed, although afterward, when ports are broken through, it weighs less than 60 tons.

29 The maximum width of plates is determined by the clearance between the columns in the forging press; at South Charleston it will be very nearly 14 ft.

30 Although the plant and equipment have been provided of sufficient size to manufacture 20-in. 50-caliber guns, the present program will be to make the forgings for twenty-four 16-in. 50-caliber guns per year. There are 14 forgings for a 16-in. gun, a liner, a tube, a jacket, and 11 hoops, four of which are locking rings. Rough-bored and turned, the liner will weigh approximately 46,000 lb., the tube 62,000 lb.; the jacket 34,000 lb.; the hoops will weigh from 30,000 to 42,000 lb. each, and the locking rings from 300 to 1300 lb. each. The tube and liner forgings are somewhat over 800 in. in length.

31 The ingot for the tube forgings must be something over 60 in. in diameter and weigh approximately 180,000 lb., of which 40,000 lb. is the weight of the sink head and 140,000 lb. the weight of the body. The forged weight of the tube is approximately 88,000 lb. or less than half the weight of the ingot.

32 As a rough average for both armor and guns only about 25 to 30 per cent of the ingot weight is shipped away as product.

OPEN-HEARTH BUILDING

33 Open-hearth buildings have become in recent years more or less standard in general design and consist principally of a pouring aisle, a charging aisle, the floor of which is at a higher level, and usually a lean-to. An approach trestle leads to the charging floor up which the charge for the open-hearth furnaces is brought in charging boxes on cars from a remote stock yard or in the shape of hot metal directly from the blast furnaces. The unusual features of the open hearth at the Ordnance Plant consist of the arrangement of the stock yard alongside the building, the 100-ft.-wide pouring aisle with exceptionally heavy cranes, the arrangement for "duplexing" by placing the electric furnaces in the pouring aisle and the dump trestle and storage bins in the crane-served charging aisle.

34 Figs. 5 to 8, inclusive, show the general arrangement and give the important dimensions of the building which including the stock yard is 331 ft. wide by 516 ft. long (12 bays at 43 ft. each). The pouring aisle is 107 ft. wide, the charging aisle 86 ft. wide, the lean-to 32 ft. and the stock yard 106 ft. The height to crane rails in the pouring aisle is 55 ft. The crane rails of the charging aisle are at the same elevation but the charging floor is 17 ft. higher than the floor of the pouring aisle. The charging floor has been designed to withstand a live load of 800 lb. per sq. ft.

35 The charging floor of the open-hearth building is extended 30 ft. out under the crane runway of the stock yard to form a charging platform. Two parallel tracks are arranged upon this with cross-overs conveniently located and a turn-out leads to the charge track directly in front of the open-hearth furnace doors. One of the platform tracks continues to the east and is connected into the three-way approach trestle.

36 A track connected at both ends to the main system runs through the stock yard at the ground level adjacent to the charging platform. Materials for the charge, such as scrap, pig iron, etc., received on this track can be unloaded directly by the stock-yard cranes by magnet and placed in charging boxes on charging cars on the charging platform. The cars are arranged for four boxes of 5000 lb. capacity each. These cars will be made up into trains, each

containing a complete charge for an open-hearth furnace, or the incoming cars of material may be unloaded directly into the stock piles. The charge train will be shifted by an electric storage-battery locomotive over the scale, located near the entrance to the charging aisle, directly to the track in front of the open-hearth furnace doors and into position for the charging machine to start charging the furnaces. The empty train of charging boxes will be shunted back to the charging platform over the same track it entered, or out of the building at the east on to the trestle and back. In certain cases charging boxes will be filled with turnings in the machine shop and brought directly up the trestle to the open-hearth furnaces.

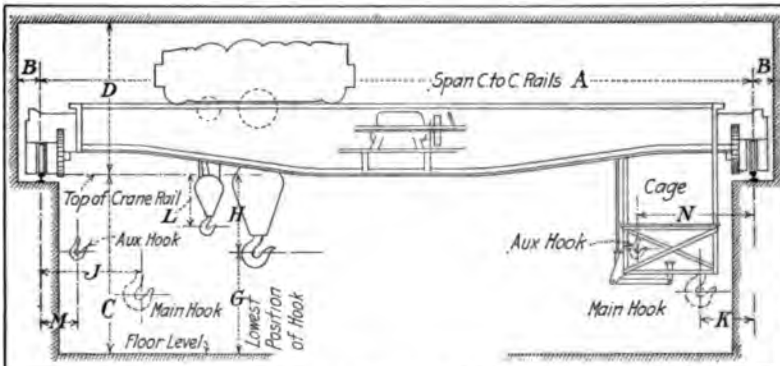
37 The charging floor carries three tracks, two of standard gage which are connected into the three-way approach trestle at the east of the building and the third of 24 ft. 6 in. gage for the low-type charging machine. The floor is paved with brick. The charging track is located directly in front of the furnaces and the so-called "hot-metal track" is at the far side of the aisle. It is not contemplated that blast furnaces will be built and the chance of ever using hot metal to charge the open-hearth furnace is remote. This track originates in the approach trestle and continues at the east end of the building over a series of concrete bins arranged so that cars of material can be bottom-dumped. Fluorspar, sand, iron ore, dolomite, limestone, etc., will be stored in these bins. The track then leads to the main charging floor where it will be used for handling materials with which to repair the furnaces and thus avoid interference with operation.

38 Space has been provided for three 65-ton open-hearth furnaces, two of which are now being completed. An electric traveling crane of 25 tons capacity with an auxiliary hoist of 10 tons capacity serves the entire length of the charging aisle.

39 Underneath the charging floor and back of the open-hearth furnaces is the pit for the checker chambers enclosed by a reinforced-concrete retaining wall. The lean-to contains the offices, the quick-test laboratory, chimneys, storage areas, lockers, toilets, etc.

40 The pouring aisle is served by three electric traveling cranes, specifications for which are given in Fig. 4. The distance from the center to center of crane runway rails is 100 ft. Crane No. 1 is a standard bridge crane 250–50 tons capacity, which will be used mainly as a pit crane in stripping and handling ingots. Crane No. 2 is a 4-girder ladle crane of 125 tons capacity with a 25-ton auxiliary trolley, and crane No. 3 is a 4-girder ladle crane

of 75 tons capacity with a 15-ton auxiliary trolley. Jib cranes of 6 tons capacity with a radius of 25 ft. are mounted on main building



	OPEN-HEARTH CRANES				FORGE AND FURNACE BUILDING						MACHINE SHOP		HEAT TREATMENT	
CRANE NUMBER	1	2	3	4	5,6	7,8	9,10	11, 12	13, 14	15	16,17	18,19 20,21	22	23,24
LOCATION	O.H. POURING SLIDE	O.H. POURING SLIDE	O.H. POURING SLIDE	O.H. CHARGING SLIDE	STOCK YARD	MAIN AISLE	MAIN AISLE	PRESS ROOM		PRESS ROOM				
CAPACITY (NET TONS)														
MAIN HOIST	250	125	75	25	15	200	100	250		75	150	75	75	NO 21
AUXILIARY HOIST	40	25	15	10	NONE	25	25	25		15	25	15	NONE	
SPEED (FT. PER MIN.)														NO 20
MAIN HOIST	10	10	14	23	50	8	10	7		14	10	14	HOIST 50 LOWER [illegible]	
AUXILIARY HOIST	20	23	30	30	0	23	23	30		30	23	30	0	NO 19
BRIDGE	200	200	200	250	300	200	200	150		200	200	200	100	
TROLLEY	75	75	100	150	150	75	100	50		70	100	100	50	NO 18
CRANE RAILS	175LB	LORRAINE STEEL CO		80LB ASCE		175LB LS Co		2-175LB LS 2-0" C TO C		175LB LORRAINE STEEL CO				
CLEARANCES														
A	100'-0"	100'-0"	100'-0"	80'-0"	100'-0"	100'-0"	100'-0"	70'-9"		70'-0"	100'-0"	100'-0"	104'-0"	SAME AS CRANES
B	1'-11¾"	1'-11¾"	1'-11¾"	1'-3¼"	1'-2"	1'-10½"	1'-10½"	2'-10"		2'-9"	1'-10½"	1'-10½"	1'-10½"	
C	55'-0"	55'-0"	55'-0"	54'-10¼"	39'-0"	40'-0"	40'-0"	46'-9"		78'-0"	40'-0"	40'-0"	165'-0"	
D	19'-0"	19'-0"	19'-0"	19'-1¾"	12'-0"	19'-0"	19'-0"	21'-0"		13'-6"	17'-6"	17'-6"	13'-6"	
G	53'-0"	40'-0"	43'-0"	53'-10½"	38'-0"	37'-0"	38'-0"	25'-0"		78'-0"	37'-0"	38'-0"	165'-0"	
H	2'-0"	15'-0"	12'-0"	1'-0"	1'-0"	3'-0"	2'-0"	21'-6"		0'-0"	3'-0"	2'-0"	0'-0"	
J	8'-6"	7'-0"	7'-0"	6'-0"	3'-6"	9'-3"	7'-6"	3'-0"		7'-0"	8'-6"	7'-0"	10'-0"	
K	9'-0"	8'-0"	7'-0"	4'-0"	3'-6"	10'-0"	5'-0"	5'-0"		6'-0"	7'-9"	6'-0"	10'-0"	
L	0'-0"	0'-0"	0'-0"	0'-0"		0'-0"	0'-0"	26'-6"		0'-0"	0'-0"	0'-0"		
M	3'-6"	7'-0"	3'-6"	3'-6"		3'-6"	3'-6"	7'-0"		3'-6"	3'-6"	3'-6"		
N	14'-0"	6'-0"	6'-6"	6'-6"		15'-9"	9'-0"	7'-0"		9'-6"	12'-9"	9'-6"		
REMARKS		FOUR GIRDER LADLE CRANES			CAGE AT CENTER OF SPAN			HYDRAULIC HOIST TUMBLING BLOCK SPEED 6FT PER MIN		VERTICAL LIFT			REMOTE CONTROL	

FIG. 4 SPECIFICATIONS FOR ELECTRIC TRAVELING CRANES

columns adjacent to the open-hearth furnaces for handling the spouts.

41 The unusual width of the pouring aisle was to give space

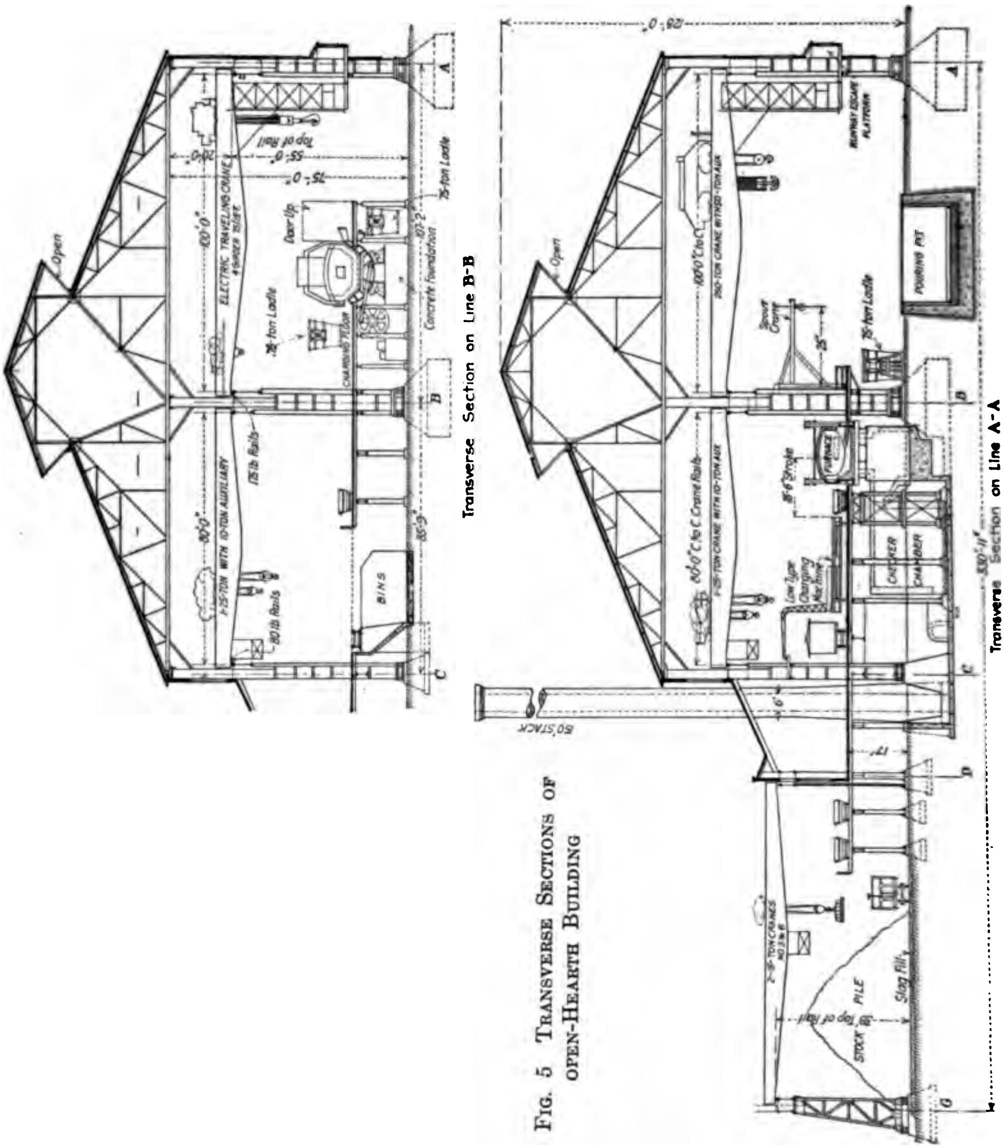


FIG. 5 TRANSVERSE SECTIONS OF OPEN-HEARTH BUILDING

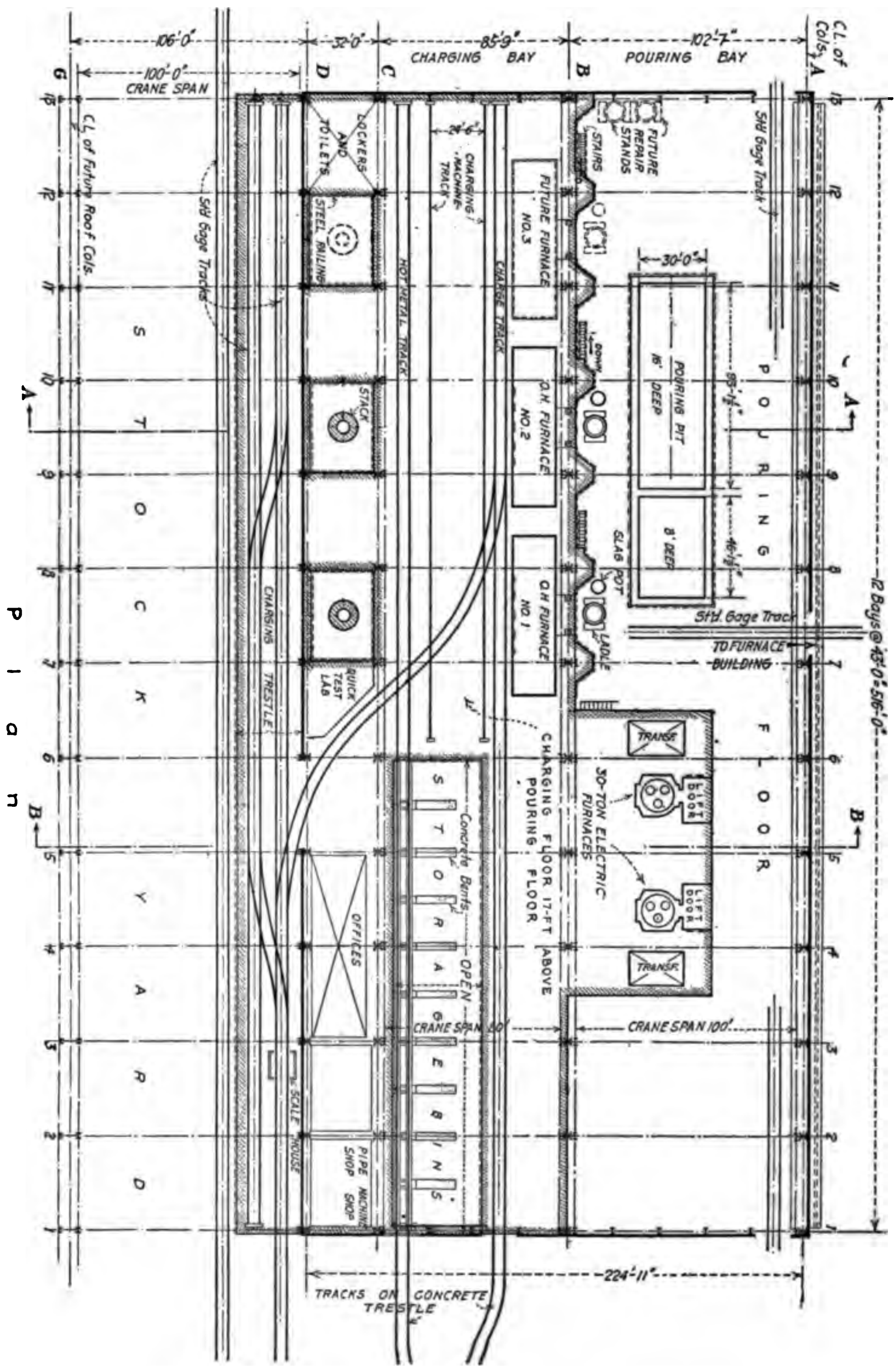


Fig. 6. Floor Plan of Open-Hearth Building

in the open-hearth building for the storage and chipping of ingots, and on account of the large pouring pit in which it is planned to ash-anneal the ingots. The location of the electric furnaces for duplexing in this aisle was an important factor in determining this width.

42 The pouring pit is divided for convenience in operation into two levels and is of reinforced concrete lined with 13 in. of

Fig. 7 Pouring Aisle of Open-Hearth Building Looking East from Crane Girder, July 14, 1920

Casting pit is in immediate foreground; electric furnaces just beyond. Charging floor, when completed, will extend completely around furnaces. Locations for open-hearth furnaces Nos. 1 and 2 are shown to the right just behind middle row of column. Box cars on charge track of charging aisle indicate scale. Crane span of charging aisle is 100 ft. wide.

red brick. The lower level is 16 ft. below the floor, 30 ft. wide and 93 ft. long and the upper level is 8 ft. deep, 30 ft. wide and 46 ft. long.

43 For open-hearth steel the furnaces will be tapped into the ladles and carried directly to the pouring pit and bottom-poured directly into the ingot molds. In obtaining electric steel by duplexing, the open-hearth furnaces will be used simply for melting the charge which will be tapped into ladles and then carried down the shop to charge the electric furnaces.

FIG. 8 OPEN-HEARTH BUILDING FROM SOUTHEAST, AUGUST 2, 1920

Alongside the open-hearth building is the stockyard with 100-ft. crane span. Classification yard is to extreme left. Note track doors at charging-floor level. Three-way concrete trestle will replace temporary wooden trestle. In immediate foreground is general storehouse and blacksmith and pattern shop (No. 5), twin buildings, each 130 ft. by 300 ft., received through Army salvage.

44 Two 30-ton Heroult electric furnaces have been erected as shown in Fig. 7, on independent concrete foundations, and are surrounded by a charging platform which is an extension of the charging floor and on which are placed the transformer houses. Flux material will be brought over to this floor from the bins in the charging aisle. The furnaces will be charged from the rear. The capacity of one open hearth is sufficient to charge the two large electric furnaces. After the refining process, a ladle will be brought close to the front of the electric furnace by lifting a large counterbalanced and motor-operated trap door and the furnace tilted forward and the contents discharged into the ladle, which is then carried back to the pouring pit and the molten steel is bottom-poured into ingots in the usual manner. It is not contemplated that the 30-ton electric furnaces will be charged otherwise than with hot metal. It is planned to move over two of the three 6-ton electric furnaces now located in the projectile plant in order to consolidate the steel ingot making of the entire plant.

45 In casting the largest ingot, which will weigh approximately 400,000 lb. and will be done half a dozen times in the course of a year, the open-hearth furnaces will be overloaded to 80 tons each and will be brought out at the same time as one or both of the 30-ton electric furnaces overcharged if required.

46 Tracks enter the pouring aisle at either end for a length of one or two car lengths and the south end of the shuttle track is at the middle of the building adjacent to the pouring pit, conveniently located for the transportation of ingots to the forge shop.

47 Attention is invited to the runway escape platform for the crane operators at the level of the bottom of the crane cages, also to the provision in the structural design for ventilation. The entire wall area of the building is left open to a distance 8 ft. above the ground, although arrangements are made for vertically lifting doors in case of severe weather. The sides of the large monitor are left entirely open and without louvers, high curbs being provided on the main roof and the overhang of the monitor roof being made excessive for protection against the weather.

FORGE AND FURNACE BUILDING

48 The design of the forge and furnace building of the U. S. Naval Ordnance Plant is believed to be a radical departure from anything that has been built for the purpose.

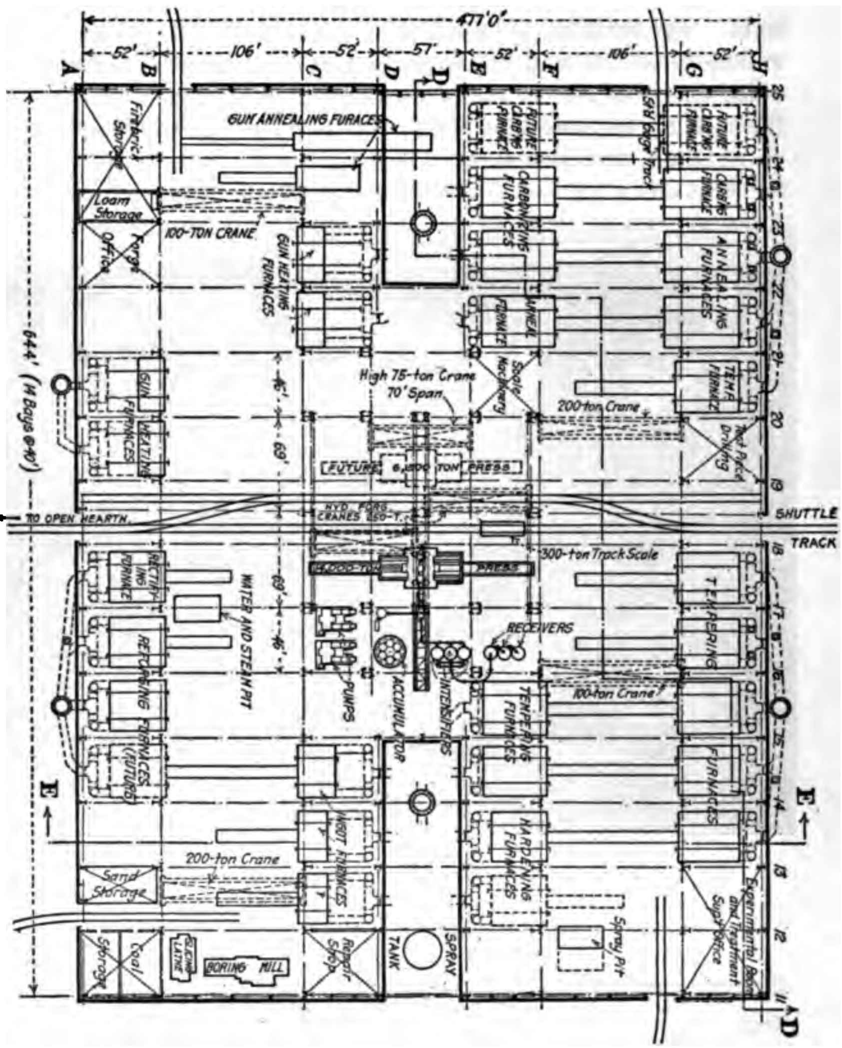


FIG. 9 FLOOR PLAN OF FORGE AND FURNACE BUILDING

49 The forge shops of the existing armor plants consist of buildings with one main crane aisle and a lean-to on one or both sides. The furnaces are located in the lean-tos, the presses in the middle or at one end of the main aisle. Forging cranes are provided on either side of the press on crane rails which at one end of the span are supported on the top of the press and at the other end on separate steel structures. Forging heats are brought from the furnaces down the main aisle by the overhead cranes and placed in a position where they can be picked up by the forging cranes, al-

FIG. 10 FORGE AND FURNACE BUILDING, SOUTH LEAN-TO, LOOKING NORTHWEST, JULY 16, 1920

Foundations for reforging furnaces with glimpse of south aisle in background.

though in certain cases the ingot furnaces are in direct reach of the forging cranes. In order that the presses and the forging cranes may be cleared, the overhead crane in the main aisle must be placed at a relatively great height above the ground. This makes an expensive building. Had the forge shop of the Naval Ordnance Plant been built in accordance with the arrangements of the existing plants, and been consolidated with the treatment plant as has been done and the same number of furnaces installed, a building approximately 2000 ft. long with a height to crane rails of over 80 ft. above the ground throughout the length of the building would have been necessary.

50 The treatment shops of the existing armor plants are located in separate structures from the forge shops, and as plates must be returned to the press for bending, straightening, etc., considerable railroad traffic between the two buildings results, unless two very large presses are provided.

51 At the Naval Ordnance Plant the forge shop and the

FIG. 11 FORGE AND FURNACE BUILDING, SOUTH AISLE, LOOKING NORTHEAST FROM CRANE GIRDER, JULY 12, 1920

Deeper crane girders in left foreground and heavy columns are at 70-ft. spans where cross-tracks will lead to press room on left.

treatment shop for both armor plate and gun forgings have been consolidated in the "H"-type forge and furnace building. The general arrangement is shown by Figs. 9, 10, 11, and 12. The over-all dimensions of the building are 644 ft. (14 bays at 46 ft.) by 477 ft. The building consists of two main aisles, each 106 ft. wide and each having lean-tos 52 ft. wide on either side. The crane span of the north and south main aisles is 100 ft. from center to center of rails and rails are 40 ft. above the floor level. Each aisle is served with a

200-ton and a 100-ton crane. The main aisles are connected by the press room which forms the cross-bar of the "H." The shuttle track goes directly through the center of the building and is double-tracked with cross-overs as indicated throughout the width of the building.

FIG. 12 PRESS ROOM, FORGE AND FURNACE BUILDING, LOOKING EAST FROM CRANE GIRDER, JULY 16, 1920

Foundations for 14,000-ton press in middle foreground. In immediate foreground is excavation for double column which will support inside crane girders running directly across the top of the press. Note high crane for erection and repair of press and forging crane. Note double rails on outside girders for forging cranes. Intensifiers will be located at position of locomotive crane and a 32-in. accumulator where sand pile is shown.

52 The furnaces, mostly of the Carbottom type, are placed in the lean-tos on either side of the two main aisles and each occupies one bay in length (Fig. 13). The south aisle contains the ingot and reforging furnaces for the armor plate, and the heating and annealing

furnaces for the gun forgings. It also contains the ingot-slicing lathe and a double-end boring mill as well as a water and steam pit of reinforced concrete for the rectifying of armor. Area in the lean-tos

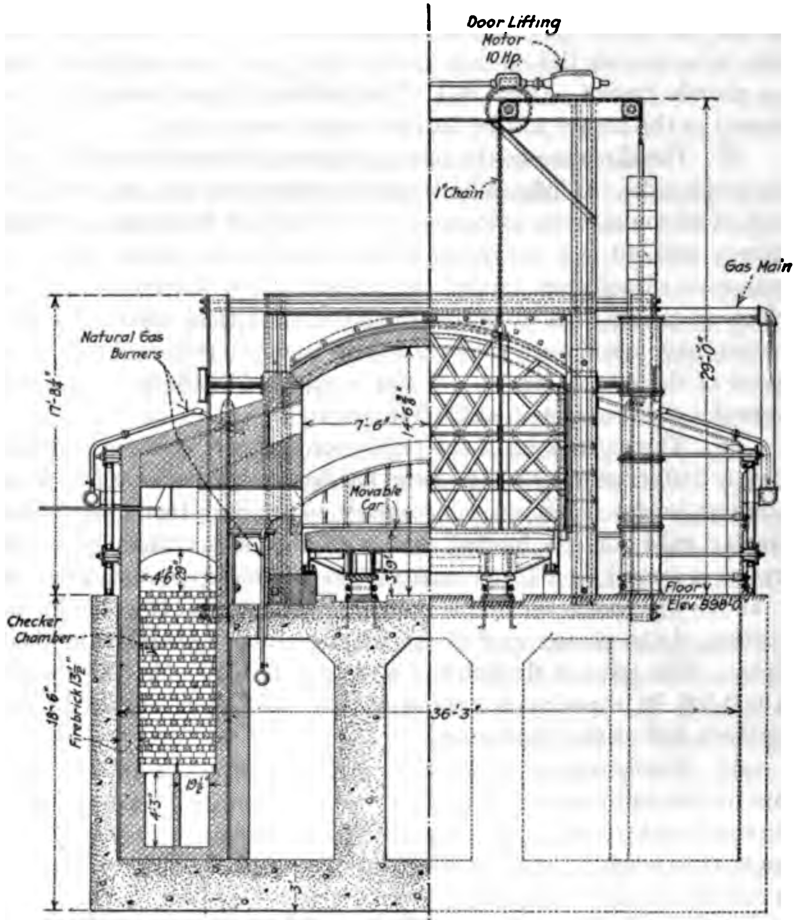


FIG. 13 SECTION AND END VIEW OF CARBOTTOM FURNACE FOR ARMOR PLATE

not occupied by furnaces is given over to storage for firebrick, sand, loam, etc., and the forge superintendent's office.

53 The north aisle contains the tempering, annealing, carbonizing and hardening furnaces for armor plate. It also contains the spray pit for the quenching of armor and in lean-to space un-

occupied by furnaces, the experimental room, treatment superintendent's office and machinery for scaling and test-piece drilling.

54 The 14,000-ton steam-intensified hydraulic forging press and its driving equipment including the 2500-lb. hydraulic-pressure pumps, the 32-in. accumulator and the steam receivers and intensifiers, is located in the eastern half of the press room as divided by the shuttle tracks. (Fig. 14.) The 6500-ton press which will be erected in the future will be located in the western half.

55 The direction of the forging is from the south aisle towards the north aisle. A 250-ton hydraulic forging crane with an auxiliary hook of 25-ton capacity is located on either side of the press on double runway rails 46 feet above the floor. The inside runway rails are supported directly on top of the press. These forging cranes, of which all the motions excepting the hoisting motions are electrically driven, have a sufficient range of bridge travel to pick up loads from either of the shuttle tracks. Water is supplied to the hoist motions from the pressure system at 2500 lb. per sq. in.

56 The entire area of the press room proper, which is approximately 140 ft. wide by 160 ft. long, has been made entirely free from columns except for the row necessary to support the inside crane runway rails for the forging cranes. The central area over the presses is served by a 75-ton crane located on rails approximately 80 ft. above the floor level. This crane has been placed to serve in the erection of the presses and of the forging cranes as well as in their repair. The span of the runway girders is 140 ft. The track scale of 600,000 lb. capacity is located on the main shuttle track in the northern half of the press room.

57 Briefly and generally the process for armor-plate manufacture in this building will be as follows. The ingot will be received in the south aisle from the open-hearth building on a flat car by way of the shuttle track. It will be lifted by an overhead crane and placed in one of the ingot furnaces by means of a porter bar. To heat the ingot will require up to 20 hr. The hot ingot will then be withdrawn from the furnace and carried down the aisle to either one of the shuttle tracks and deposited on a car which will then be pushed through the few feet necessary by an electric locomotive into the press room. The ingot will be picked up by a forging crane and carried under the press.

58 The first forging operation will last about an hour. It will bring the plate down within an inch or so of its finished thickness and will include cropping the ingot. The plate, which is then 35

per cent lighter, is returned to the south aisle, deposited on the car of one of the Carbottom type reforging furnaces, which will then be rolled into the furnace and the plate will be reheated preparatory to reforging. After reforging and rectifying, the plate will come out into the north aisle for the carbonizing, annealing, tempering and hardening operations, and under normal conditions will not return again to the south aisle although returning occasionally to the press room, as required, for bending and straightening. It then goes via shuttle track to the machine shop.

59 The "H"-type of building consolidating the forging plant with the treatment building presents the following main advantages:

(*a*) A low first cost. An estimated saving of half a million dollars was secured by keeping the height of the run-way rails in the main aisles down to 40 ft. rather than going to the 80 ft. that would have been necessary in the more usual type of forge shop.

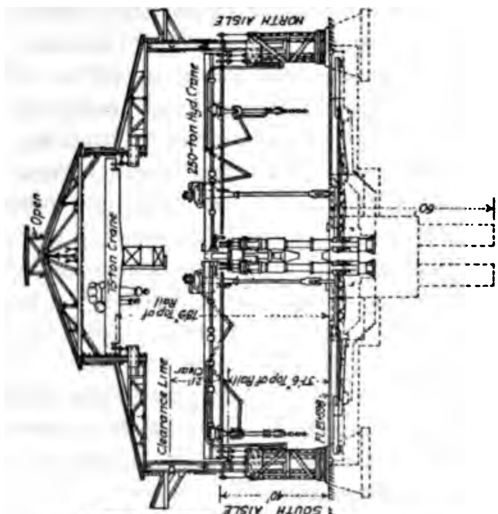
(*b*) The maximum economy of space in the layout of the entire plant is secured by consolidating the forge and treatment operations in one building.

(*c*) The maximum economy in operation should result due to the compactness of the building itself, the relatively short haul from any furnace to the press room, and the time saved over present methods in operation.

(*d*) The presses are very accessible to the furnaces and any press can receive work directly from any furnace in either aisle. There should be none of the expensive "waiting for heats" so common in many forge shops due to the fact that the press is available to a limited number of furnaces only.

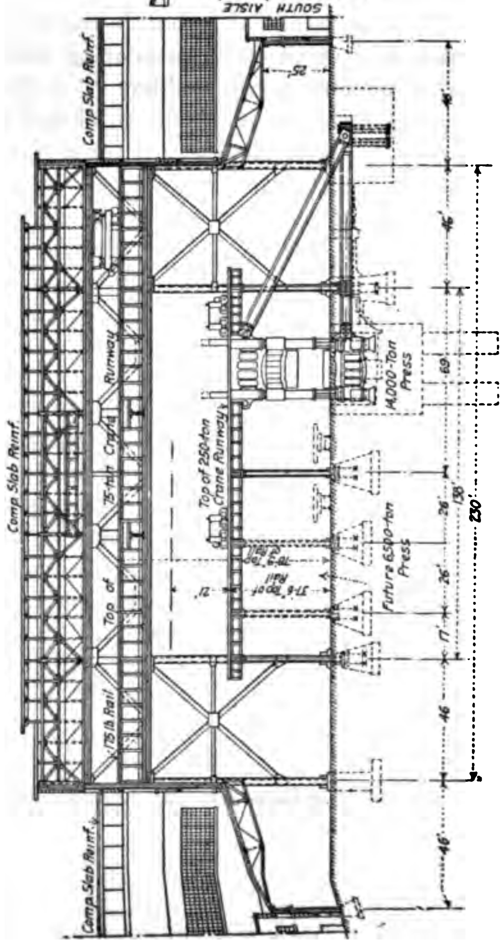
(*e*) Operation of cranes in the furnace aisles will not interfere with or be interfered with by forging operations in the press room.

(*f*) Operating conditions are ideal in so far as light and ventilation are concerned. The sash area in the walls on either side of the press room is unusually large and affords direct daylight to the entire area. The temperature in the press room will always be moderate as the press is not confined between ingot-heating furnaces in full blast. The monitor on the top of the press room is open and the walls surrounding the entire building are open to a height of 8 ft. above the ground, which should



Section on Line C-C

Cross Section Showing Hydraulic Cranes



Section on Line A-A

Longitudinal Section Showing Press and Crane Runways

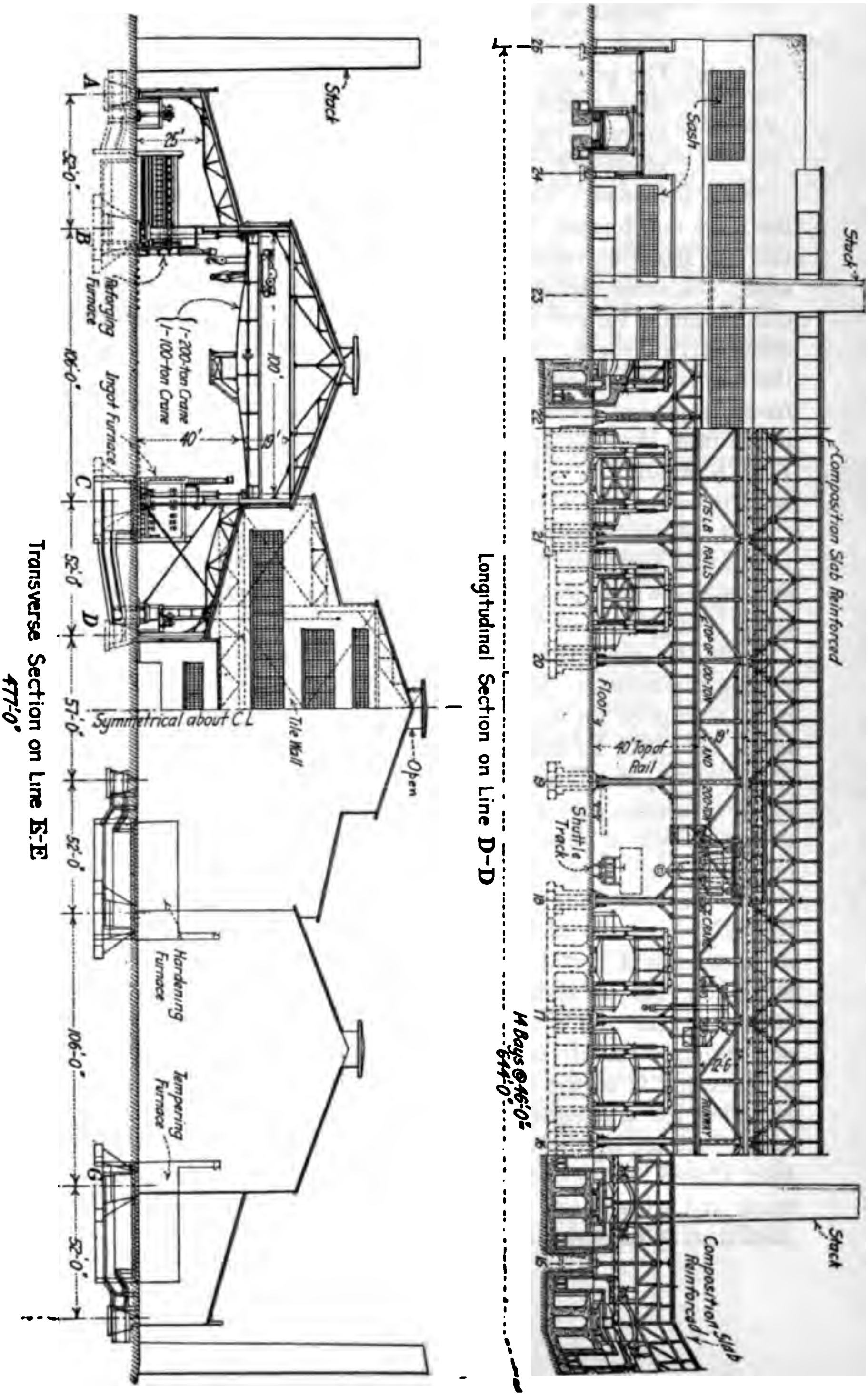


FIG. 14. SECTIONS OF FORGE AND FURNACE BUILDING

permit a continuous draft of fresh air in the press room.

(*g*) The presses and press-drive equipment are concentrated in a central area, thereby avoiding any expensive distribution systems for steam, high-pressure water or electricity.

60 *Furnaces.* Twenty-five furnaces are being completed in the forge and furnace building. Of these the three ingot furnaces the two reforging furnaces and the rectifying furnace in the south aisle; the three carbonizing, three annealing, eight tempering and one hardening furnace in the north aisle are identical in section and differ only in length. They are of the regenerative Carbottom type, the fuel is natural gas, and the air preheated. There are three different lengths of furnace, 50, 42 and 36 ft., respectively. The height inside from the floor line to the top of the underside of the arch is 14 ft. 6 in. and the width inside from face to face of the brickwork 15 ft. in each case. The general arrangement is shown in Fig. 13.

61 A movable car, designed to carry a maximum load of 700 tons, forms the bottom of the furnace. The cars move on independent rollers which roll in specially constructed steel tracks. The track is supported on reinforced-concrete foundations.

62 An interesting feature of the furnace equipment is the car-pulling mechanism. The usual method of moving the Carbottoms is by means of an hydraulic cylinder and plunger located underneath the car or by individual electric motor-driven hoists of about 75 to 100 hp., each located under the floor in front of the furnaces and moving them in and out by means of a wire rope wound around drums. Both of these methods are expensive in installation and operation.

63 Having in mind the "electric mules" which draw the ships through the locks at Panama, the idea was conceived of using an independent portable electric-driven rack-rail locomotive which would operate on standard-gage tracks placed between the Carbottom tracks and which would draw the Carbottoms by means of a standard M.C.B. coupling. A spring reel of cable was arranged for plugging in to the nearest electric-power socket and rings were provided for conveniently lifting and carrying the mule to any furnace desired by the overhead crane. The device has subsequently been altered by the operating division to exclude the standard-gage track and wheels, and as built the car-pulling mechanism will be carried on an extension of the girders of the Carbottoms.

64 The green-annealing furnaces for gun forgings, one of which will be 100 ft. long, are also of the Carbottom type. The balance of the furnaces are of the solid-bottom type.

65 Six radial brick chimneys with independent firebrick linings are being erected to serve the furnaces by groups. The chimneys on the north and south elevation of the building are 6 ft. 6 in. in diameter. The chimneys in the interior courts are 8 ft. in diameter. All are 150 ft. high.

66 *Forging Press.* The forging press is of 14,000 gross tons capacity and is of the steam-intensified hydraulic type. The maximum working stroke is 90 in. and the four press columns are 19 ft. by 8 ft. 6 in. on centers. The columns are each 30 in. in diameter. There are three cylinders each 44 in. in diameter and the working pressure is 7000 lb. per sq. in.

67 There are two double-acting manipulating jacks in pits on either side of the press, each of 150 tons capacity and having a maximum stroke of approximately 18 ft.

68 Valves are arranged so that one forging cylinder alone may be used, giving greater speed for forging gun work under these conditions.

69 Water is supplied at 2500 lb. pressure by three 250-gal-per-min. pumps. A 32-in. accumulator operates on this system which also supplies the manipulating cylinders, pull-back plungers and the 250-ton hydraulic forging cranes. The triple intensifier operating on steam at 200 lb. pressure from the boiler plant, increases the pressure in the press cylinders to 7000 lb. per sq. in. The general arrangement is shown in Figs. 12 and 14.

70 As a matter of general interest, the total weight of the press alone is nearly 5,000,000 lb. The total load at the bottom of the foundation, assuming both cranes loaded directly over the press, and including the dead weight of the press and of the concrete foundations, is approximately 13,000,000 lb. To support this load it was necessary to place the press on four concrete piers, each 9 ft. in diameter, which bear on rock about 60 ft. below the level of the press-room floor.

71 Under conditions of eccentric loading there is an extremely remote possibility that a very great side thrust may occur at the top of the press. At the insistence of the manufacturer, a structural steel brace was designed to take this load, as is shown in the section of the press room (Fig. 14).

72 All motions of the presses and jack are controlled from a

pulpit located on the floor of the press room on line with the press and about 15 ft. away. The hoist motions of the 250-ton hydraulic cranes are also controlled at this point. The bridge and trolley motions of the crane are controlled from cages mounted at the far end of each forging crane whence the operator can most readily line up the piece in the press.

73 The spray for quenching armor plate is placed at the east end of the north aisle. This consists of a reinforced-concrete pit

FIG. 15 MACHINE SHOP, MIDDLE AISLE, LOOKING EAST, JULY 20, 1920

Note lighting. Immediate foreground is location of shuttle track. Crane span 100 ft.; girders support one 150-ton and one 75-ton crane. Columns are 46 ft. on centers.

16 by 32 ft. by 8 ft. deep. The bottom is covered by a grillage of small pipes perforated with $\frac{1}{4}$-in. holes. A carriage on tracks carrying a like grillage straddles the pit. The plate is taken from the hardening Carbottom furnace nearby by a crane and placed on supports in the pit. The carriage is then moved over it and the plate drenched on either or both sides with water. Approximately 25,000 gal. of water per min. are required during the first twenty minutes of the operations. The water from the spray is wasted into the adjacent gully.

MACHINE SHOP

74 The general design of the machine shop is shown by Figs. 15 and 16. The building is 320 ft. wide by 552 ft. long. It comprises three main aisles, each of which is 100 ft. wide center to center of crane rails. Crane rails are 40 ft. above the floor level. Columns are spaced 46 ft. on centers.

75 The south aisle and the middle aisle are to be used for the finishing of the armor plate. Each is served with a 150-ton and a

FIG. 16 MACHINE SHOP, NORTH AISLE, LOOKING SOUTHEAST FROM FLOOR LEVEL, JULY 13, 1920

Boom of locomotive crane is 70 ft. long.

75-ton crane with 25-ton and 10-ton auxiliary hoists, respectively. The principal machine tools in the south aisle are an 84-in. by 84-in. planer with a 24-ft. platen, three 15-ft. by 10-ft. planers, two universal borers and drillers of the car type, two universal boring, drilling and milling machines, a universal radial drill and two armor-plate grinders. At the west end of the south aisle will be located the burning equipment; gas will be piped from a separate oxy-hydrogen plant which has been built 100 ft. south as a matter of safety. At the west end of the south aisle is the erection floor and the surface plates for the fitting up of armor plate prior to shipment. The erection floor will be 168 ft. long by 93 ft. wide and will consist of a thick reinforced-concrete slab with rails embedded in the surface. The surface plates cover an area 85 ft. by 48 ft. and are of steel. Special arrangements are made in the foundations so that an absolutely level surface may be accurately maintained.

76 The middle aisle contains an 18-in. by 48-in. by 15-ft. open-side planer, a vertical and horizontal planer, two 120-in. armor-plate rotary planers and saws, a cutting off machine and a double armor-plate breast planer. Offices and a tool room will be located in the farthest bay west.

77 The north aisle will be devoted to the machining of gun forgings. The north aisle is served by two 75-ton cranes with 15-ton auxiliary hoists. Equipment of ample size to rough-bore and turn 20-in. 50-caliber gun forgings is provided. The principal items are as follows: an 80-in. by 40-ft. hoop borer, a double-end borer which will take a piece nearly 100 ft. long and which requires a foundation 270 ft. long; two 120-in. boring and turning machines and an 85-in. turning lathe; a 6-ft. radial drill, a 10-ft. boring mill, 2 horizontal boring. drilling and milling machines, and two large cold saws.

78 The shuttle track ends in the middle aisle of the machine shop. Gun forgings from the forge shop will arrive by way of the main track system in order not to make necessary swinging the great length of the forgings through 90 deg., although this can be very conveniently done if required, due to the 100 ft. width of the aisle. Railroad tracks enter both ends of the north and middle aisles and the east end of the south aisle. A track scale of 100-ton capacity will be placed on the latter track.

79 The elevation of the machine shop at 595 is 3 ft. lower than any of the other buildings, all of which have been maintained at an elevation of 598. This was necessary on account of the topography of the site. The whole area of the machine shop was filled and rolled to an average depth of about 4 ft. Arrangements have been made for the ultimate installation of a reinforced-concrete floor surfaced with wood block. This will be done in the course of a year or so, when the fill has been worked on and had a chance to settle.

80 Special attention is invited to the method of securing an abundance of daylight. Somewhat over 60 per cent of the side and end wall area is in steel sash. Top lighting is provided by longitudinal sawtooth monitors in the side aisles and a double sawtooth monitor

in the middle aisle. The construction is shown in Fig. 17 and the result secured by the photograph in Fig. 15. The steel design is clean cut and simple, although unusually heavy columns and crane girders are necessary to support the crane loads. The roof trusses in all three aisles are duplicates and the building has been designed and located so that it may be extended as required for several bays in either direction.

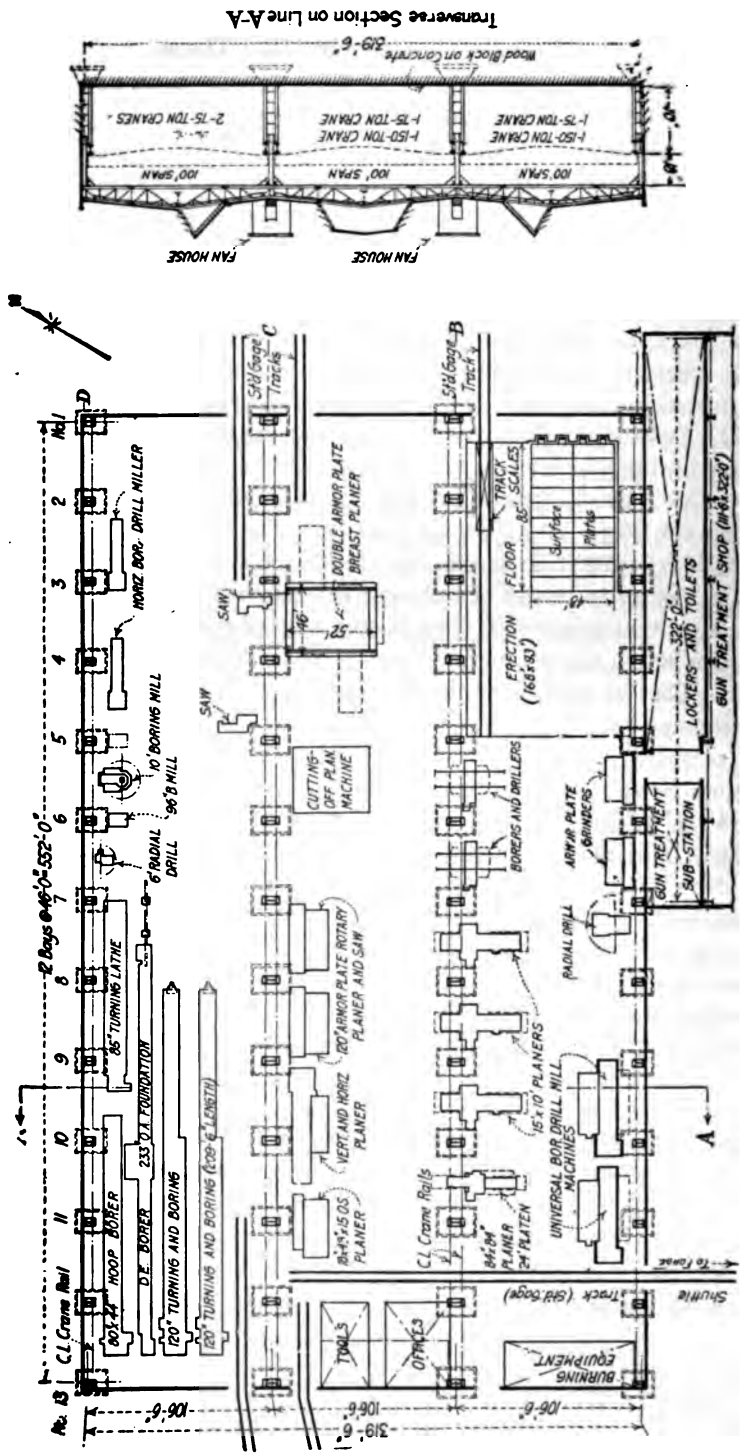
GUN-TREATMENT BUILDING

81 This building is located alongside the machine shop, the intervening space 40 ft. wide being roofed over to provide for lockers, toilets, offices and a two-story electric substation for the gun-treatment furnaces. Figs. 18 and 19 show the general arrangement.

82 The building consists of a low portion and a high portion. The low portion has a single crane span 100 ft. wide center to center of rails and is 230 ft. long. The crane rails are 40 ft. above the floor and run out 46 ft. into the high portion. The high portion is two bays of 46 ft. or 92 ft. in length and 116 ft. in width. The crane span is 104 ft. center to center of rails and the rails are 165 ft. above the floor. A railroad connection from the main track system runs through the entire length of the building close to the south wall.

83 The low portion of the building provides space for a future straightening press, storage area for forgings, while awaiting treatment or results of tests, cutting-off saws and slotting and boring machines for the taking of test specimens.

84 The high portion provides for the vertical treatment of gun forgings. A reinforced-concrete pit approximately 80 ft. square and 55 ft. deep has been built and will contain the 10-ft. by 105-ft. vertical electric heat-treatment furnace and a quenching tank 10 ft. in diameter and 105 ft. high. A 50-ft. furnace is also to be erected at the present time and is located on a line with the high furnace and the quenching tank. Space is provided in the pit and on the floor for a duplication of this equipment. Structural-steel platforms will be erected around the furnaces in galleries 10 ft. apart for the convenient inspection of the forging in the furnace during operation through peep holes in the side of the furnace. The steel structure will also support the rolling doors which cover the top of the furnaces and the bridges from which the forgings are hung while in the furnaces. An interesting detail is the installation of an elevator to carry the operator from the bottom of the pit to any one of the ten galleries in the 105 ft. height of the treatment furnace structure.



Plan

Fig. 17 Machine Shop, General Layout and Transverse Section

85 The reinforced-concrete pit is of unusually heavy construction in order to withstand the pressure due to its depth, and due to the additional 20 ft. head of water to which the bottom is subjected. The design is unique in that no interior cross-bracing is used. Each of the four sides of the lower half of the pit is designed as a slab supported between the bottom of the pit and a flat horizontal beam at approximately one-half the total depth. The upper halves of the walls are designed as retaining walls, for which the flat beam forms

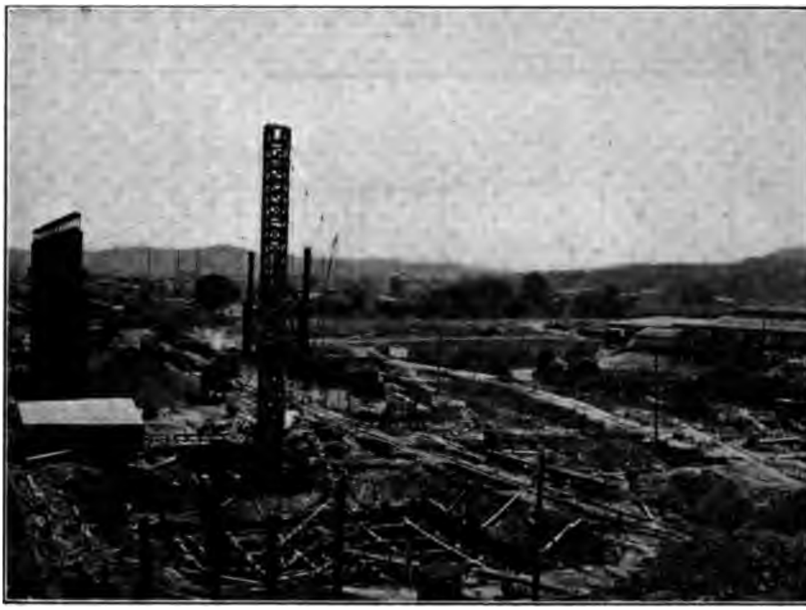


FIG. 18 GUN-TREATMENT BUILDING, GUN PIT IN FOREGROUND, JULY 12, 1920

Note concreting plant to pour 5000 yd. of concrete involved in pit. Sheet piles are 40 ft. long.

the base. Special precautions were taken in the construction of the pit to secure watertightness and the entire concrete work was divided into units each consisting of one full day's pouring of concrete, metal water stops being provided at each construction joint and also where the needle beams of the shoring pass through the wall.

86 The high portion is served by a 75-ton crane with a 104-ft. span which has a 50-ft.-per-min. hoisting speed at full load, and a 100-ft.-per-min. lowering speed. These severe requirements were necessary in order to get the gun forging from the vertical furnace

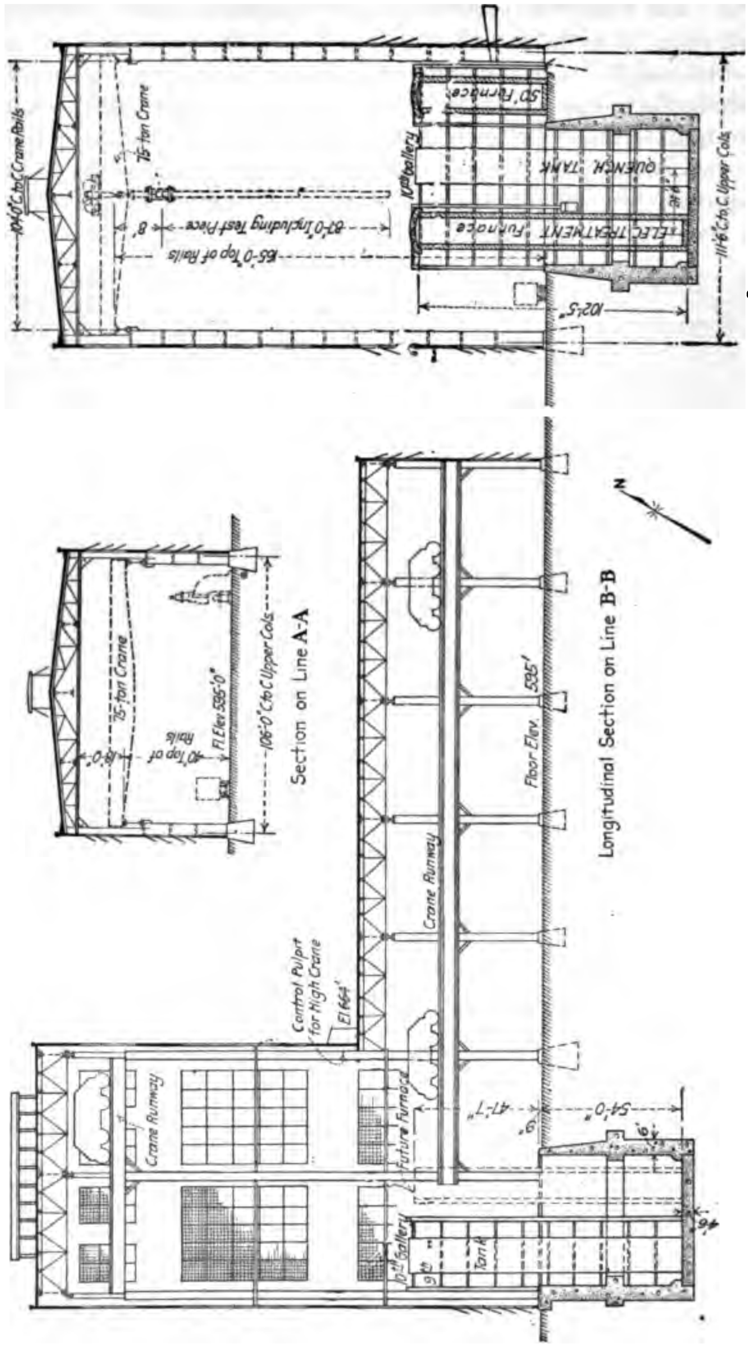


FIG. 19-A SECTIONS OF GUN-TREATMENT BUILDING (SEE ALSO FIG. 19-B)

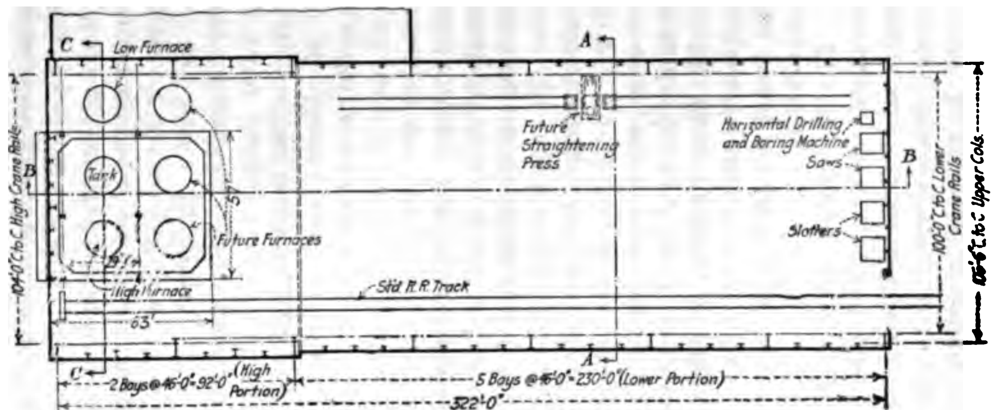


FIG. 19-B FLOOR PLAN OF GUN-TREATMENT BUILDING

into the quenching tank in the shortest possible time. Both upper and outer limit stops are provided. All motions of this crane are controlled from a pulpit located on line with the quenching tank and at a height just above the top of the furnaces and tanks, on the east wall of the high portion of the building. The furnaces, it will be noted, are placed on line with the tank so that only a hoist and a trolley motion will be required in carrying out the quenching operation.

87 The low portion is served with two cranes which are duplicates of the machine-shop 75-ton cranes. The runway of the low portion has been extended into the high portion so that pieces may be picked up and carried directly under the area served by the high crane and thus avoid an additional transfer by rail.

PLANT SERVICES

88 The service features of the plant including the industrial water supply, the sewerage, electrical power and lighting, steam, air, natural gas, etc., cannot be discussed in detail in a paper of this length but an attempt will be made to give a general perspective of the systems as planned and built. The electric, water, and compressed-air systems are centered in the service building, which consists of two component parts, an outdoor substation and the service building proper.

89 *Service Equipment.* The building houses four 1500-kw. rotary converters, synchronous condensers, three air compressors, six water pumps, various step-down transformers, necessary auxiliaries and the switchboards for controlling this equipment.

90 Electric power will enter the building at 6600 volts (3 phase, 60 cycles) and be distributed at that voltage from the main buses to all parts of the Naval Ordnance Plant. Part of the power will be stepped down to 2300 volts, 3-phase, for the air compressors and pump motors. Part will be transformed in constant-current transformers for series lighting and part will be stepped down to 115–230 volts single-phase for lighting and 230 volts 3-phase for small motors. Three 667-kilovolt-ampere 60-cycle single-phase 6600 to 2300-volt transformers are also provided.

91 Two of the air compressors have a capacity of 3000 cu. ft. of free air per min., one a capacity of 2000 cu. ft. of free air per min. Air is distributed at 100 lb. pressure.

92 Three 10,000-gal.-per-min. centrifugal motor-driven water pumps are provided for the spray system and three 3000-gal.-per-min. water pumps for the service system, one of which will be steam-driven.

93 The outdoor electric substation is for the control of the two 44,000-volt and one 66,000-volt three-phase 60-cycle incoming circuits and consists of 3 banks of transformers and the necessary switches, buses, grounds, and supporting framework.

94 In general the *electrical distribution system* may be outlined as follows: For the distribution of alternating currents, three centers in addition to the main substation and the projectile plant substation are provided. One of these is located adjacent to the gun treatment building in the west end of the lean-to between the gun treatment building and the machine shop, and controls gun-treatment furnaces, lights, and 220-volt a.c. power in both buildings,

and in addition, the direct-current feeders to the gun-treatment building. The second center is adjacent to the press room of the forge and furnace building and controls the hydraulic pumping equipment and the lighting for the building. The third center is under the charging platform near the electric furnaces in the open-hearth building and controls the electric furnaces and the lighting in that building.

95 Secondary distribution of alternating-current power in small quantities at 220 volts, 3-phase, 60 cycles, is arranged from the transformer substation at the gun-treatment shop and the main substation.

96 The lighting system in the various buildings is at 220–110 volts, three-wire single-phase.

97 Special voltages for special requirements and not for general distribution have been provided for the pumps and compressors in the main substation, the gun-treatment furnaces and the electric furnaces in the open-hearth.

98 Direct current will be distributed at 230 volts. In the machine shop there will be six distributing centers each covering machine tools in the adjacent area. Distributing centers for direct current are also located near to the a.c. switchboards in the gun-treatment building, forging furnace and open-hearth.

99 It is estimated that the kilowatts maximum demand totals approximately 21,000 kw., of which 10,500 is required for the electric melting furnaces, 2850 for the electric gun-treatment furnaces, 1020 for the 2500-lb. high-pressure hydraulic systems, 1250 for the spray pumps, 1800 for crane service, 700 for lighting and the balance for operating the machine shop tools, the furnace doors, the service pumps, air compressors, etc. A 70 per cent diversity factor has been assumed.

100 *Water Supply.* The industrial water supply is taken from the Great Kanawha River which flows west on the north boundary of the reservation. The pool stage of the river is at elevation 550. The river pump station is constructed of reinforced concrete directly in the river bank and is shown in outline by Fig. 20. Two 2000 gal.-per-min. pumps are being installed and space is provided for a third pump. The pump floor is underneath the river level and a positive head thereby assured. Occasional floods bring the river up 30 ft. or in extreme conditions, 35 ft. The pumps raise the water approximately 50 ft. to a surge tank whence it flows in a 30-in. tile pipe by gravity to the lower end of the reservoir settling basin. This basin

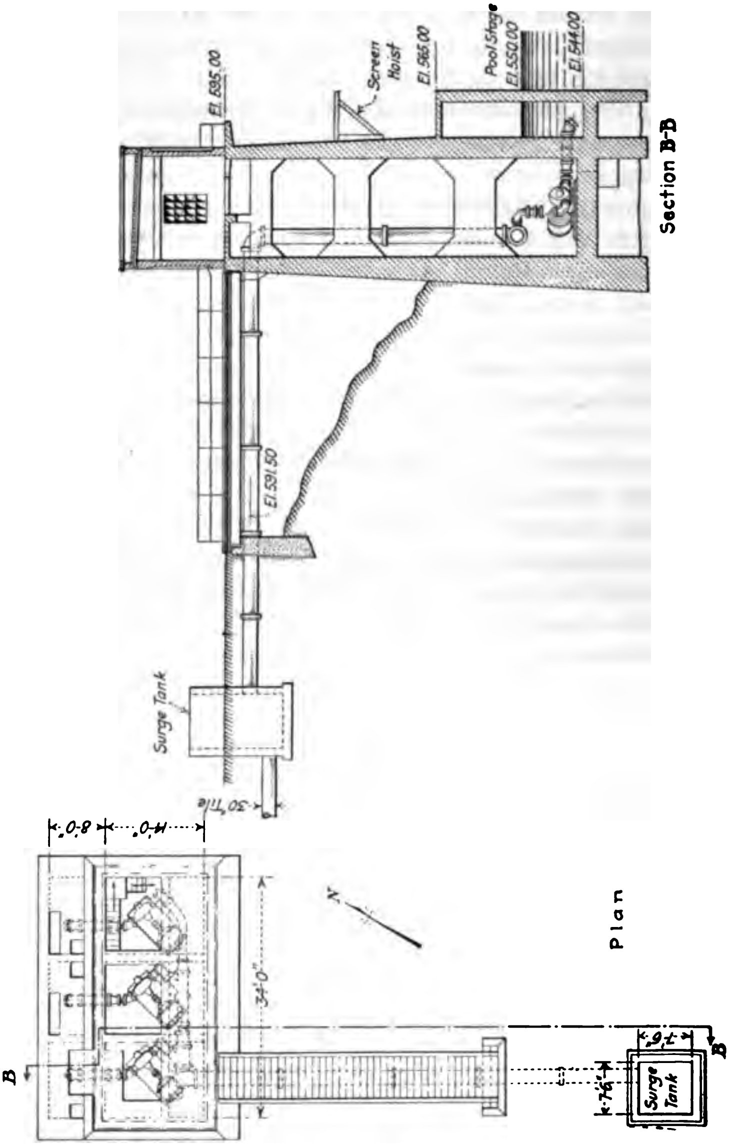


FIG. 20 GENERAL ARRANGEMENT OF RIVER PUMP HOUSE

has a capacity of 25,000,000 gal. The maximum demand of the plant for industrial water is estimated to be approximately 5,000,000 gal. per 24 hr., thus the reservoir provides at least a five-day settling period for the raw river water.

101 The river water was analyzed by the U. S. Geological Service in order to determine its adaptability for industrial uses within the plant, and while the turbidity is fairly high, it was rated as "fair for boiler purposes." It was decided that by taking more than the usual precautions in screening at the river and providing the five-day settling period it could be used satisfactorily for all industrial purposes. The surge tank provided an opportunity for the introduction of a coagulant if required.

102 Both the upper and lower dams were formed by rolling 6-in. layers of the spoil from building excavations and from the bed of the basin when the top soil had been stripped. As the natural drainage of the surface of more than half the reservation comes into the valley in which the settling basin was located, it was necessary to provide a concrete conduit underneath the full length of the reservoir and arrangements have been made for this conduit to be extended the full distance across the property in the future.

103 The water is carried from the reservoir to the service building in a reinforced-concrete flume where the distributing pumps are located in a pump pit which insures a positive head for the pumps. Distribution is at 60 lb. pressure.

104 An attempt has been made to provide a complete loop system around the plant with plenty of cross-connections to insure efficient operation.

105 As an additional precaution in the remote case that the spray pumps should fail to operate immediately, when the plate is placed in the spray pit for quenching, a standpipe 25 ft. in diameter and 100 ft. high is being provided in the east court of the forge shop. The failure of a plate may result unless it is quenched within a very few minutes after being taken from the furnace. As an average armor plate at this stage of the process and at the present prices for armor is worth about \$50,000, the expenditure of \$30,000 to meet and provide for the contingency is justified. Cross-connections have been provided so that the water in this standpipe is also available for the cooling systems throughout the plant in the extremely remote possibility of all sources of power going off simultaneously.

106 A separate *city water system* has been provided to supply

water for drinking and lavatory purposes. This water will also be used for 2500-lb. pressure water at the hydraulic press and forging cranes.

107 *Sewerage.* Sewerage, waste water and, in certain cases, roof water are gathered from branch systems into the trunk line which along with the other distribution systems of the plant, including gas, water and electrical conduit, follows the general line of the main west road and the main railroad track across the upper dam and ultimately reaches an outfall several hundred feet below the river pump station. The trunk line is 24 in. in diameter.

108 *Compressed Air.* Compressed air is piped at 100 lb. pressure from the service building to the main buildings in a loop system, large quantities being used in the open-hearth, forge shop and machine shop, and also in the blacksmith shop, where air will be used to operate the hammers. A complete loop system is provided.

109 *Natural Gas.* Natural gas is obtained from the fields some miles away and will be delivered at the plant at 50 lb. pressure. A loop system is provided with reducer valves located at convenient points throughout the plant. The largest quantities are used in the open-hearth furnaces and in the furnaces in the forge and furnace building. In the latter building a double loop is provided and meters are arranged so that the quantities of gas used in the forge shop and in the treatment shop may be measured separately.

110 A complete producer plant was designed and laid out south of the open-hearth stock yard before it was definitely established that natural gas would be obtained. It was at one time contemplated that oil would be burned in the forge furnaces. The natural-gas supply may be exhausted, it is believed, within the course of the next ten years and with this in view sufficient space has been provided for double checker chambers for the open-hearth furnaces in case producer gas must be resorted to and the gas preheated as well as the air. The foundations for the typical Carbottom furnaces in the forge building were also arranged so that double checker chambers could be installed. It is, however, the writer's personal opinion that the greatest development in the future is going to come in the use of pulverized coal as a fuel. This was in mind when the foundations for the furnaces were designed. Space left for double checker chambers can be equally well used for ash baffles.

111 *Boiler Plant.* The question of providing the best method of drive for the 14,000-ton hydraulic press was gone into very

thoroughly and a most interesting method of adapting the reversing rolling-mill drive system to a pressure pump to work momentarily against 7000 lb. per sq. in. was developed. Although the initial cost would have been higher, a considerable saving each year would have resulted over the method of steam intensification adopted.

112 It so developed, however, that the initial cost of this latter drive was reduced possibly half a million dollars by the obtaining of a complete boiler plant and a building in which to house it through army salvage.

113 Eight 823-hp. Stirling boilers were obtained from the power plant of the Old Hickory Powder Plant at Nashville, Tenn., and also a separate building of sufficient size to house them. The boilers and the steel frame of the buildings were transported to Charleston and after considerable modification have been re-erected as a modern boiler plant, to furnish steam at 200 lb. pressure for the 14,000-ton press which is now being installed and the 6500-ton press which will be erected in the future. Additional capacity will be available to operate a stand-by power plant of generous proportions and take care of the heating of the machine shop and gun treatment buildings.

114 The eight boilers are arranged four on a side facing each other. The automatic stokers are fed from an overhead concrete bunker of approximately 1000 tons capacity. This in turn is filled by a bucket conveyer from a hopper and crusher located under the railroad siding at the end of the building.

115 The operating floor of the boiler plant is 17 ft. above the ground level so that the grates will be dumped into hopper cars on narrow-gage track and the ash used for fill. Two 12-ft.-diameter by 200-ft.-high radial-brick chimneys have been erected.

116 A 16-in. steam main will run from the boiler plant directly to the press room of the forge shop and an additional line will run to the service building.

HOUSING

117 Five quarters for officers have been built. There are two housing developments: Armor Park comprises 85 hollow-tile houses and was built by the Housing Bureau during the War; Bungalow Park comprises 33 six-room ready-cut bungalows and was built by the construction division for the supervisory and drafting force.

PROGRESS OF CONSTRUCTION

118 Ground was broken for temporary storage sheds the latter part of September 1918, and barracks and mess halls were built to accommodate about 1000 men.

119 A factor which aided in getting the job under way was the obtaining of surplus material and used equipment from the Potomac Park construction work in Washington, from Navy construction work at Erie, Pa., and from the Army Powder Plant at Nitro, W. Va., about 12 miles away. A large quantity of equipment and material for the entire plant has been obtained from Army salvage through the efforts of the Supply Officer, Commander W. A. Merritt.

120 Grading for the classification yard and excavation and foundation work for the open-hearth building and machine shop were in full swing by January 1919. The first steelwork was erected in the open-hearth building on May 30, 1919.

121 Excavation and concreting of the foundations for the forge and furnace buildings were undertaken in July and completed during the month of August, about 8000 cu. yd. of concrete being placed over the extended area in less than six weeks. Fig. 21 shows a typical main-row column footing for this building.

122 During the past year the steelwork has been completed on all the buildings except the gun-treatment building. All buildings have been enclosed with walls, sash and roof. Foundations have been completed for the 25 furnaces in the forge shops and for the 14,000-ton press, involving in all some 15,000 cu. yd. of concrete. The 30-ton electric furnaces have been erected. The majority of the cranes have been put into operation. The boiler plant is nearing completion, also the service station. The gun pit involving nearly 5000 cu. yd. of concrete and the river pump station are practically done.

123 It should be stated that great difficulty has existed throughout the entire job in securing and maintaining sufficient qualified engineers and draftsmen, and despite continued efforts there has been at no time sufficient labor to secure entirely satisfactory progress. All employment has been through the Civil Service. The general unsettled condition in the material market both before and after the armistice and the great uncertainty of deliveries due to transportation conditions have been, as in other lines, a great hindrance.

124 The construction work remaining to be completed con-

sists chiefly of the distribution systems for the plant services, the concreting of the roads and the installation of lockers, toilets and offices. Bottoms should be burned in the open-hearth furnaces and the first armor ingots cast by the time this paper is presented. The majority of forge furnaces are now completed and the 14,000-ton press should be erected and in operation by the first of the year.

ORGANIZATION OF PROJECT AND PERSONNEL

125 The United States Naval Ordnance Plant is operated under the direction of the Bureau of Ordnance of the Navy Depart-

Fig. 21 Typical Column Footing for Row *F*, Forge and Furnace Building, August 29, 1919

ment. It was designed and built under the joint direction of that Bureau and the Bureau of Yards and Docks. The entire design and construction of the armor and gun forging plant was done within the Navy and without recourse to a general contractor.

126 Admiral Ralph Earle was Chief of Bureau of Ordnance until recently relieved by Admiral McVey. The Armor Plant Division of the Bureau has been under the direction of Commander Logan Cresap.

127 The Chief of Bureau of Yards and Docks has been Admiral C. M. Parks. The Assistant Chief has been Captain R. E. Bakenhus who has also been Project Manager of the Ordnance Plant Division of the Bureau of Yards and Docks as well as Manager of Shipyard Plants for the Emergency Fleet Corporation.

128 The original Inspector of Ordnance in charge at the Naval Ordnance Plant was Commander J. B. Rhodes. He was relieved about a year ago by Captain George R. Marvel.

129 The specifications for and purchasing of the 14,000-ton press and its driving equipment and the detailed design of all furnaces and similar equipment were handled by the steel superintendent, Mr. W. J. Priestly, who is head of the Hot Metal Department. The machine-tool layout and the specifying and purchasing of the equipment was done by the machine-shop superintendent, Mr. W. E. Hayes, who is the head of the Cold Metal Department.

130 The general layout, industrial and structural design, design of plant services, crane specifications, foundation design for all equipment, etc., were made under the author's direction in the Bureau of Yards and Docks at Washington, and details completed at South Charleston when the engineering force, with the exception of the Electrical Division, was transferred there in the spring of 1919 to be in closer touch with the work. A Construction and Maintenance Department was organized at the Naval Ordnance Plant to build the new plant and carry on the maintenance of the Projectile Plant. It employed at maximum strength some 40 to 50 designers and draftsmen and somewhat over 1200 men in the field. The principal assistants of the writer, as head of the Construction and Maintenance Department, were Mr. J. Hepinstall, supervising engineer of design; Mr. T. H. Callahan, supervising engineer of construction; Mr. W. H. Sears, materials and office, and Mr. H. M. Cogan, supervising engineer of electrical design.

CONTRACTS

131 The major contracts were awarded as follows: American Bridge Company, structural steel ($2,500,000); Morgan Engineering Company, Manning, Maxwell and Moore Company, Cleveland Crane Company, cranes ($1,000,000); Detroit Steel Products Company, steel sash; Keystone Gypsum Fireproofing Company, roof decks; Johns-Manville Company, roofing; General Electric Company, electrical equipment for all cranes; Westinghouse Company,

principal units of substation. All radial-brick chimneys have been erected by the M. W. Kellogg Company. The boiler plant was dismantled at Old Hickory and re-erected at South Charleston by the Shevlin Engineering Company. The Mesta Machine Company is building the 14,000-ton press. The Swindell Company is building the open-hearth and forge-shop furnaces. The majority of the machine tools have been built by the Niles-Bement-Pond Company. Foundations for press, furnaces and machine tools were built by the Construction Division.

COSTS

132 No paper of this nature would be complete without some reference as to cost. In round numbers the principal funds available for the Naval Ordnance Plant were the original Armor Plant appropriation of about \$11,000,000, and an additional appropriation of about \$4,500,000 to cover the consolidation of a gun-forging plant with the armor plant. A separate appropriation for the Projectile Plant of about \$2,000,000 was provided. A large amount of material and equipment obtained after the armistice from Army salvage and from Navy cost-plus plants has been used.

133 Final figures of cost are not yet available. In round numbers it may be stated that the entire plant when completed will represent an investment of about \$20,000,000. Construction costs of the plant including cost of all structures completely enclosed and with services including overhead cranes, but exclusive of operating equipment and equipment foundations, will total somewhat over \$10,000,000. The major items may be very roughly classified as follows and are given purely as orders of magnitude.

Open-hearth building	\$1,800,000
Forge and furnace building	2,100,000
Machine shops	1,600,000
Gun-treatment building	800,000
Plant services, buildings, equipment and distribution systems for water, electric power and light, air, gas, etc.	2,500,000
Surface drainage, roads, etc.	300,000
Track systems and yards	400,000

134 The cost records, kept both by the intricate Navy system of accounting which imposes a more or less arbitrary overhead based on the amount of productive labor; and on a man-hour and material

basis kept separately by the author, indicate that the work has been done economically. The percentage profit of a general contractor has been saved to the Department. A saving has also been made by the use of Navy cost-plus and Army salvage material and equipment.

135 The method adopted of handling the entire construction of the armor and gun-forging plant within the Department proved very flexible and made it possible to keep the design work and the construction work in close harmony. It also permitted essential eleventh-hour changes to be made with less inconvenience and delay than would have been possible dealing through a general contractor.

136 If armor ingots are cast in December, as is confidently expected, it will have taken two years and six months, in exceedingly trying times as to labor and material, to bring the plant from the "open-lot and blank-paper" stage to operation.

137 At the time of going to press, October 1921, the open-hearth and electric furnaces have been in operation since February 2, 1921.

No. 1778

A. S. M. E. BOILER CODE

PART I—SECTION III

RULES FOR THE CONSTRUCTION OF BOILERS OF LOCOMOTIVES WHICH ARE NOT SUBJECT TO FEDERAL INSPECTION AND CONTROL

Selection of Materials

L–1 Specifications are given in the Rules for Power Boilers, Pars. 23 to 178, for the important materials used in the construction of boilers, and where so given, the materials herein mentioned shall conform thereto, except as noted in Par. L–18.

L–2 Steel plates for any part of a boiler when exposed to the fire or products of combustion, and under pressure, *excepting front tube sheets*, shall be of firebox quality as designated in the Specifications for Boiler Plate Steel.

L–3 Steel plates for any part of a boiler, where firebox quality is not specified, when under pressure, shall be of firebox or flange quality, as designated in the Specifications for Boiler Plate Steel.

L–4 Braces, when welded, shall be of wrought iron of the quality designated in the Specifications for Refined Wrought-Iron Bars.

L–5 Manhole and handhole covers and other parts subjected to pressure, and braces and lugs when made of steel plate, shall be of firebox or flange quality, as designated in the Specifications for Boiler Plate Steel.

L–6 Steel bars for braces and for other boiler parts, except as otherwise specified herein, shall be of the quality designated in the Specifications for Steel Bars.

L–7 Staybolts shall be of iron of the quality designated in the Specifications for Staybolt Iron.

L–8 Rivets shall be of steel or iron of the quality designated in the Specifications for Boiler Rivet Steel, or in the Specifications for Boiler Rivet Iron.

L–9 Throttle and throttle pipe, dry pipe or dry pipe ring, tee head, superheater header and steam pipes to cylinders, may be of cast iron.

L–10 Water-leg and door-frame rings shall be of wrought iron or steel, or cast steel of Class A or Class B grade, as designated in the Specifications for Steel Castings. The OG or other flanged construction may be used as a substitute in any case.

Ultimate Strength of Material Used in Computing Joints

L–11 In determining the maximum allowable working pressure, the tensile strength used in the computations for steel plates shall be that stamped on the plates as herein provided, which is the minimum of the stipulated range, or 55,000 lb. per sq. in. for all steel plates, except for special grades having a lower tensile strength.

L–12 The resistance to crushing of steel plate shall be taken at 95,000 lb. per sq. in. of cross-sectional area.

L–13 In computing the ultimate strength of rivets in shear, the following values in pounds per square inch of the cross-sectional area of the rivet shank shall be used:

Iron rivets in single shear	38,000
Iron rivets in double shear	76,000
Steel rivets in single shear	44,000
Steel rivets in double shear	88,000

The cross-sectional area used in the computations shall be that of the rivet shank after driving.

Minimum Thickness of Plates and Tubes

L–14 The minimum thickness of any boiler plate under pressure shall be $\frac{1}{4}$ in.

L–15 The minimum thickness of shell plates, and dome plates after flanging, shall be as follows:

When the Inside Diameter of Shell is			
36 in. or Under	Over 36 in. to 54 in.	Over 54 in. to 72 in.	Over 72 in.
$\frac{1}{4}$ in.	$\frac{5}{16}$ in.	$\frac{3}{8}$ in.	$\frac{1}{2}$ in.

L–16 The minimum thickness of butt straps for double strap joints shall be as given in Table L-1. Intermediate values shall be determined by interpolation. For plate thicknesses exceeding $1\frac{1}{4}$ in., the thickness of the butt straps shall not be less than two-thirds of the thickness of the plate.

TABLE L-1 MINIMUM THICKNESS OF BUTT STRAPS

Thickness of Shell Plates, Inches	Minimum Thickness of Butt Straps, Inches	Thickness of Shell Plates, Inches	Minimum Thickness of Butt Straps, Inches
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{17}{32}$	$\frac{7}{16}$
$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{7}{16}$
$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$
$\frac{11}{32}$	$\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	$\frac{5}{8}$
$\frac{13}{32}$	$\frac{5}{16}$	1	$\frac{11}{16}$
$\frac{7}{16}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$
$\frac{15}{32}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{7}{8}$
$\frac{1}{2}$	$\frac{7}{16}$		

L–17 The minimum thickness of tube sheets for locomotive boilers shall be as follows:

When Diameter of Front Tube Sheet is

25 in. or Under	Over 25 in. to 42 in.	Over 42 in. to 54 in.	Over 54 in.
$\frac{5}{16}$	$\frac{3}{8}$ in.	$\frac{7}{16}$ in.	$\frac{1}{2}$ in.

L–18 The gage thickness of tubes or flues exposed to the products of combustion on the inside shall not be less than that specified in Table L-2 for the various pressures and outside diameters given.

TABLE L-2 GAGE THICKNESS OF WALLS OF FIRE TUBES OR FLUES

Outside Diameter, In.	Maximum Allowable Working Pressure, Lb. per Sq. In.				
	160	180	200	225	250
$1\frac{1}{2}$	13	13	12		
$1\frac{3}{4}$	13	12	12	..	..
2	12	12	12	11	11
$2\frac{1}{4}$	12	12	12	11	11
$2\frac{1}{2}$	12	12	11	11	11
3	12	11	11	11	10
$3\frac{1}{2}$	11	11	11	10	10
4	11	10	10	10	9
$4\frac{1}{2}$	10	10	10	9	9
5	10	9	9	9	8
$5\frac{1}{2}$	9	9	9	8	8
6	9	9	8	8	7

The gage thickness specified in Table L-2 is that measured by the B. W. gage; the thickness at any section must not vary more than one gage below or one gage above that specified in the case of cold-finished tubes, or more than one gage below or two gages above that specified in the case of hot-finished tubes. In the case of superheater flues which are expanded, the gage of the expanded end may be 1½ gages lighter and the swaged end two gages heavier than the gage thickness.

NOTE:—Variations in gage thickness permitted in above paragraph and gage thickness of fire tubes specified in Table L-2 are subject to revision in event of uniform tube specification for all boilers being adopted by interests concerned in these Rules.

L-19 Brick arch tubes shall be of seamless steel or lap-welded charcoal iron and their minimum thickness shall be determined by the following formula:

$$t = \frac{PD}{16{,}000} + \frac{1}{8} \text{ in.}$$

where P = allowable boiler pressure, lb. per sq. in.
t = thickness of walls, in.
D = outside diameter, in.

CONSTRUCTION AND MAXIMUM ALLOWABLE WORKING PRESSURE FOR BOILERS OF LOCOMOTIVES

L-20 The maximum allowable working pressure is determined by employing the factors of safety, stresses and dimensions designated in these Rules.

The factor of safety used in design and construction of new boilers shall not be less than 4.5.

The factor of safety used in determining the maximum allowable working pressure calculated on the conditions actually obtaining in service shall not be less than 4.0.

The maximum allowable working pressure determined by conditions obtaining in service shall not exceed that for which the boiler was designed.

No boiler shall be operated at a higher pressure than the maximum allowable working pressure, except when the safety valve or valves are blowing, at which time the maximum allowable working pressure shall not be exceeded by more than 5 per cent.

L-21 The maximum allowable working pressure on the shell of a boiler shall be determined by the strength of the weakest course, computed from the thickness of the plate, the tensile strength stamped thereon, as provided for in Specifications for Boiler Plate Steel, the efficiency of the longitudinal joint, the inside diameter of the course, and the factor of safety.

$$\frac{TS \times t \times E}{R \times FS} = \text{maximum allowable working pressure, lb. per sq. in.}$$

where

TS = ultimate tensile strength stamped on shell plates, as provided for in Specifications for Boiler Plate Steel, lb. per sq. in.

t = minimum thickness of shell plates in weakest course, in.

E = efficiency of longitudinal joint

R = inside radius of the weakest course of the shell, in.

FS = factor of safety, or the ratio of the ultimate strength of the material to the allowable stress.

Boiler Joints

L-22 The efficiency of a joint is the ratio which the strength of the joint bears to the strength of the solid plate. In the case of a riveted joint this is determined by calculating the breaking strength of a unit section of the joint, considering each possible mode of failure separately, and dividing the lowest result by the breaking strength of the solid plate of a length equal to that of the section considered. (See Appendix, Pars. 410 to 416, Power Boilers, for detailed methods and examples.)

L-23 The distance between the center lines of any two adjacent rows of rivets, or the "back pitch" measured at right angles to the direction of the joint, shall have the following minimum values:

a If $\frac{p}{d}$ is 4 or less, the minimum value shall be 1.75 d;

b If $\frac{p}{d}$ is over 4, the minimum value shall be:

$$1.75\, d + 0.1\, (p - 4\, d)$$

where

p = pitch of rivets in outer row where a rivet in the inner row comes midway between two rivets in the outer row, in.

p = pitch of rivets in the outer row less pitch of rivets in the inner row where two rivets in the inner row come between two rivets in the outer row, in. (It is here assumed that the joints are of the usual construction where the rivets are symmetrically spaced.)

d = diameter of the rivet holes, in.

L-24 On longitudinal joints the distance from the centers of rivet holes to the edges of the plates, except rivet holes in the ends of butt straps, shall be not less than one and one-third times the diameter of the rivet holes.

L-25 The strength of circumferential joints of boilers shall be at least 50 per cent of that required for the longitudinal joints of the same structure.

L-26 The longitudinal joints of a shell which exceeds 36 in. in diameter shall be of butt and double-strap construction.

This rule does not apply to the portion of a boiler shell which is stayed to the firebox or combustion chamber.

L-27 With butt and double-strap construction longitudinal joints of any length may be used, provided the tension test specimens are so cut from shell plates and butt-strap plates that their lengthwise direction is parallel with the circumferential seams of the boiler, and the tests meet the standards prescribed in the Specifications for Boiler Plate Steel.

L-28 Butt straps and the ends of shell plates forming the longitudinal joints shall be rolled or formed by pressure, not blows, to the proper curvature.

L-29 The ultimate strength of a joint which has been properly welded by the forging process shall be taken as 28,500 lb. per sq. in. with steel plates having a range in tensile strength of 45,000 to 55,000 lb. per sq. in. Autogenous welding may be used in boilers in cases where the strain is carried by other construction which conforms to the requirements of the Code and where the safety of the structure is not dependent upon the strength of the weld.

L-30 The longitudinal joint of a dome 24 in. or over in inside diameter shall be of butt and double-strap construction or made without a seam of one piece of steel pressed into shape, and its flange shall be double-riveted to the boiler shell. In the case of a dome less than 24 in. in diameter, for which the product of the inside diameter and the maximum allowable working pressure does not exceed

4000 in.-lb., its flange may be single-riveted to the boiler shell and the longitudinal joint may be of the lap type provided it is computed with a factor of safety not less than 8.

When boiler shells are cut to apply steam domes or manholes, the net area of metal, after rivet holes are deducted, in flange and liner, if used, must be not less than the area required by these rules for a length of boiler shell equal to the length removed. A height of vertical flange equal to three times the thickness of the flange shall be included in the area of the flange.

Braced and Stayed Surfaces

L-31 The maximum allowable working pressure for various thicknesses of braced and stayed flat plates and those which by these Rules require staying as flat surfaces with braces or stays of uniform diameter symmetrically spaced, shall be calculated by the formula:

$$P = C \times \frac{T^2}{p^2}$$

where

P = maximum allowable working pressure, lb. per sq. in.

T = thickness of plate in sixteenths of an inch

p = maximum pitch measured between straight lines passing through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined, in.

C = 125 for stays screwed through plates not over $\frac{7}{16}$ in. thick with ends riveted over, see Table L-3

C = 135 for stays screwed through plates over $\frac{7}{16}$ in. thick with ends riveted over, see Table L-3

C = 150 for stays screwed through plates and fitted with single nuts outside of plate

C = 165 for stays with heads not less than 1.3 times the diameter of the stays, screwed through plates or made a taper fit and having the heads formed on the stays before installing them and not riveted over, said heads being made to have a true bearing on the plate.

If flat boiler plates not less than $\frac{5}{16}$ in. thick are strengthened with doubling plates securely riveted thereto and having a thickness of not less than $\frac{2}{3}$ T, then the value of T in the formula shall be three-quarters of the combined thickness of the boiler plate and doubling plates, but not more than one and one-half times the thickness of the boiler plate, and the values of C given above may also be increased 15 per cent.

When two sheets are connected by stays and but one of these sheets requires staying, the value of C is governed by the thickness of the sheet requiring staying.

Where values for screwed stays with ends riveted are required for conditions not given in Table L-3, they may be computed from the formula in Par. L-31 and used, provided the pitch does not exceed $8\frac{1}{2}$ in.

L-32 The ends of screwed staybolts shall be riveted over or upset by equivalent process. The outside ends of solid staybolts other than crown stays 14 in. and less in length shall be drilled with a hole at least $\frac{3}{16}$ in. in diameter, to a depth extending at least $\frac{1}{2}$ in. beyond the inside of the plates, or hollow staybolts may be used. Crown stays and flexible staybolts of either the jointed or ball and socket type need not be drilled.

Staybolts behind permanent brickwork, frame braces, or grate bearers shall have holes $\frac{3}{16}$ in. diameter for their entire length, which must be kept open at all times.

L-33 When channel irons or other members are securely riveted to the boiler heads, the stress on such members shall not exceed 12,500 lb. per sq. in. In computing the stress the section modulus of the member shall be used without addition for the strength of the plate. The spacing of the rivets over the supported surface shall be in conformity with that specified for staybolts.

If the outstanding legs of the two members are fastened together so that they act as one member in resisting the bending action produced by the load on the rivets attaching the members to the head of the boiler, and provided that the spacing of those rivets attaching the members to the head is approximately uniform, the members may be computed as a single beam uniformly loaded and supported at the points where the through braces are attached.

L-34 The ends of stays fitted with nuts shall not be exposed to the direct radiant heat of the fire.

L-35 *a* The maximum spacing between centers of rivets or between the edges of tube holes and the centers of rivets attaching the crowfeet of braces to the braced surface, shall be determined by the formula in Par. L-31, using 150 for the value of C.

b The maximum distance between the edges of tube holes and the centers of other types of stays shall be determined by the formula in Par. L–31, using the value of C given for the thickness of plate and type of stay used.

c The maximum spacing between the inner surface of the shell and lines parallel to the surface of the shell passing through the centers of the rivets attaching the crowfeet of braces to the head shall not exceed p as determined by the formula in Par. L–31, using 150 for the value of C, plus the outside radius of the flange.

d The maximum distance between the inner surface of the shell and the centers of braces of other types shall not exceed p as determined by the formula in Par. L–31, using that value of C which applies to the thickness of plate and type of stay as therein specified, plus the outside radius of the flange.

e In applying these Rules and those in Par. L–31 to a head or plate having a manhole or reinforced opening, the spacing applies only to the plate around the opening and not across the opening.

L–36 *a* When the edge of a stayed plate is flat and is fastened by riveting, the distance from the center line of the rivets to a line through the centers of the nearest row of stays may be made to equal the pitch of the stays as given in Table L-3 plus twice the thickness of the plate.

b When the edge of a flat stayed plate is flanged and riveted, the distance from the center of the outermost stays to the inside of the supporting flange shall not exceed the pitch of the stays as given in Table L-3 plus the inside radius of the flange.

L–37 The distance between the edges of the staybolt holes may be substituted for p for staybolts adjacent to a furnace door or other boiler fitting, tube hole, handhole or other opening.

L–38 The diameter of a screw stay shall be taken at the bottom of the thread, or at the body of the bolt between the threads, whichever is the lesser.

L–39 The least cross-sectional area of a stay shall be taken in calculating the allowable stress, except when the stays are welded and have a larger cross-sectional area at the weld than at some other point, in which case the strength at the weld shall be computed as well as in the solid part and the lower value used.

L–40 Holes for screw stays shall be drilled full size or punched not to exceed $\frac{1}{4}$ in. less than full diameter of the hole for plates over $\frac{5}{16}$ in. in thickness, and $\frac{1}{8}$ in. less than the full diameter of the hole for plates not exceeding $\frac{5}{16}$ in. in thickness, and then drilled or reamed to the full diameter. The holes shall be tapped fair and true, with a full thread.

TABLE L-3 MAXIMUM ALLOWABLE PITCH, IN INCHES, OF SCREWED STAYS, ENDS RIVETED OVER, FOR FLAT PLATE

Pressure, Lb. per Sq. In.	Thickness of Plate, In.						
		3/8	7/16	1/2	9/16	5/8	11/16
	Maximum Pitch of Staybolts, In.						
100	5.590	6.708	7.826	..	..	..	..
110	5.331	6.396	7.462	..	..	..	..
120	5.103	6.124	7.144	..	..	..	..
125	5.000	6.000	7.000	..	..	..	..
130	4.903	5.883	6.864	..	..	..	..
140	4.725	5.670	6.614	7.856	..	..	..
150	4.564	5.477	6.390	7.590	..	..	..
160	4.419	5.303	6.187	7.349	..	..	..
170	4.287	5.145	6.003	7.129	..	..	..
180	4.167	5.000	5.833	6.928	7.794	..	..
190	4.056	4.867	5.678	6.743	7.587	..	..
200	3.953	4.743	5.534	6.573	7.394	..	..
225	3.727	4.472	5.218	6.197	6.971	7.746	..
250	3.536	4.243	4.950	5.879	6.614	7.349	..
300	3.227	3.873	4.519	5.367	6.037	6.708	7.379

L-41 The ends of steel stays upset for threading shall be thoroughly annealed.

L-42 In curved sheets of a combustion chamber subject to external pressure, of radius R in., the maximum allowable working pressure shall be calculated by the formula

$$P = C\frac{T^2}{p^2} + 250\frac{T}{R}$$

provided p^2 does not exceed 0.008 CTR and in no case shall p exceed $2T$. The stress per square inch in staybolts shall not exceed 7500 lb., based on the unit pressure $P - 250\frac{T}{R}$

L-43 *a* The maximum allowable working pressure for any curved stayed surface subject to internal pressure shall be obtained by the two following methods, and the minimum value obtained shall be used:

First, the maximum allowable working pressure shall be computed without allowing for the holding power of the stays, due allowance being made for the weakening effect of the holes for the stays. To this pressure there shall be added the pressure secured by the formula for braced and stayed surfaces given in Par. L–31, using 80 for the value of C.

Second, the maximum allowable working pressure shall be computed without allowing for the holding power of the stays, due allowance being made for the weakening effect of the holes for the stays. To this pressure there shall be added the pressure corresponding to the strength of the stays for the stresses given in Table L–4, each stay being assumed to resist the steam pressure acting on the full area of the external surface supported by the stay.

TABLE L–4 MAXIMUM ALLOWABLE STRESSES FOR STAYS AND STAYBOLTS

Description of Stays	Stresses, Lb. Per Sq. In.	
	For lengths between supports not exceeding 120 diameters	For lengths between supports exceeding 120 diameters
a Unwelded or flexible stays less than twenty diameters long, screwed through plates, with ends riveted over..............	7500	.....
b Hollow stays less than 20 diameters long, screwed through plates, with ends riveted over........................	8000	.....
c Unwelded stays and unwelded portions of welded stays, except as specified in line *a* and line *b*..................	9500	8500
d Welded portions of stays..................	6600	6600

b The maximum allowable working pressure for a stayed wrapper sheet of a locomotive boiler shall be determined by the two methods given above and by the method which follows, and the minimum value obtained shall be used:

$$P = \frac{55{,}000}{FS} \times \frac{t \times E}{R - s\Sigma \sin \alpha}$$

in which

α = angle any crown stay makes with vertical axis of boiler

$\Sigma \sin \alpha$ = summated value of $\sin \alpha$ for all crown stays considered in one transverse plane and on one side of vertical axis of boiler

s = transverse spacing of crown stays in crown sheet, in.

E = minimum efficiency of wrapper sheet through joints or stay holes

t = thickness of wrapper sheet, in.
R = radius of wrapper sheet, in.
P = maximum allowable working pressure, lb. per sq. in.
FS = factor of safety, or the ratio of the ultimate strength of the material to the allowable stress.

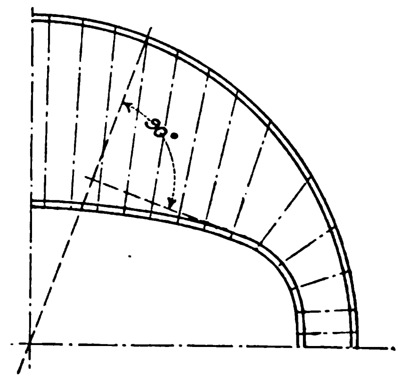


FIG. L-1 STAYING FOR WRAPPER SHEET OF LOCOMOTIVE BOILER

The above formula applies to the longitudinal center section of the wrapper sheet, and in cases where E is reduced at another section, the maximum allowable working pressure, based on the strength at that section, may be increased in the proportion that the distance from the wrapper sheet to the top of the crown sheet at the center, bears to the distance, measured on a radial line through the other section, from the wrapper sheet to a line tangent to the crown sheet and at right angles to the radial line. See Fig. L-1.

L-44 A segment of a head shall be stayed by through, diagonal, crowfoot or gusset stays.

L-45 The area of a segment of a head to be stayed shall be the area enclosed by lines drawn 2 in. from the tubes and at a distance d from the shell as shown in Figs. 15 and 16, Part I, Section I. The value of d used may be the larger of the following values:

(1) d = the outer radius of the flange, not exceeding 8 times the thickness of the head

(2) $$d = \frac{5 \times T}{\sqrt{P}}$$

where

d = unstayed distance from shell in inches
T = thickness of head in sixteenths of an inch
P = maximum allowable working pressure in lb. per sq. in.

The feet for braces to back head and front tube sheet should be distributed so as not to concentrate the stress on any one section; preferably a proportion of the braces should be attached to the second course from the back head or front tube sheet.

No supporting value shall be assigned to the stiffness of inside liner plates on flat surfaces except as provided in L–31.

L–46 In calculating stresses for stays and their attachments the angularity of the stays, if in excess of 15 deg., must be taken into account.

L–47 All rivet and pin holes shall conform to the requirements in Par. L–53 and the pins shall be made a neat fit. To determine the sizes that shall be used, proceed as follows:

1 Determine the "required cross-sectional area of the brace" by first computing the total load to be carried by the brace, and dividing the total load by the values of stresses given in Table L–4.
2 Design the body of the brace so that the cross-sectional area shall be at least equal to the "required cross-sectional area of the brace."
3 Make the area of pins to resist double shear at least three-quarters of the "required cross-sectional area of the brace."
4 Make the combined cross-section of the eye at the side of the pin (in crowfoot braces) at least 25 per cent greater than the "required cross-sectional area of the brace."
5 Make the cross-sectional areas through the blades of diagonal braces where attached to the shell of the boiler at least equal to the required rivet section; that is, at least equal to one and one-quarter times the "required cross-sectional area of the brace."
6 Design each branch of a crowfoot to carry two-thirds of the total load on the brace.
7 Make the net sectional areas through the sides of the crowfoot, tee irons, or similar fastenings at the rivet holes at least equal to the required rivet section; that is, at least equal to one and one-quarter times the "required cross-sectional area of the brace."
8 Make the combined cross-sectional area of the rivets at each end of the brace at least one and one-quarter times the "required cross-sectional area of the brace."

L–48 Gusset stays when constructed of triangular web plates secured to single or double angle bars along the two sides at right angles shall have a cross-sectional area (in a plane at right angles to the longest side and passing through the intersection of the two shorter sides) not less than 10 per cent greater than would be required for a diagonal stay to support the same surface, assuming the diagonal stay is at the same angle as the longest side of the gusset plate.

L-49 Crown bars and girder stays for tops of combustion chambers and back connections, or wherever used, shall be proportioned to conform to the following formula:

$$\text{Maximum allowable working pressure} = \frac{C \times d^2 \times t}{(W - P) \times D \times W}$$

where

W = extreme distance between supports, in.
P = pitch of supporting bolts, in.
D = distance between girders from center to center, in.
d = depth of girder, in.
t = thickness of girder, in.
C = 7000 when the girder is fitted with one supporting bolt
C = 10,000 when the girder is fitted with two or three supporting bolts
C = 11,000 when the girder is fitted with four or five supporting bolts
C = 11,500 when the girder is fitted with six or seven supporting bolts
C = 12,000 when the girder is fitted with eight or more supporting bolts.

Example: Given W = 34 in., P = 7.5 in., D = 7.75 in., d = 7.5 in., t = 2 in.; three stays per girder, C = 10,000; then, substituting in formula:

Maximum allowable working pressure =

$$\frac{10{,}000 \times 7.5 \times 7.5 \times 2}{(34 - 7.5) \times 7.75 \times 34} = 161.1 \text{ lb. per sq. in.}$$

In boilers with crown bars supported on firebox side sheets and sling stays, the sling stays shall be considered as carrying the entire load.

L-50 *Tubes.* Tube holes shall be drilled full size from the solid plate, or they may be punched at least $\frac{1}{8}$ in. smaller in diameter than full size, and then drilled, reamed or finished full size with a rotating cutter.

L-51 The sharp edges of tube holes shall be taken off on both sides of the plate with a file or other tool.

L-52 The ends of the tubes shall be substantially rolled and beaded, or rolled and welded, at the firebox or combustion-chamber end, and rolled at the smokebox end; 10 per cent of the tubes shall be beaded at the smokebox end.

The ends of arch tubes must be belled out to a diameter $\frac{1}{4}$ in. larger than the hole in the sheet to which they are connected.

Riveting

L-53 All rivet holes and staybolt holes and holes in braces and lugs shall be drilled full size, or they may be punched not to exceed $\frac{1}{4}$ in. less than full diameter for material over $\frac{5}{16}$ in. in thickness, and $\frac{1}{8}$ in. less than full diameter for material not exceeding $\frac{5}{16}$ in. in thickness, and then drilled or reamed to full diameter. Plates, butt straps, braces, heads and lugs shall be firmly bolted in position by tack bolts for drilling or reaming all rivet holes in boiler plates, except those used for the tack bolts.

L-54 Rivets shall be of sufficient length to completely fill the rivet holes and form heads at least equal to those shown in Fig. 20, Part I, Section I.

L-55 Rivets shall be machine-driven wherever practicable, with sufficient pressure to fill the rivet holes, and shall be allowed to cool and shrink under pressure. Barrel pins fitting the holes and tack bolts to hold the plates firmly together shall be used. A rivet shall be driven each side of each tack bolt before removing the tack bolt.

L-56 The calking edges of plates, butt straps and heads shall be beveled to an angle not less than 70 deg. to the plane of the plate, and as near thereto as practicable. Every portion of the sheared surfaces of the calking edges of plates, butt straps and heads shall be planed, milled or chipped to a depth of not less than $\frac{1}{8}$ in. Calking shall be done with a round-nosed tool.

L-57 A locomotive boiler shall have washout handholes or screw plugs as follows: One at each corner of firebox just above mud ring; one in back head over fire door; one or more on each side of roof sheet above the level of crown sheet, which shall be staggered on opposite sides; one or more in barrel of boiler; and one or more in back head above crown sheet.

Screw plugs must have at least four full threads in the sheet, including reinforcement, if such is used.

L-58 All holes for injector checks, whistle, and safety valves when screwed into boiler, and all holes in boiler barrel, firebox, roof sheet, and all unstayed surfaces when diameter of the hole is over 3½ in. and exceeds 4½ times the thickness of the plate must be reinforced with a liner or flange riveted to the boiler.

The thickness of the liner or flange must be at least 75 per cent of the thickness of the plate. The rivets must have a shearing strength of at least 52 per cent of the tensile strength of the metal removed.

Safety Valves

L-59 Every locomotive boiler shall be equipped with at least two safety valves, the capacity of which shall be sufficient to prevent, under any conditions of service, an accumulation of pressure more than five per cent above the specified boiler pressure.

L-60 Safety valves shall be set to pop at pressures not exceeding 6 lb. above the working steam pressure. When setting safety valves, two steam gages shall be used, one of which must be so located that it will be in full view of the person engaged in setting such valves; and if the pressure indicated by the gages varies more than 3 lb., they shall be removed from the boiler, tested and corrected before the safety valves are set. Gages shall in all cases be tested immediately before the safety valves are set or any change is made in the setting. When setting safety valves, the water level in the boiler shall not be above the highest gage cock.

L-61 Safety valves may have the seat and bearing surface of the disk inclined at any angle between 45 deg. and 90 deg. to the center line of the spindle. The valves shall be rated at a pressure 3 per cent in excess of that at which the valve is set to blow.

All safety valves shall be so constructed that no detrimental shocks are produced through the operation of the valve.

L-62 Each safety valve shall be plainly marked by the manufacturer. The markings may be stamped on the body, cast on the body, or stamped or cast on a plate or plates permanently secured to the body, and shall contain the following:

a The name or identifying trademark of the manufacturer
b The nominal diameter
c The steam pressure at which it is set to blow
d Blow down, or difference between the opening and closing pressures
e The weight of steam discharged in pounds per hour at a pressure

3 per cent higher than that for which the valve is set to blow *f* A.S.M.E. Std.

L–63 Safety-valve capacity may be checked in the following manner; and, if found sufficient, additional capacity need not be provided:

By making an accumulation test with fire in good bright condition and all steam exits closed; with the fire forced under these conditions the safety valves should relieve boiler and not allow an excess pressure of more than 5 per cent above the working pressure.

L–64 Each safety valve shall have full-sized direct connection to the boiler.

L–65 If a muffler is used on a safety valve it shall have sufficient outlet area to prevent back pressure from interfering with the proper operation and discharge capacity of the valve. The muffler plates or other devices shall be so constructed as to avoid any possibility of restriction of the steam passages due to deposit.

L–66 When a boiler is fitted with two or more safety valves on one connection, this connection to the boiler shall have a cross-sectional area not less than the combined area of all of the safety valves with which it connects.

L–67 The seats and disks of safety valves shall be of non-ferrous material.

L–68 Springs used in safety valves shall not show a permanent set exceeding $\frac{1}{16}$ in. ten minutes after being released from a cold compression test closing the spring solid. The spring shall be so constructed that the valve can lift from its seat at least $\frac{1}{10}$ the diameter of the seat before the coils are closed or before there is other interference.

L–69 The spring in a safety valve shall not be used for any pressure more than 10 per cent above or below that for which it was designed.

L–70 When the valve body is marked with the letters A.S.M.E. Std. as required by Par. L–62, this shall be a guarantee by the manufacturer that the valve conforms to the details of construction herein specified.

L–71 Every boiler shall have proper outlet connections for the required safety valve or valves, independent of any other steam outlet connection or of any internal pipe in the steam space of the boiler, the area of opening to be at least equal to the aggregate nominal area of all of the safety valves to be attached thereto.

L-72 Each boiler shall have at least one water glass provided with top and bottom shut-off cocks, and lamp and two gage cocks for boilers 36 in. in diameter and under, and three gage cocks for boilers over 36 in. in diameter.

The lowest gage cock and the lowest reading of water glass shall not be less than 2 in. above the highest point of crown sheet on boilers 36 in. in diameter and under, nor less than 3 in. on boilers over 36 in. in diameter. These are minimum dimensions, and on large locomotives, and those operating on steep grades, the height should be increased if necessary to compensate for change of water level on descending grades.

The bottom mounting for water glass, and for water column if used, must extend not less than 1½ in. inside the boiler and beyond any obstacle immediately above it, and the passage therein must be straight and horizontal.

Tubular water glasses must be equipped with a protecting shield.

L-73 No water-glass connection shall be fitted with an automatic shut-off valve.

L-74 Every boiler shall have at least one steam gage which will correctly indicate the working pressure. Care must be taken to locate the gage so that it will be kept reasonably cool, and can be conveniently read by the enginemen.

Every gage shall have a siphon of ample capacity to prevent steam entering the gage. The pipe connection shall enter the boiler direct, and shall be maintained steam-tight between boiler and gage. The siphon shall be of brass, copper or bronze composition.

L-75 The dial of the steam gage shall be graduated to not less than 1½ times the maximum allowable working pressure on the boiler.

L-76 Each boiler shall be provided with a valved connection not less than ¼-in. pipe size for attaching a test gage when the boiler is in service, so that the accuracy of the boiler steam gage can be ascertained.

Fittings and Appliances

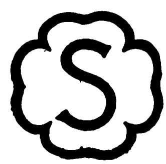
L-77 Locomotive boilers are to be equipped with at least one blow-off cock located at the lowest water space practicable, directly connected to the boiler, either with screw connections or flanged.

L-78 Each boiler shall be equipped with two injectors, or two pumps, or one injector and one pump, with separate delivery pipes connected to checks. The water shall be delivered to the boiler at a point nearer the front than the back flue sheet.

Hydrostatic Tests

L-79 After a boiler has been completed, it shall be subjected to a hydrostatic test of one and one-fourth times the maximum allowable working pressure. The pressure shall be under proper control so that in no case shall the required test pressure be exceeded by more than 6 per cent.

L-80 During a hydrostatic test the safety valve or valves shall be removed, or each valve disk shall be held to its seat by means of a testing clamp, and not by screwing down the compression screw upon the spring.



Official Symbol for Stamp to Denote the American Society of Mechanical Engineers Uniform Standard

L-81 In laying out shell plates, furnace sheets and heads in the boiler shop, care shall be taken to leave at least one of the stamps specified in Par. 36 of Part I, Section I, so located as to be plainly visible when the boiler is completed. Butt straps shall have at least a portion of such stamps visible, sufficient for identification when the boiler is completed.

L-82 Each boiler shall conform in every detail to these Rules, and shall be distinctly stamped with the symbol as shown in Fig. 23, Part I, Section I, denoting that the boiler was constructed in accordance therewith.

After obtaining the stamp to be used when boilers are to be constructed to conform with the A.S.M.E. Boiler Code, a state inspector, municipal inspector, or an inspector employed regularly by an insurance company which is authorized to do a boiler insurance business in the state in which the boiler is built and in the state in which it is to be used, if known, is to be notified that an inspection is to be made and he shall inspect such boilers during construction and after completion. At least two inspections shall be made, one before reaming rivet holes and one at the hydrostatic test. In stamping the boiler after complction, if built in compliance with the Code, the builder shall stamp the boiler in the presence of the inspector, after the hydrostatic test, with the A.S.M.E. Code stamp, the builder's name and the serial number of the manufacturer. A data sheet shall be filled out and signed by the manufacturer and the inspector. This data sheet together with the stamp on the boiler shall denote that it was constructed in accordance with the A.S.M.E. Boiler Code.

S	.... (State in which boiler is to be used)
	.... (Manufacturer's state standard number)
	.... (Name of manufacturer
.... Manufacturer's Serial number	.... (State's number) .... (Year put in service)
	.... (Working pressure when built)

FORM OF STAMPING

Each boiler shall be stamped adjacent to the symbol as shown in Fig. 24, Part I, Section I, with the following items with intervals of about one-half inch between the lines:

1. Manufacturer's serial number
2. State in which boiler is to be used
3. Manufacturer's State standard number
4. Name of manufacturer
5. State's number
6. Year put in service
7. Maximum allowable working pressure when built.

Items 1, 2, 3, 4 and 7 are to be stamped at the shop where boiler is built.
Items 5 and 6 are to be stamped by the proper authority at point of installation.

L-83 The stamps shall be located on the dome.
L-84 Each boiler shall be equipped with a metal badge plate showing the maximum allowable working pressure, which shall be attached to boiler head in the cab. If the boiler head is lagged the lagging and jacketing shall be cut away a sufficient amount to leave the plate visible.

No. 1779

REPORT OF COMMITTEE ON LIMITS AND TOLERANCES IN SCREW THREAD FITS

To the Council of The American Society of Mechanical Engineers:

THE Committee on Limits and Tolerances in Screw Thread Fits was assigned by the Council of The American Society of Mechanical Engineers the task "to prescribe the permissible tolerances in the commercial manufacture of taps, dies, bolts, nuts, and screws, including the method of measuring of the same."

2 During the years following the appointment of this committee, many meetings have been held, and work has been done through sub-committees, involving a great amount of investigation and study.

METHODS OF INVESTIGATION

3 Careful study has been given to data already published and assistance has been secured from the U. S. Bureau of Standards, The Franklin Institute, the Navy Department, and from other sources.

4 A request sent to many tap makers for confidential information showing the limits allowed for their commercial work, led to a response by a number of leading manufacturers, giving such information. This was tabulated and compared. Later, some of the tap makers assisted by having over 4000 taps of commercial sizes from ½ in. to 2 in., secured from a number of different makers, measured for errors in lead, in order to obtain the average variation of commercial taps which are in use today.

5 Early in the investigation, a meeting of screw manufacturers and users was called at the headquarters of The American Society of Mechanical Engineers in New York, at which about forty represen-

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tatives were present, and the matter of tolerances and limits in screws was thoroughly discussed in the light of a tentative report which this committee had prepared. This meeting resulted in the appointment of a sub-committee, consisting of Messrs. E. H. Ehrman of the Chicago Screw Company, E. A. Darling of the Draper Company, and C. B. Young, engineer of tests of the Pennsylvania Railroad Company, to coöperate with the general committee by obtaining data from the screw manufacturers.

6 Through this committee, and the officers of the A.S.M.E., over 5000 screws were obtained from the regular commercial stock of many different manufacturers, these representing work of various grades and sizes, and with cut and rolled threads. These screws were measured and the results tabulated.

7 Sample screws and nuts were prepared having varying degrees of error in diameter and lead, and from these it was determined what would be the maximum error allowable, and charts were made to show the relation of taps and screws measured to these allowable limits.

8 Sample gages were also made to a closer limit than those now proposed by the committee, in order to learn how close it was practicable to make commercial work. These gages were distributed without stating what the allowance was in order that the users might not be prejudiced by thinking the limits were closer than they could work to.

9 Comparisons have also been made with the allowances and tolerances recommended by the British Engineering Standards Committee.

10 The recommendations of the report are the outcome of all this study and investigation.

11 While separate diagrams have not been provided for manufacturers' standards and users' or consumers' standards, it is expected that manufacturers will aim to work within the zones established by the diagrams so as to produce work that will come within these limits; and that gages will be so made as to ensure this result.

12 Allowances to be provided for are wear of tools, unavoidable imperfection of workmanship, taps cutting large, etc.

GAGING SYSTEMS

13 The gaging tools required *for the threaded hole* are:

a Threaded "go" plug of a length equal to the longest engagement of work *b* Threaded "not go" plug, made short and with clearance for full and root diameters

c Plug for bottom of thread;

and *for bolt or screw*:

a Threaded "go" ring of a length equal to the longest engagement of work

b Threaded "not go" ring made short and with clearance for full and root diameters

c Plain ring for outside diameter.

14 The study given to gaging systems has led to the conclusion that no one system is best adapted to all needs; and that for a variety of work made in moderate quantities a gage for measuring errors of diameter and lead combined in the same instrument may give the best results, while for manufacturing in large quantities a fixed gage for one size only and having separate means for measuring errors in diameter and lead may be best.

15 There is also the need in many cases of master gages, inspection gages and workman's gages, each so made as to suit the particular needs.

16 A number of designs of gages for these various purposes have been submitted to 40 prominent manufacturers and users, and following their recommendations selections have been made which are illustrated and described in this report.[1]

17 The illustrations, Figs. 9 to 17, give general suggestions only of what it is recommended to use as it would require too voluminous a report to fully cover the ground.

TABLES FOR LIMITS AND TOLERANCES

18 It is believed that eventually three grades should be established, to cover not only general work, such as is here provided, but also that of more restricted and more liberal tolerances. This report deals with limits and tolerances for general work only.

19 Tables covering medium-grade work for general use have been prepared for diameters for $\frac{1}{4}$ in. to 2 in. but the formulæ can be used for sizes beyond this range. They can also be used for different numbers of threads for a given diameter within ordinary range, provided the thread is of the U. S. S. form.

[1] Reference can also be made to "Report No. 38 on British Standards for Limit Gages for Screw Threads."

20 The variations which affect the fit between screw and nut are those of diameter, lead, including length of engaged thread, and angle of thread, besides others of a minor character, such as the crookedness of tap, the condition of its cutting edge, the kind of metal being tapped, etc. The first three of these variations have been definitely taken into consideration in the tables included in this report, and it is believed that the allowance is sufficient to provide also for the other variations mentioned, unless they are extreme.

21 The effect of errors of lead on the quality of fit is proportional to the length of fit; but the effect of this is modified by the error in pitch diameter. Thus, if a tap, for example, is materially oversize, it can have a greater error of lead than would be the case if it were nearer to the standard size, and still give satisfactory results in use, because the error in lead counteracts the increased diameter of the tapped hole to an extent dependent on the length of fit in the tapped hole.

22 The available variations, when the length of fit is not in excess of one diameter, are shown by the triangular zones of Tables 1 to 18. They are such that any screw having a diameter and error of lead which would come within such zones would enter any tapped hole which would also pass like inspection in the zone established for holes; and the extremes which would pass inspection as to looseness would not be so loose but that they would be considered mechanically satisfactory for general work.

23 The zones in Tables 1 to 18, as has been stated, are based on the engagement between screw and tapped hole with a length of fit equal to one diameter. If the length of fit is greater than one diameter there is a possibility of interference in *extreme cases* such as where a screw having the longest allowable lead is screwed into a hole of a depth greater than one diameter of screw, which has been tapped with a tap having the shortest allowable lead.

24 Under these conditions, however, unless the length considerably exceeds 1½ diameters of the screw, the flow or distortion of metal when forced by the wrench will allow the parts to be screwed together. Actual tests made under the direction of the committee show this to be so. After the first engagement, where the fit might seem unduly tight it would be materially easier, — in fact, for many uses it would be better than a shaky fit, even if within the prescribed limits.

25 For this reason it is believed that taps, nuts and screws passing inspection within these limits, even where it is not known

what length of thread may be required in actual use, can be used with the expectation that the work will be interchangeable even when the length of engagement is greater than one diameter, although theoretically there might be the interference in lead above pointed out. The keeping within the prescribed limits would be a radical improvement over the variations of taps and screws in general use today, because of a common standard serving as a "bull's-eye" at which all would aim.

26 If in any case it should be important to entirely avoid interference in lead, the narrower zones shown by the triangles for the larger-sized screws having a length equal to the length of fit between screw and nut to be used, will show the limiting zone. This method can be used in any case where greater accuracy is desired, and is further explained in Pars. 49 to 51. The plan here submitted is based on having the maximum screws basic in pitch, outside and root diameters.

27 Generally stated, all tapped holes should be above basic standard and all screws, below; the more above or below in pitch diameter the greater the allowance possible in error of lead, while still maintaining a satisfactory fit.

28 In Tables 1 to 18 the figures for taps are held to a limit slightly above the largest allowable screws to provide for wear of the tap, the greatest allowance being made at perfect lead where a reduction in diameter due to wear would be most objectionable. The fact that a tap usually cuts large tends to produce work with whatever variation there is on the safe side. In applying the tables and diagrams to the use of fixed gages, a rectangle representing a given maximum and minimum in pitch diameter and a given error in lead within the triangular zone can be established. Work failing to pass inspection with such gages can be then measured for diameter and lead, and if coming within the triangular zone even though outside the limits of the gages need not be thrown out but can be accepted for use.

CHARTS AND TABLES FOR LIMITS AND TOLERANCES

Shown in Tables 1 to 18, Inclusive

29 When a screw or nut has an error in lead, the amount of that error varies directly as the length of thread on screw or depth of threaded hole, i.e., the longer the screw or the deeper the hole,

the greater the total error in lead; and where a definite quality of fit between a screw and nut is desired, less error in lead per inch can be allowed for a long thread than for a short one.

30 In these tables and charts, therefore, the length of thread is made the governing feature and any chart applies equally well to a screw or nut of any diameter or pitch, for the length of thread specified.

31 The limits for the lengths of threads as adopted and shown by the charts are 0 for the minimum limit and once the nominal diameter of U. S. S. thread for the maximum limit. In this way each chart is especially applicable to a definite size and becomes a standard for that size between the lengths of thread specified.

32 Another factor in the fit between a screw and nut is the pitch diameter measured on the "V" of the thread. For a 60-deg. thread of a *given length* the error in pitch diameter bears a definite relation to the error in lead.

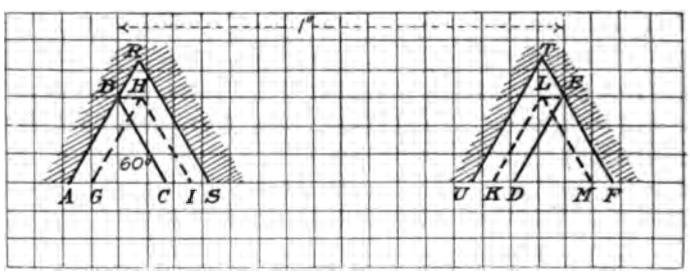


FIG 1 EFFECT OF VARIATION IN LEAD

33 This is illustrated in Fig. 1 where *ABC* and *DEF* represent two threads 1 in. apart on a standard thread gage. Let *GHI* and *KLM* represent two actual spaces cut by a tap of the same pitch diameter with an error in lead equal to *BH* and *LE*. Then the stock *ABHG* and *LMEF* would interfere with the entrance of the threaded gage. But if the tap were increased in radius by the amount *RH*, its lead remaining the same, then it would cut the spaces *ARS* and *UTF* and the thread gage would enter full length and bear along the surfaces *AB* and *FE*.

34 From this it follows that to obtain a fit for a definite length of thread the pitch diameter can be made to compensate for any error in lead within reasonable limits.

35 This relation between the pitch diameter and lead, when *plotted* on the chart, becomes a straight line.

36 In Fig. 2 a tap or screw with perfect lead and pitch diameter would fall at the intersection of the two zero lines at A and, we will assume, would cut a perfect thread for a threaded hole 1 in. deep, i.e., a hole in which a standard threaded plug gage would fit.

37 Another tap having an error in lead of 0.0010 in. per inch but having an increased pitch diameter of 0.00173 in. would fall at

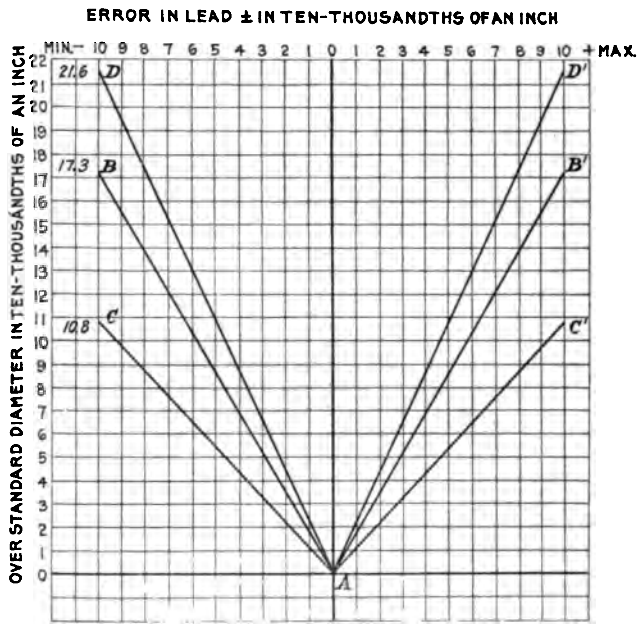


FIG. 2 EFFECT OF VARIATION IN LENGTH OF FIT BETWEEN SCREW AND NUT

B' and for a hole 1 in. deep would be the equivalent of the first tap. If two nuts 1 in. thick were tapped one with each of these taps, a standard threaded plug gage when clear through the nut would fit with equal shake in both nuts.

38 The line AB' passing through zero represents *all* oversize threads 1 in. long with a long or plus lead that are equivalent to a standard or perfect thread. The line AB similarly represents *all* oversize threads 1 in. long with a short or minus lead.[1]

[1] The shorter the length of thread the more nearly horizontal this line becomes, and for a zero length of thread it becomes horizontal, coinciding with the horizontal zero line OA. Similarly the greater the length of thread the more nearly vertical this line becomes.

39 The lines AC and AC' represent taps equivalent to standard for threads ⅝ in. long and AD and AD' for threads 1.25 in. long, the amounts over standard for CC' and DD' being respectively ⅝ and 1¼ times B (17.3) for a lead error of 0.001 in.

40 The oblique lines on the charts, Tables 1 to 18, are similar lines for the maximum length of thread specified in the table alongside of each chart. For screws these lines pass through zero and represent equivalents of perfect screws. For taps and nuts these

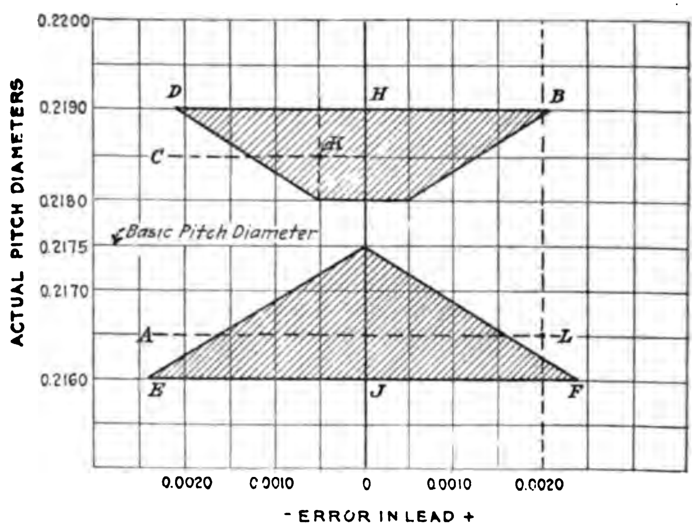


FIG. 3 DIAGRAM ILLUSTRATING METHOD OF USING CHARTS SHOWN IN TABLES 1 TO 18

lines pass through a point slightly above the zero point in order to keep all taps and nuts slightly over standard.

41 In Fig. 3, the points HJ where the upper and lower lines intersect the perfect lead line, represent the extreme limits for the pitch diameters of a nut and screw with perfect lead, and the distance between these two points represents the maximum diametrical shake between any nut and screw falling within the shaded areas or zones of the chart, while the average shake would be about one-half of this maximum shake.

42 It is assumed simply for the purpose of demonstration that a tap makes a hole the exact counterpart of itself, therefore taps and nuts are referred to as having identically the same pitch diameter and lead.

HOW TO USE THE TABLES

43 *For a Tap:*

a Find the actual pitch diameter of the tap with a "V"-thread micrometer. Look in left-hand column at the bottom of the table for taps for the required size (Tables 1 to 18) for this diameter. If the diameter is less than the first figure or greater than the last figure of the column, the tap is not within the limits required.

b When the diameter is found in the table, read to the right in the next column the amount it is *over basic* pitch diameter. Read again to the right in the third column the allowable "errors in lead" for this pitch diameter.

c Next find the actual error in lead per inch with some lead measuring instrument. If the actual error is within the limits given in the third column, then the tap is correct for both pitch diameter and lead.

44 *Example:* For ¼-in., 20-thread tap, for threads from 0 to ¼ in. long, refer to Table 1 and under "For a Tap or Nut." Suppose the pitch diameter of a tap is 0.2185 in. Find this figure in the first column; to the right the next column shows the tap as 0.0010 in. over basic pitch diameter, and the third column shows that its lead must be between 0.0018 in. fine (minus) to 0.0018 in. coarse (plus) per inch to be within the limits of this table.

45 *For a Screw:*

d Find the actual pitch diameter of the screw with a "V"-thread micrometer. Look in the last column at the bottom of the tables for screws for the required size (Tables 1 to 18), for this diameter. If the diameter is less than the first figure or greater than the last figure in the column, the screw is not within the limits required.

e When the diameter is found, read to the left in the next column to the left the amount that it is *under basic* pitch diameter. Read again in the third column to the left the allowable error in lead for this pitch diameter.

f Next find the actual error in lead per inch with some lead measuring instrument. If the actual error is within the limits given in the third column to the left, then the screw is correct for both pitch diameter and lead.

46 *Example:* For ¼-in., 20-thread screw, for threads from 0 to ¼ in. long refer to Table 1, and under "For a Screw or Bolt." Suppose the pitch diameter of a screw is 0.2160 in. Find this figure in the last column; the next column to the left shows the screw as 0.0015 in. under basic pitch diameter, and the third column to the

left shows that its lead must be between 0.0033 in. fine (minus) and 0.0033 in. coarse (plus) to be within the limits of the table.

47 *For Screws and Taps:*

g If the actual pitch diameter for either tap or screw is within the range of the tables but the exact diameter is not given in the proper column, it can be interpolated, or reckoned as between the two nearest values given, and a proportionate and corresponding limit can be likewise interpolated in order to find the error in lead allowable for the actual pitch diameter in hand.

48 *Example:* Same table. Suppose the pitch diameter of a tap is 0.2188 in. In the first colunn for taps it would come between 0.2185 in. and 0.2190 in. and the value for the next column would be $\frac{3}{5}$ of the way between 0.0010 in. and 0.0015 in. or 0.0013 in., and the value for the third column would be $\frac{3}{5}$ of the way between 0.0018 in. and 0.0029 in., or 0.0025 in.

49 If the length of engagement is greater than that provided for in the tables. reference can be made to the table for a larger size having the required length, and the allowable variations in lead thus found.

50 *Example.* Suppose a $\frac{1}{4}$-in. 20 screw is to extend into a tapped hole to a depth of $\frac{1}{2}$ in., or two diameters. Refer to the diagram Table 5 for a hole $\frac{1}{2}$ in. deep, this being for $\frac{1}{2}$ in. 13, which allows for a fit $\frac{1}{2}$ in. long. This will show by the angular lines what variations in lead are allowable, bearing in mind that the variations in pitch diameter must still be kept within the limits given in Table 1 for $\frac{1}{4}$ in.

51 *Example:* Suppose a 1-in. 8 bolt is to be used with a nut $\frac{3}{4}$ in. thick. Refer to the diagram Table 8 which allows for a fit $\frac{3}{4}$ in. long. This allows for a greater variation in lead than the diagram for 1 in., Table 10. The greater limits in pitch diameter allowed for 1 in. can also be used, however.

APPLYING FORMULÆ AND TABLES FOR PITCHES OTHER THAN U. S. S.

52 The diagram, Fig. 4, illustrates the relation of the screw to the tapped hole, showing maximum and minimum allowances.

53 As the formulæ (Par. 61), for external and root diameters are based on the number of threads per inch, the maximum and minimum limits are not changed by a change in diameter and the allowances given in the formulæ for the U. S. Standard diameter and pitch can be used.[1]

54 *Example:* 1 in. diameter, 20 threads to the inch. For external diameter for $\frac{1}{4}$ in. 20 it will be found from the formula (or

[1] These limits can also be obtained by reference to Figs. 7 and 8.

by reference to Fig. 7) that the maximum for screws is basic while the minimum is 0.0060 under. The maximum for tap or nut is +0.0057 in., while the minimum is +0.0019 in., these allowances being always the same for 20 threads to the inch for any diameter. The allowances for pitch diameter and lead, however, are based on formulæ having the diameter as a factor.

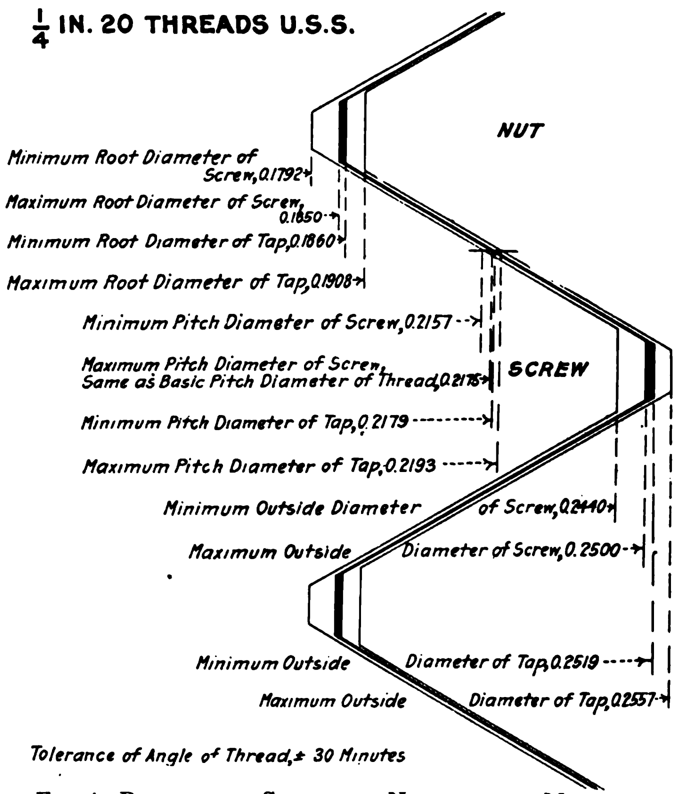


FIG. 4 RELATION OF SCREW AND NUT, SHOWING MAXIMUM AND MINIMUM ALLOWANCES

55 *Example:* 1 in. diameter, 20 threads to the inch. It will be found by the formula (or by reference to Figs. 5 and 6) that the maximum pitch diameter of screw is basic. Minimum pitch diameter $= 1 - (0.0045 \times \sqrt{1} - 0.005) = 1 - 0.004 = 9.996$ min. pitch diam. of screw.

$$\text{Maximum pitch diameter of tap} = 1 + 0.004 = 1.004 \text{ in.}$$

$$\text{Minimum pitch diameter of tap} = 1 + \frac{0.004}{4} = 1.001 \text{ in.}$$

Maximum error (± normal) allowable in lead of screws if the length of engagement = 1 diameter (in this example 1 in.) =

$$\frac{0.57735\ (0.0045 \times \sqrt{1} - 0.005)}{1} = 0.0023 \text{ max. error in lead.}$$

56 If, however, the length of engagement is only ½ in. instead of 1 in., a proportionately greater error in lead can be allowed.

$0.0023 \div \frac{1}{2} = 0.0046$ in. maximum error in lead for 1 in. 20 threads with ½ in. length of engagement.

Maximum error in lead for tap for 1 in. length of engagement

$$0.0023 - \frac{0.0023^2}{0.03} = 0.0017 \text{ max. error in lead.}$$

For ½ length of engagement, $0.0017 \div \frac{1}{2} = 0.0034$.

TO FIND POSITION OF A TAP OR SCREW IN THE CHART, FIG. 3

57 *First*, find its actual pitch or angular diameter as measured with a "V"-thread micrometer. Subtract from this diameter the basic pitch diameter as given in the table. The difference will be its deviation from basic pitch diameter. If plus, lay off its value to scale above the horizontal coördinate line; and if minus, below the horizontal line.

58 *Second*, find the "error in lead per inch" with a lead measuring instrument. If the lead is coarse, equal plus, lay off its value to scale to the right of the vertical coördinate line, or if fine, equal minus, lay it off to the left of the vertical line.

59 The point where these first and second values intersect is the position of the tap or screw on the chart. If it is within the shaded area, the tap or screw is within the prescribed limits, and if outside of the shaded area it is not up to the standard represented by the chart and table. Taps and nuts must fall within the shaded area above the horizontal line; while screws and bolts must fall within the shaded area below the horizontal line.

60 *Example for a tap:* Suppose a tap measured 0.001 in. over basic pitch diameter; this value would fall on the horizontal line CK (Fig. 3), and if its lead should be 0.0005 in. fine or minus in 1 in., this value would fall on the vertical dotted line passing through K, and the tap falls within the area of the chart.

61 *Example for a Screw:* Suppose a screw measured 0.0010 in. under basic pitch diameter, it would then fall on the horizontal line AL (Fig. 3), and if its lead were 0.0020 in. coarse or plus it would fall on the vertical dotted line passing through L, and the screw falls outside the area of the chart.

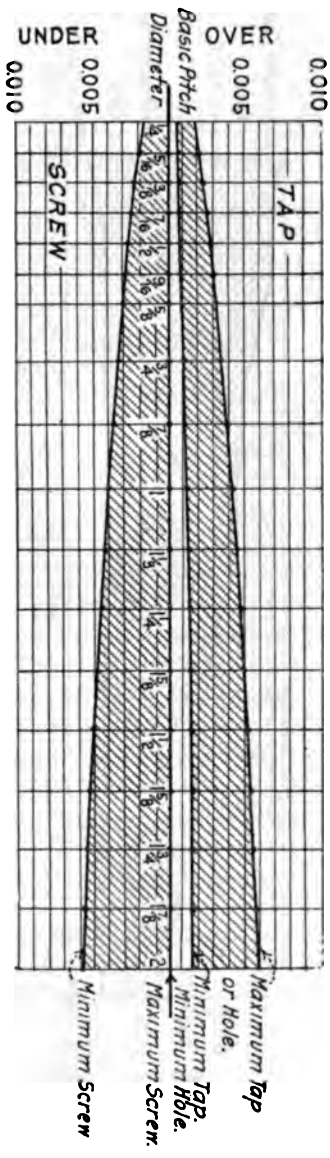


FIG. 5 MAXIMUM AND MINIMUM CLEARANCE, PITCH DIAMETER

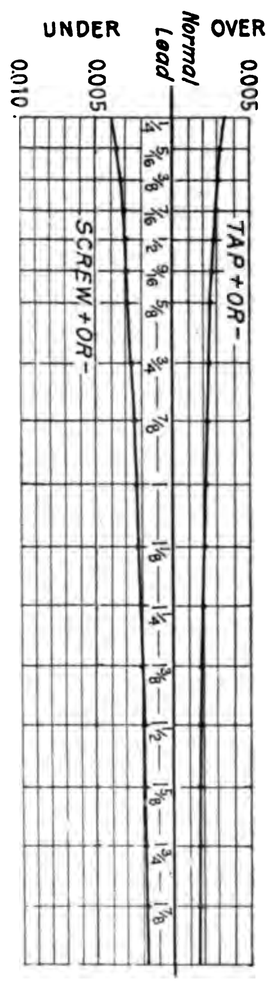


FIG. 6 ALLOWABLE VARIATION IN LEAD OVER OR UNDER NORMAL FOR MAXIMUM ERROR IN DIAMETER

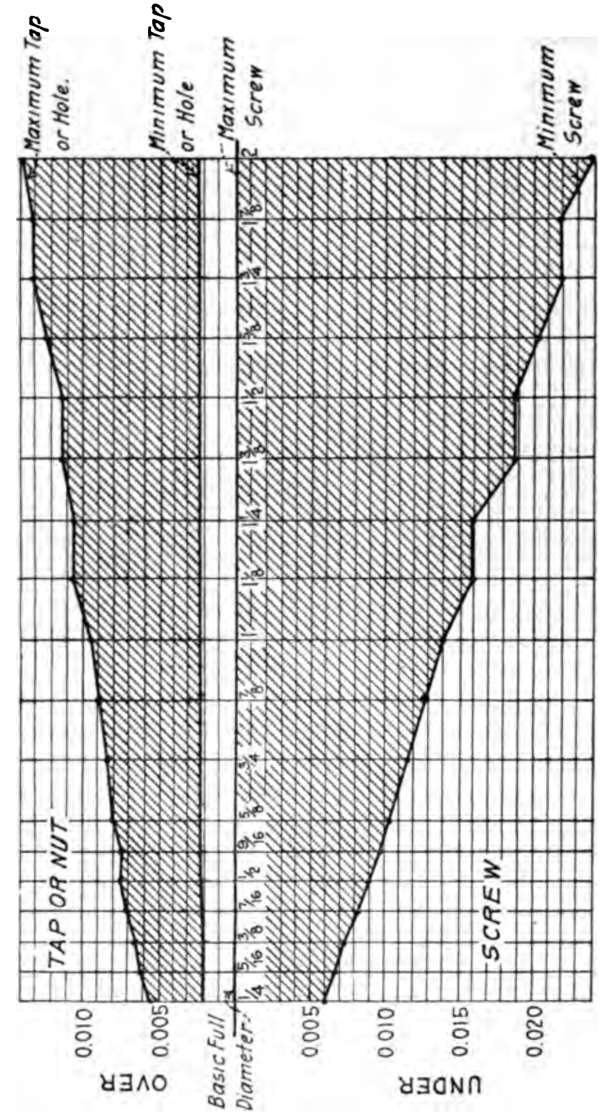


FIG. 7 MAXIMUM AND MINIMUM CLEARANCE, FULL OR EXTERNAL DIAMETER

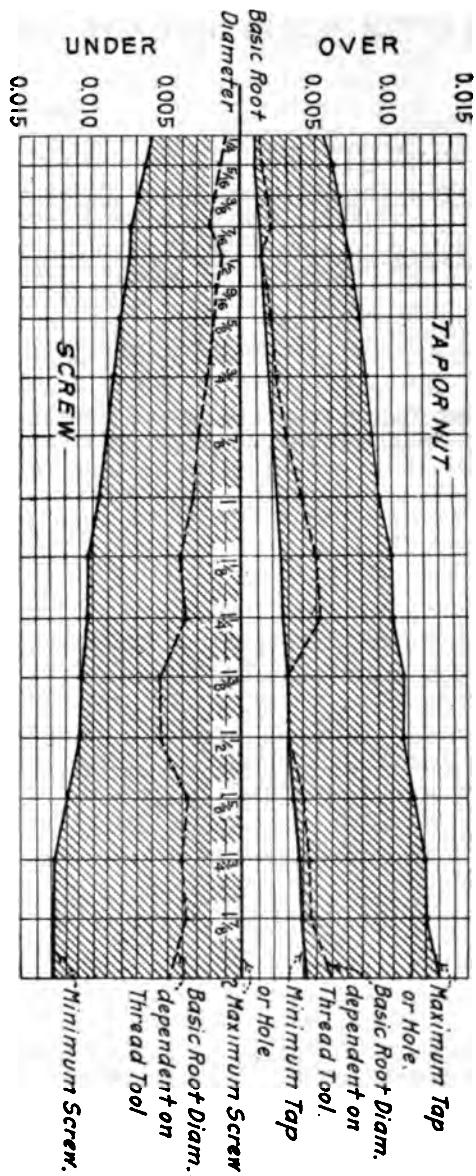


FIG. 8 MAXIMUM AND MINIMUM CLEARANCE, ROOT DIAMETER

TABLE 1

¼ (0.250) INCH 20 THREADS U.S.S.
BASIC PITCH DIAMETER 0.2175
Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 18 PITCH TOOL = 0.0069		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 22 PITCH TOOL = 0.0057	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.2519	0.2557	0.2500	0.2500	0.2440
ROOT "	0.1867	0.1877	0.1850	0.1850	0.1792

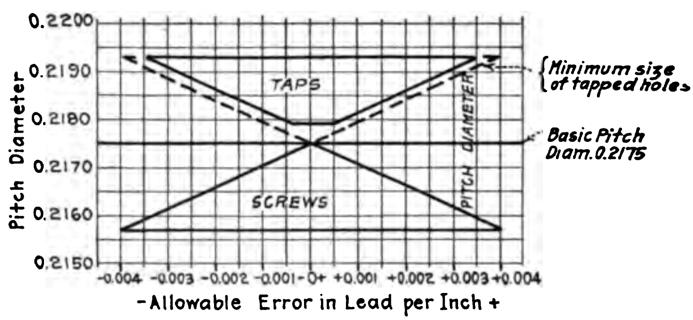
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.2193	MAX. 0.2193	0.0035	0.0035	0.2157 MIN.
MIN. 0.2179	MIN. 0.2175	0.0004	0.0000	0.2175 MAX.

Tolerance for Thread Angle ± 30 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT 0 TO ¼ INCH (ONCE THE DIAM.)

NOTE: In Practice these can be recommended for length of thread engagement to one and one half diam's (⅜") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE READ FROM LEFT TO RIGHT			
ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.2179	0.0004	± 0.0005	±0.0009
0.2180	0.0005	± 0.0007	±0.0011
0.2185	0.0010	± 0.0018	±0.0022
0.2190	0.0015	± 0.0029	±0.0033
0.2193	0.0018	± 0.0035	±0.0040

FOR A SCREW OR BOLT READ FROM RIGHT TO LEFT		
ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
0.0000	0.0000	0.2175
±0.0012	0.0005	0.2170
±0.0022	0.0010	0.2165
±0.0033	0.0015	0.2160
±0.0040	0.0018	0.2157

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TABLE 2

$\frac{5}{16}$ (0.3125) INCH 18 THREADS U.S.S.

BASIC PITCH DIAMETER 0.2764

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 16 PITCH TOOL=0.0078		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 20 PITCH TOOL=0.0062	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.3144	0.3186	0.3125	0.3125	0.3059
ROOT "	0.2414	0.2464	0.2403	0.2403	0.2342

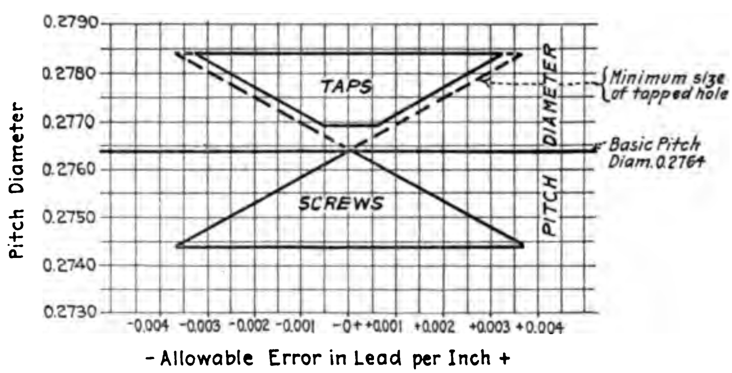
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.2784	MAX. 0.2784	0.0040	0.0040	0.2744 MIN.
MIN. 0.2769	MIN. 0.2764	0.0005	0.0000	0.2764 MAX.

Tolerance for Thread Angle ± 30 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{5}{16}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($\frac{15}{32}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE READ FROM LEFT TO RIGHT			
ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.2769	0.0005	± 0.0006	± 0.0010
0.2770	0.0006	± 0.0008	± 0.0011
0.2775	0.0011	± 0.0016	± 0.0021
0.2780	0.0016	± 0.0025	± 0.0030
0.2784	0.0020	± 0.0033	± 0.0037

FOR A SCREW OR BOLT READ FROM RIGHT TO LEFT		
ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
± 0.0000	0.0000	0.2764
± 0.0008	0.0004	0.2760
± 0.0017	0.0009	0.2755
± 0.0027	0.0014	0.2750
± 0.0037	0.0019	0.2744

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TABLE 3

$\frac{3}{8}$ (0.375) INCH 16 THREADS U.S.S.

BASIC PITCH DIAMETER 0.3344

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 14 PITCH TOOL=0.0089		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 18 PITCH TOOL=0.0069	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.3770	0.3815	0.3750	0.3750	0.3677
ROOT "	0.2949	0.3001	0.2936	0.2936	0.2871

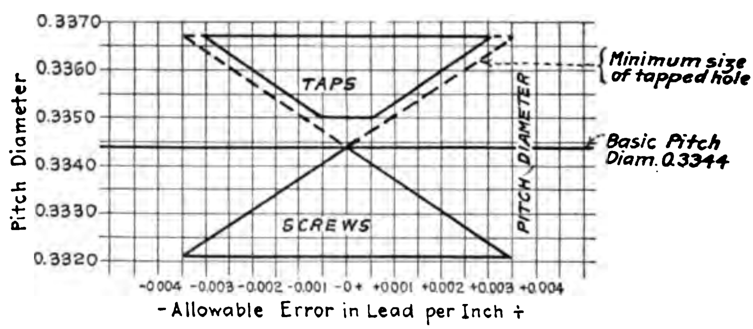
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.3367	MAX. 0.3367	0.0046	0.0046	0.3321 MIN.
MIN. 0.3347	MIN. 0.3344	0.0003	0.0003	0.3344 MAX.

Tolerance for Thread Angle ± 30 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{3}{8}$" INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($\frac{9}{16}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.3350	0.0006	± 0.0005	± 0.0009
0.3355	0.0011	± 0.0013	± 0.0017
0.3360	0.0016	± 0.0020	± 0.0024
0.3365	0.0021	± 0.0028	± 0.0032
0.3367	0.0023	± 0.0031	± 0.0035

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM
± 0.0000	0.0000	0.3344
± 0.0006	0.0004	0.3340
± 0.0014	0.0009	0.3335
± 0.0021	0.0014	0.3330
± 0.0029	0.0019	0.3325
± 0.0035	0.0023	0.3321

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TABLE 4

$\frac{7}{16}$ (0.4375) INCH 14 THREADS U.S.S.

BASIC PITCH DIAMETER 0.3911

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 13 PITCH TOOL=0.0096		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 16 PITCH TOOL=0.0078	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.4396	0.4444	0.4375	0.4375	0.4292
ROOT "	0.3461	0.3516	0.3447	0.3447	0.3378

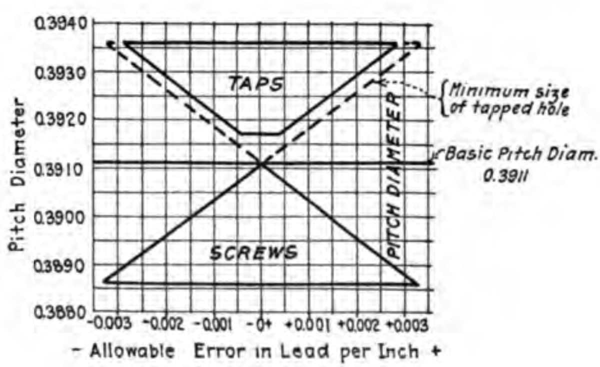
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.3936	MAX. 0.3936	0.0050	0.0050	0.3886 MIN.
MIN. 0.3917	MIN. 0.3911	0.0006	0.0000	0.3911 MAX.

Tolerance for Thread Angle ± 30 Minutes.

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{7}{16}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($\frac{21''}{32}$) because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE READ FROM LEFT TO RIGHT			
ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.3917	0.0006	±0.0004	±0.0008
0.3920	0.0009	±0.0008	±0.0012
0.3925	0.0014	±0.0015	±0.0019
0.3930	0.0019	±0.0021	±0.0025
0.3935	0.0024	±0.0028	±0.0032
0.3936	0.0025	±0.0033	±0.0038

FOR A SCREW OR BOLT READ FROM RIGHT TO LEFT		
ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	0.3911
±0.0008	0.0006	0.3905
±0.0015	0.0011	0.3900
±0.0021	0.0016	0.3895
±0.0028	0.0021	0.3890
±0.0033	0.0025	0.3886

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TABLE 5

½ (0.500) INCH 13 THREADS U.S.S.
BASIC PITCH DIAMETER 0.4501
Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 12 PITCH TOOL = 0.0104		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 14 PITCH TOOL = 0.0089	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.5021	0.5073	0.5000	0.5000	0.4911
ROOT "	0.4016	0.4074	0.4001	0.4001	0.3928

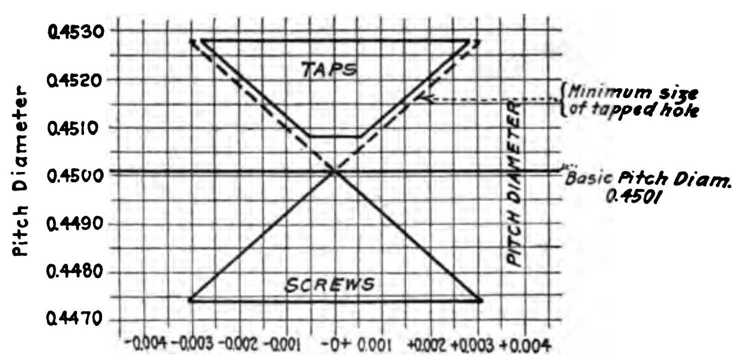
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX 0.4528	MAX. 0.4528	0.0054	0.0054	0.4474 MIN.
MIN. 0.4508	MIN. 0.4501	0.0007	0.0000	0.4501 MAX.

Tolerance for Thread Angle ± 30 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO ½ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's (¾") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE READ FROM LEFT TO RIGHT			
ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.4508	0.0007	±0.0006	±0.0008
0.4510	0.0009	±0.0009	±0.0010
0.4515	0.0014	±0.0013	±0.0016
0.4520	0.0019	±0.0020	±0.0022
0.4525	0.0024	±0.0025	±0.0028
0.4528	0.0027	±0.0028	±0.0031

FOR A SCREW OR BOLT READ FROM RIGHT TO LEFT		
ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	0.4501
±0.0007	0.0006	0.4495
±0.0013	0.0011	0.4490
±0.0018	0.0016	0.4485
±0.0024	0.0021	0.4480
±0.0030	0.0026	0.4475
±0.0031	0.0027	0.4474

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TABLE 6

$\frac{9}{16}$ (0.5625) INCH 12 THREADS U.S.S.

BASIC PITCH DIAMETER 0.5084

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 11 PITCH TOOL=0.0114		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 13 PITCH TOOL=0.0096	
	MIN.	MAX.		MAX	MIN.
OUTSIDE DIAMETER	0.5646	0.5701	0.5625	0.5625	0.5580
ROOT "	0.4559	0.4618	0.4542	0.4542	0.4466

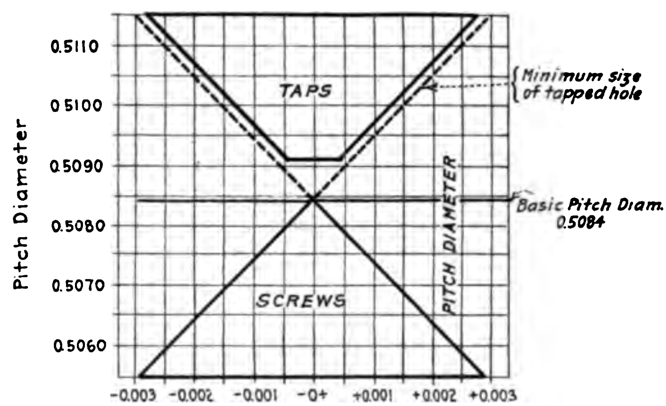
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.5112	MAX. 0.5112	0.0056	0.0056	0.5056 MIN.
MIN. 0.5091	MIN. 0.5084	0.0007	0.0000	0.5084 MAX.

Tolerance for Thread Angle ± 30 Minutes.

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{9}{16}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($\frac{27}{32}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLES
0.5091	0.0007	±0.0005	±0.0007
0.5095	0.0011	±0.0009	±0.0011
0.5100	0.0016	±0.0014	±0.0016
0.5105	0.0021	±0.0019	±0.0022
0.5110	0.0026	±0.0025	±0.0027
0.5113	0.0033	±0.0027	±0.0029

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	0.5084
±0.0004	0.0004	0.5080
±0.0010	0.0009	0.5075
±0.0015	0.0014	0.5070
±0.0020	0.0019	0.5065
±0.0025	0.0024	0.5060
±0.0029	0.0029	0.5055

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TABLE 7

$\frac{5}{8}$ (0.625) INCH 11 THREADS U.S.S.

BASIC PITCH DIAMETER 0.5660

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 10 PITCH TOOL=0.0125		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 12 PITCH TOOL=0.0104	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.6272	0.6330	0.6250	0.6250	0.6147
ROOT "	0.5087	0.5148	0.5069	0.5069	0.4990

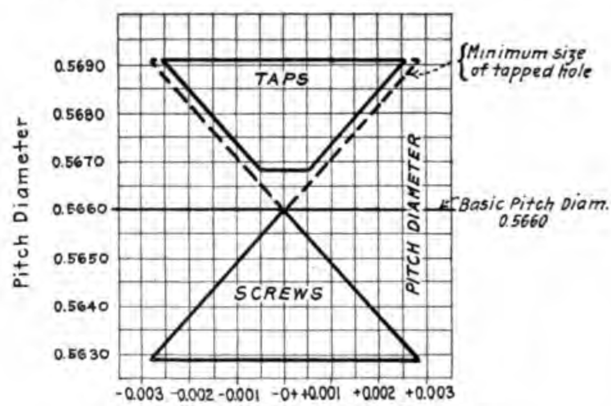
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.5691	MAX. 0.5691	0.0062	0.0062	0.5629 MIN.
MIN. 0.5668	MIN. 0.5660	0.0008	0.0000	0.5660 MAX.

Tolerance for Thread Angle ± 15 Minutes.

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{5}{8}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($\frac{15}{16}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



− Allowable Error in Lead per Inch +

ALLOWABLE ERROR IN LEAD PER INCH
FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.5668	0.0008	±0.0006	±0.0008
0.5670	0.0010	±0.0008	±0.0009
0.5675	0.0015	±0.0012	±0.0014
0.5680	0.0020	±0.0016	±0.0018
0.5685	0.0025	±0.0021	±0.0023
0.5690	0.0030	±0.0025	±0.0027
0.5691	0.0031	±0.0026	±0.0028

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
0.0000	0.0000	0.5660
±0.0005	0.0005	0.5655
±0.0009	0.0010	0.5650
±0.0014	0.0015	0.5645
±0.0018	0.0020	0.5640
±0.0023	0.0025	0.5635
±0.0027	0.0030	0.5630
±0.0028	0.0031	0.5629

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TABLE 8

$\frac{3}{4}$ (0.750) INCH 10 THREADS U.S.S.

BASIC PITCH DIAMETER 0.6851

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 9 PITCH TOOL=0.0139		U.S.S BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 11 PITCH TOOL=0.0114	
	MIN.	MAX		MAX	MIN.
OUTSIDE DIAMETER	0.7522	0.7585	0.7500	0.7500	0.7387
ROOT "	0.6221	0.6284	0.6201	0.6201	0.6118

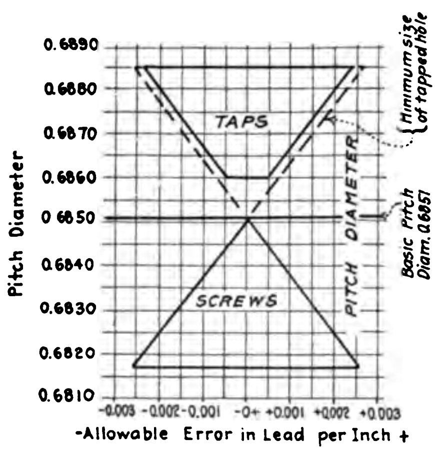
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX 0.6885	MAX. 0.6885	0.0068	0.0068	0.6817 MIN
MIN. 0.6860	MIN. 0.6851	0.0009	0.0000	0.6851 MAX

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{3}{4}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($1\frac{1}{8}''$) because of the partial rectifying of errors in lead by "flow of metal" see Pars 24 and 25



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.6860	0.0009	±0.0005	±0.0007
0.6865	0.0014	±0.0009	±0.0011
0.6870	0.0019	±0.0012	±0.0015
0.6875	0.0024	±0.0016	±0.0018
0.6880	0.0029	±0.0020	±0.0022
0.6885	0.0034	±0.0024	±0.0026

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
0.0000	0.0000	0.6851
± 0.0005	0.0006	0.6845
± 0.0009	0.0011	0.6840
± 0.0012	0.0016	0.6835
± 0.0016	0.0021	0.6830
± 0.0020	0.0026	0.6825
± 0.0024	0.0031	0.6820
± 0.0026	0.0034	0.6817

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TABLE 9

$\frac{7}{8}$ (0.875) INCH 9 THREADS U.S.S.
BASIC PITCH DIAMETER 0.8029
Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 8 PITCH TOOL=0.0156		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 10 PITCH TOOL=0.0125	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	0.8773	0.8840	0.8750	0.8750	0.8625
ROOT "	0.7329	0.7395	0.7307	0.7307	0.7219

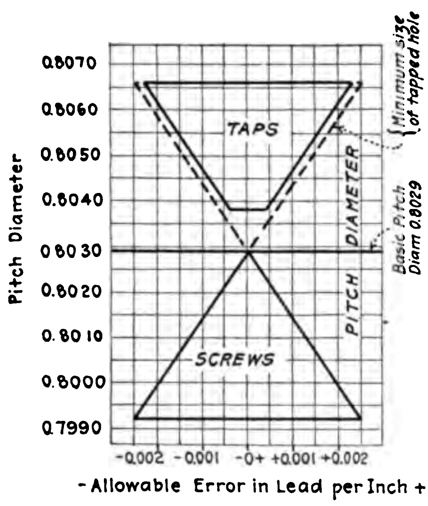
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.8066	MAX. 0.8066	0.0054	0.0054	0.7992 MIN.
MIN. 0.8038	MIN. 0.8029	0.0009	0.0000	0.8029 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $\frac{7}{8}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($1\frac{5}{16}$) because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.8038	0.0009	±0.0004	±0.0006
0.8040	0.0011	±0.0005	±0.0008
0.8045	0.0016	±0.0009	±0.0011
0.8050	0.0021	±0.0012	±0.0015
0.8055	0.0026	±0.0016	±0.0018
0.8060	0.0031	±0.0019	±0.0021
0.8065	0.0036	±0.0022	±0.0024
0.8066	0.0037	±0.0023	±0.0025

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	0.8029
±0.0003	0.0004	0.8025
±0.0006	0.0009	0.8020
±0.0010	0.0014	0.8015
±0.0013	0.0019	0.8010
±0.0016	0.0024	0.8005
±0.0020	0.0029	0.8000
±0.0023	0.0034	0.7995
±0.0025	0.0037	0.7992

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TABLE 10

1 INCH 8 THREADS U.S.S.

BASIC PITCH DIAMETER 0.9188

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 7 PITCH TOOL=0.0179		U.S.S BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 9 PITCH TOOL=0.0139	
	MIN	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.0023	1.0092	1.0000	1.0000	0.9861
ROOT "	0.8401	0.8469	0.8376	0.8376	0.8283

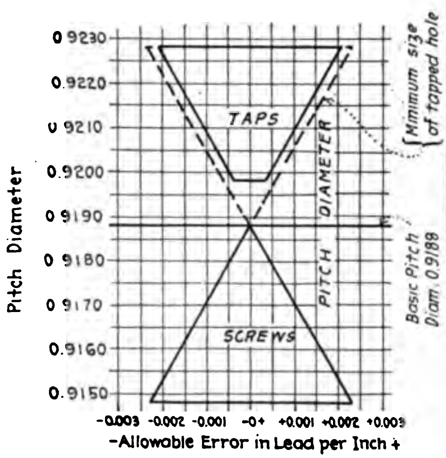
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 0.9228	MAX. 0.9228	0.0080	0.0080	0.9148 MIN.
MIN. 0.9198	MIN. 0.9188	0.0010	0.0000	0.9188 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO 1 INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's (1½") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
0.9198	0.0010	±0.0004	±0.0006
0.9200	0.0012	±0.0005	±0.0007
0.9205	0.0017	±0.0008	±0.0010
0.9210	0.0022	±0.0011	±0.0013
0.9215	0.0027	±0.0014	±0.0016
0.9220	0.0032	±0.0017	±0.0019
0.9225	0.0037	±0.0019	±0.0022
0.9228	0.0040	±0.0021	±0.0023

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
0.0000	0.0000	0.9188
±0.0002	0.0003	0.9185
±0.0005	0.0008	0.9180
±0.0008	0.0013	0.9175
±0.0010	0.0018	0.9170
±0.0013	0.0023	0.9165
±0.0016	0.0028	0.9160
±0.0019	0.0033	0.9155
±0.0022	0.0038	0.9150
±0.0023	0.0040	0.9148

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TABLE 11

$1\frac{1}{8}$ (1.125) INCH 7 THREADS U.S.S.

BASIC PITCH DIAMETER 1.0322

Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 6 PITCH TOOL=0.0208		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 8 PITCH TOOL=0.0156	
	MIN	MAX		MAX.	MIN.
OUTSIDE DIAMETER	1.1274	1.1347	1.1250	1.1250	1.1083
ROOT "	0.9422	0.9494	0.9394	0.9394	0.9294

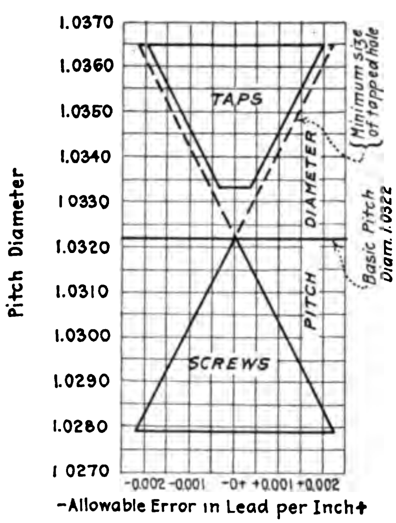
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.0365	MAX. 1.0365	0.0086	0.0086	1.0279 MIN.
MIN. 1.0333	MIN. 1.0322	0.0011	0.0000	1.0322 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $1\frac{1}{8}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engage-ment to one and one half diam's ($1\frac{11}{16}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.0333	0.0011	±0.0004	±0.0006
1.0335	0.0013	±0.0005	±0.0007
1.0340	0.0018	±0.0007	±0.0009
1.0345	0.0023	±0.0010	±0.0012
1.0350	0.0028	±0.0012	±0.0014
1.0355	0.0033	±0.0015	±0.0017
1.0360	0.0038	±0.0018	±0.0020
1.0365	0.0043	±0.0020	±0.0022

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM
±0.0000	0.0000	1.0322
±0.0004	0.0007	1.0315
±0.0006	0.0012	1.0310
±0.0009	0.0017	1.0305
±0.0011	0.0022	1.0300
±0.0014	0.0027	1.0295
±0.0016	0.0032	1.0290
±0.0019	0.0037	1.0285
±0.0021	0.0042	1.0280
±0.0022	0.0043	1.0279

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TABLE 12

1 1/4 (1.250) INCH 7 THREADS U.S.S.
BASIC PITCH DIAMETER 1.1572
Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 6 PITCH TOOL=0.0208		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 8 PITCH TOOL=0.0156	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.2524	1.2605	1.2500	1.2500	1.2343
ROOT "	1.0673	1.0744	1.0644	1.0644	1.0544

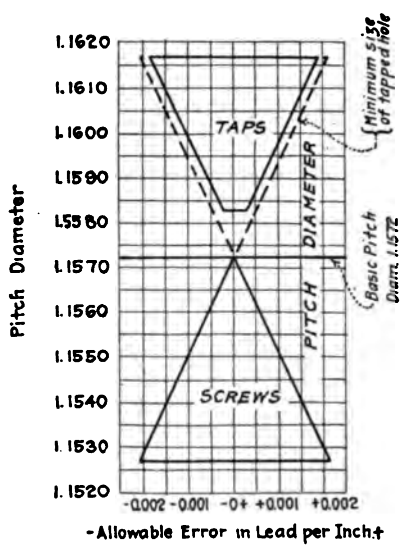
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.1617	MAX. 1.1617	0.0090	0.0090	1.1527 MIN.
MIN. 1.1583	MIN. 1.1572	0.0011	0.0000	1.1572 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO 1 1/4 INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's (1 7/8) because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.1583	0.0011	±0.0003	±0.0005
1.1585	0.0013	±0.0004	±0.0006
1.1590	0.0018	±0.0006	±0.0008
1.1595	0.0023	±0.0009	±0.0011
1.1600	0.0028	±0.0011	±0.0013
1.1605	0.0033	±0.0013	±0.0015
1.1610	0.0038	±0.0016	±0.0018
1.1615	0.0043	±0.0018	±0.0020
1.1617	0.0045	±0.0019	±0.0021

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM
±0.0000	0.0000	1.1572
±0.0003	0.0007	1.1565
±0.0006	0.0012	1.1560
±0.0008	0.0017	1.1555
±0.0010	0.0022	1.1550
±0.0013	0.0027	1.1545
±0.0015	0.0032	1.1540
±0.0017	0.0037	1.1535
±0.0020	0.0042	1.1530
±0.0021	0.0045	1.1527

TABLE 13

$1\frac{3}{8}$ (1.375) INCH 6 THREADS U.S.S.

BASIC PITCH DIAMETER 1.2668

Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS $5\frac{1}{2}$ PITCH TOOL = 0.0227		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 7 PITCH TOOL = 0.0179	
	MIN.	MAX.		MAX	MIN.
OUTSIDE DIAMETER	1.3777	1.3865	1.3750	1.3750	1.3568
ROOT "	1.1618	1.1695	1.1585	1.1585	1.1475

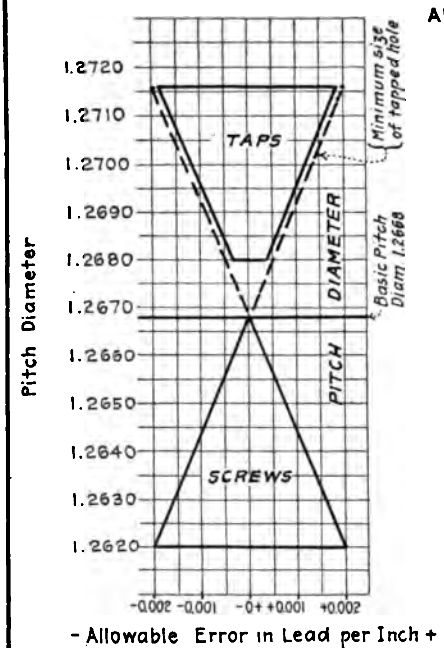
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.2716	MAX. 1.2716	0.0096	0.0096	1.2620 MIN.
MIN. 1.2680	MIN. 1.2688	0.0012	0.0000	1.2668 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT 0 TO $1\frac{3}{8}$ INCH (ONCE THE DIAM.)

NOTE: In Practice these can be recommended for length of thread engagement to one and one half diam's ($2\frac{1}{16}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERRORS IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR A TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.2680	0.0012	±0.0004	±0.0005
1.2685	0.0017	±0.0006	±0.0007
1.2690	0.0022	±0.0008	±0.0009
1.2695	0.0027	±0.0010	±0.0011
1.2700	0.0032	±0.0012	±0.0014
1.2705	0.0037	±0.0014	±0.0016
1.2710	0.0042	±0.0016	±0.0018
1.2715	0.0047	±0.0018	±0.0020
1.2716	0.0048	±0.0019	±0.0020

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	1.2668
±0.0003	0.0008	1.2660
±0.0005	0.0013	1.2655
±0.0007	0.0018	1.2650
±0.0010	0.0023	1.2645
±0.0012	0.0028	1.2640
±0.0014	0.0033	1.2635
±0.0016	0.0038	1.2630
±0.0018	0.0043	1.2625
±0.0020	0.0048	1.2620

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TABLE 14

1½ (1.500) INCH 6 THREADS U.S.S.
BASIC PITCH DIAMETER 1.3918
Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 5½ PITCH TOOL=0.0227		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 7 PITCH TOOL=0.0179	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.5024	1.5115	1.5000	1.5000	1.4818
ROOT "	1.2868	1.2944	1.2835	1.2835	1.2726

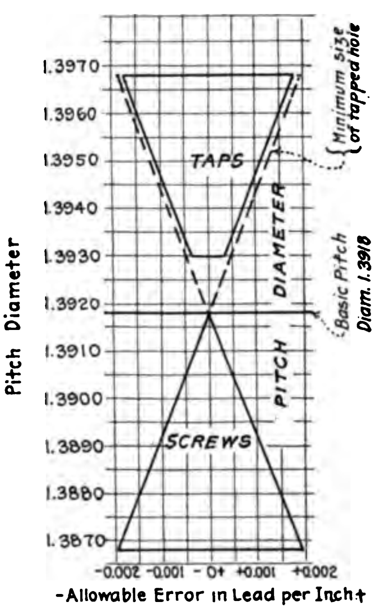
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.3968	MAX. 1.3968	0.0100	0.0100	1.3868 MIN.
MIN. 1.3931	MIN. 1.3918	0.0013	0.0000	1.3918 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO 1½ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's (2¼") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.3930	0.0012	±0.0004	±0.0005
1.3940	0.0022	±0.0008	±0.0009
1.3950	0.0032	±0.0011	±0.0012
1.3960	0.0040	±0.0015	±0.0016
1.3968	0.0050	±0.0018	±0.0019

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM
±0.0000	0.0000	1.3918
±0.0003	0.0008	1.3910
±0.0007	0.0018	1.3900
±0.0011	0.0028	1.3890
±0.0014	0.0038	1.3880
±0.0018	0.0048	1.3870
±0.0019	0.0050	1.3868

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TABLE 15

$1\frac{5}{8}$ (1.625) INCH $5\frac{1}{2}$ PITCH U.S.S.

BASIC PITCH DIAMETER 1.5069

Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 5 PITCH TOOL=0.0250		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 6 PITCH TOOL=0.0208	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.6275	1.6375	1.6250	1.6250	1.6053
ROOT "	1.3924	1.4003	1.3888	1.3888	1.3773

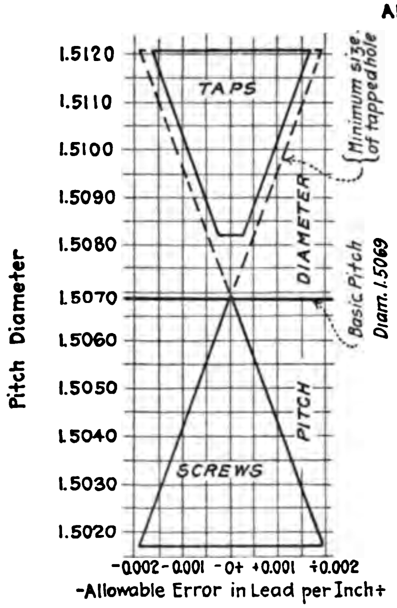
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.5121	MAX. 1.5121	0.0104	0.0104	1.5017 MIN.
MIN. 1.5082	MIN. 1.5069	0.0013	0.0000	1.5069 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $1\frac{5}{8}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($2\frac{7}{16}$) because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.5082	0.0013	±0.0003	±0.0005
1.5090	0.0021	±0.0006	±0.0008
1.5100	0.0031	±0.0009	±0.0011
1.5110	0.0041	±0.0013	±0.0015
1.5120	0.0051	±0.0016	±0.0019
1.5121	0.0052	±0.0017	±0.0019

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	1.5069
±0.0004	0.0009	1.5060
±0.0007	0.0019	1.5050
±0.0011	0.0029	1.5040
±0.0014	0.0039	1.5030
±0.0018	0.0049	1.5020
±0.0019	0.0052	1.5017

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TABLE 16

$1\frac{3}{4}$ (1.750) INCH 5 THREADS U.S.S.

BASIC PITCH DIAMETER 1.6201

Outside and Root Measurements.

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS $4\frac{1}{2}$ PITCH TOOL = 0.0278		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS $5\frac{1}{2}$ PITCH TOOL = 0.0227	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.7525	1.7630	1.7500	1.7500	1.7284
ROOT "	1.4942	1.5024	1.4902	1.4902	1.4780

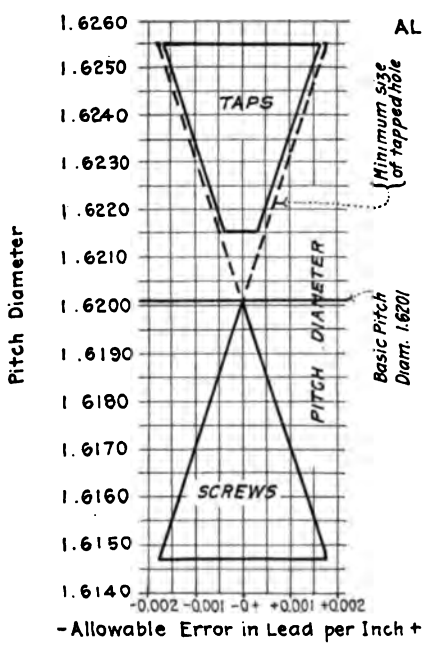
LIMITS ON PITCH DIAMETER AT PERFECT LEAD.

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.6256	MAX. 1.6256	0.0109	0.0109	1.6147 MIN.
MIN. 1.6215	MIN. 1.6201	0.0014	0.0000	1.6201 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $1\frac{3}{4}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($2\frac{5}{8}$") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.6215	0.0014	±0.0004	±0.0005
1.6225	0.0024	±0.0007	±0.0008
1.6235	0.0034	±0.0010	±0.0011
1.6245	0.0044	±0.0014	±0.0015
1.6255	0.0054	±0.0017	±0.0018

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
0.0000	0.0000	1.6201
±0.0002	0.0006	1.6195
±0.0005	0.0016	1.6185
±0 0009	0.0026	1.6175
±0.0012	0.0036	1.6165
±0.0016	0 0046	1.6155
±0.0018	0.0054	1.6147

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TABLE 17

$1\frac{7}{8}$ (1.875) INCH 5 PITCH U.S.S.

BASIC PITCH DIAMETER 1.7451

Outside and Root Measurements

	FOR A TAP OR NUT FLAT AT BOTTOM OF THREAD SAME AS $4\frac{1}{2}$ PITCH TOOL=0.0278		U.S.S. BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS $5\frac{1}{2}$ PITCH TOOL=0.0227	
	MIN.	MAX.		MAX.	MIN.
OUTSIDE DIAMETER	1.8775	1.8880	1.8750	1.8750	1.8534
ROOT "	1.6192	1.6273	1.6152	1.6152	1.6031

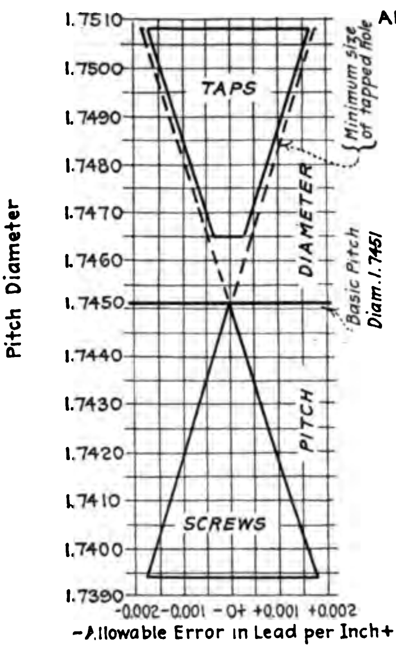
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.7508	MAX. 1.7508	0.0114	0.0114	1.7394 MIN.
MIN. 1.7465	MIN. 1.7451	0.0014	0.0000	1.7451 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO $1\frac{7}{8}$ INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's ($2\frac{13}{16}$) because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.7465	0.0014	±0.0003	±0.0004
1.7470	0.0019	±0.0005	±0.0006
1.7480	0.0029	±0.0008	±0.0009
1.7490	0.0039	±0.0011	±0.0012
1.7500	0.0049	±0.0014	±0.0016
1.7508	0.0057	±0.0017	±0.0018

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	1.7451
±0.0003	0.0011	1.7440
±0.0006	0.0021	1.7430
±0.0010	0.0031	1.7420
±0.0013	0.0041	1.7410
±0.0016	0.0051	1.7400
±0.0018	0.0057	1.7394

TABLE 18

2 INCH $4\frac{1}{2}$ THREADS U.S.S.

BASIC PITCH DIAMETER 1.8557

Outside and Root Measurements

	FOR A TAP OR NUT FLAT ON BOTTOM OF THREAD SAME AS 4 PITCH TOOL=0.0313		U.S.S BASIC SIZES	FOR A SCREW OR BOLT FLAT AT ROOT OF THREAD SAME AS 5 PITCH TOOL=0.0250	
	MIN.	MAX		MAX	MIN.
OUTSIDE DIAMETER	2.0025	2.0139	2.0000	2.0000	1.9761
ROOT "	1.7158	1.7243	1.7113	1.7113	1.6984

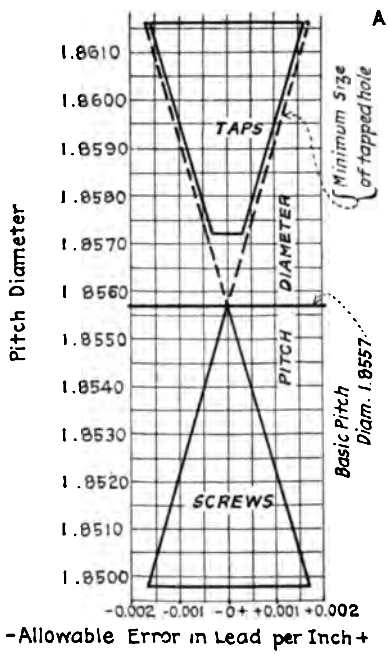
LIMITS ON PITCH DIAMETER AT PERFECT LEAD

TAP	TAPPED HOLE	CLEARANCE		SCREW OR BOLT
		TAP	HOLE	
MAX. 1.8616	MAX. 1.8616	0.0118	0.0118	1.8498 MIN.
MIN. 1.8572	MIN. 1.8557	0.0015	0.0000	1.8557 MAX.

Tolerance for Thread Angle ± 15 Minutes

CHART AND TABLE FOR TAPS, NUTS, SCREWS ETC. WITH LENGTH OF THREAD ENGAGEMENT, 0 TO 2 INCH (ONCE THE DIAM.)

NOTE: In practice these can be recommended for length of thread engagement to one and one half diam's (3") because of the partial rectifying of errors in lead by "flow of metal" see Pars. 24 and 25.



ALLOWABLE ERROR IN LEAD PER INCH FOR ACTUAL PITCH DIAMETER

FOR A TAP OR TAPPED HOLE
READ FROM LEFT TO RIGHT

ACTUAL PITCH DIAM.	AMOUNT OVER BASIC	ALLOWABLE ERROR IN LEAD PER INCH	
		TAP	TAPPED HOLE
1.8572	0.0015	±0.0003	±0.0004
1.8580	0.0023	±0.0006	±0.0007
1.8590	0.0033	±0.0009	±0.0010
1.8600	0.0043	±0.0012	±0.0013
1.8610	0.0053	±0.0014	±0.0015
1.8616	0.0059	±0.0016	±0.0017

FOR A SCREW OR BOLT
READ FROM RIGHT TO LEFT

ALLOWABLE ERROR IN LEAD PER INCH	AMOUNT UNDER BASIC	ACTUAL PITCH DIAM.
±0.0000	0.0000	1.8557
±0.0002	0.0007	1.8550
±0.0005	0.0017	1.8540
±0.0008	0.0027	1.8530
±0.0010	0.0037	1.8520
±0.0014	0.0047	1.8510
±0.0017	0.0059	1.8498

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FORMULÆ FOR MEDIUM-FIT SCREWS, NUTS, TAPS, ETC., SUITED FOR GENERAL USE

SYMBOLS USED IN FORMULÆ

Basic full or external diameter $= D$
Basic pitch diameter $= E$
Basic root diameter $= K$
Number of threads per inch $= n$
Normal lead $= L$

SCREWS

Max. external diam. $= D$
Max. pitch diam. $= E$
Max. root diam. $= K$

Min. external diam. $= D - \left(\frac{0.102}{n} + \frac{0.054}{n+40}\right)$

Min. pitch diam. $= E - (0.0045 \times \sqrt{D} - 0.0005)$

Min. root diam. $= K - \left(\frac{0.033}{n} + \frac{25}{n+40}\right)$

Max. error ($\pm$ normal) allowable in lead per inch. Length of engagement up to one diam. $= \frac{0.57735\ (0.0045 \times \sqrt{D} - 0.0005)}{D}$

(This formula also applies to tapped holes.)

TAPS AND TAPPED HOLES

Max. external diam. $= D + \frac{0.04}{n} + \frac{0.224}{n+40}$

Max. pitch diam. $= E + 0.0045 \times \sqrt{D} - 0.0005$

Max. root diam. $= K + \frac{0.033}{n} + \frac{0.25}{n+40}$

Min. external diam. $= D + \frac{0.112}{n+40}$

Min. pitch diam. $= E + \frac{0.0045 \times \sqrt{D} - 0.0005}{4}$

Min. root diam. $= K + \frac{0.02}{n}$

TAPS ONLY

Max. error ($\pm$ normal) allowable in lead =

$$\begin{array}{c}\text{Max. error } (\pm \text{ normal})\\ \text{allowable in lead of screw}\end{array} - \frac{\left(\begin{array}{c}\text{Max. error } (\pm \text{ normal})\\ \text{allowable in lead of screw}\end{array}\right)^2}{0.03}$$

GAGING DEVICES

62 In order to ascertain whether the work comes within the required limits as represented by the triangular zones of the tables, it is desirable to measure simultaneously both the pitch diameter and the lead.

63 Among the tools suggested for accomplishing this purpose are the following: In Fig. 9 is shown a gage which is adjustable for a range of diameters, the micrometer thimble being used to obtain the readings for the diameter. An adjustable block supports the work so that it will be held parallel to the center line and can be set to be measured on the center line. The grooved roll fits over the thread and is free sideways to allow for variations in lead. This roll is set to the standard pitch diameter of the work and is adjusted by the micrometer thimble. The floating point is so connected that the longer lever shows the variation in lead and the shorter lever variations in pitch diameter, each pivoting about its own center. The work is placed between the points as shown by the dotted section, and the variations from standard or pitch diameter and lead are read directly in thousandths of an inch.

64 In Fig. 10 is a combination gage for diameter and lead. Point *A* is adjustable longitudinally by means of a micrometer screw so that it can be placed in proper relation to *B*, which is adjustable for different diameters, the pitch diameter of work being read directly by means of its micrometer. The point *C* is connected to the indicator, variations being read directly on the scale in thousandths of an inch. A block supports the work in proper relation to the gage points, and is set by means of the micrometer *D*.

65 A combination gage for pitch diameter and lead, having a fixed point and two adjustable points, is shown in Fig. 11. The variations in both cases are read on the dial indicators in thousandths of an inch. Indicators must be set to a standard before testing the work. An adjustable block may be set by a vernier or micrometer, so that work resting on it will have its center line in line with the gage points.

66 In Figs. 12 and 13 is shown the type of gages now generally in use for measuring screw threads. Gages of this type do not determine the combined error of lead and diameter but are satisfactory for many classes of work. If the thickness or length of gage is made to correspond with the length of fit between the screw and nut or

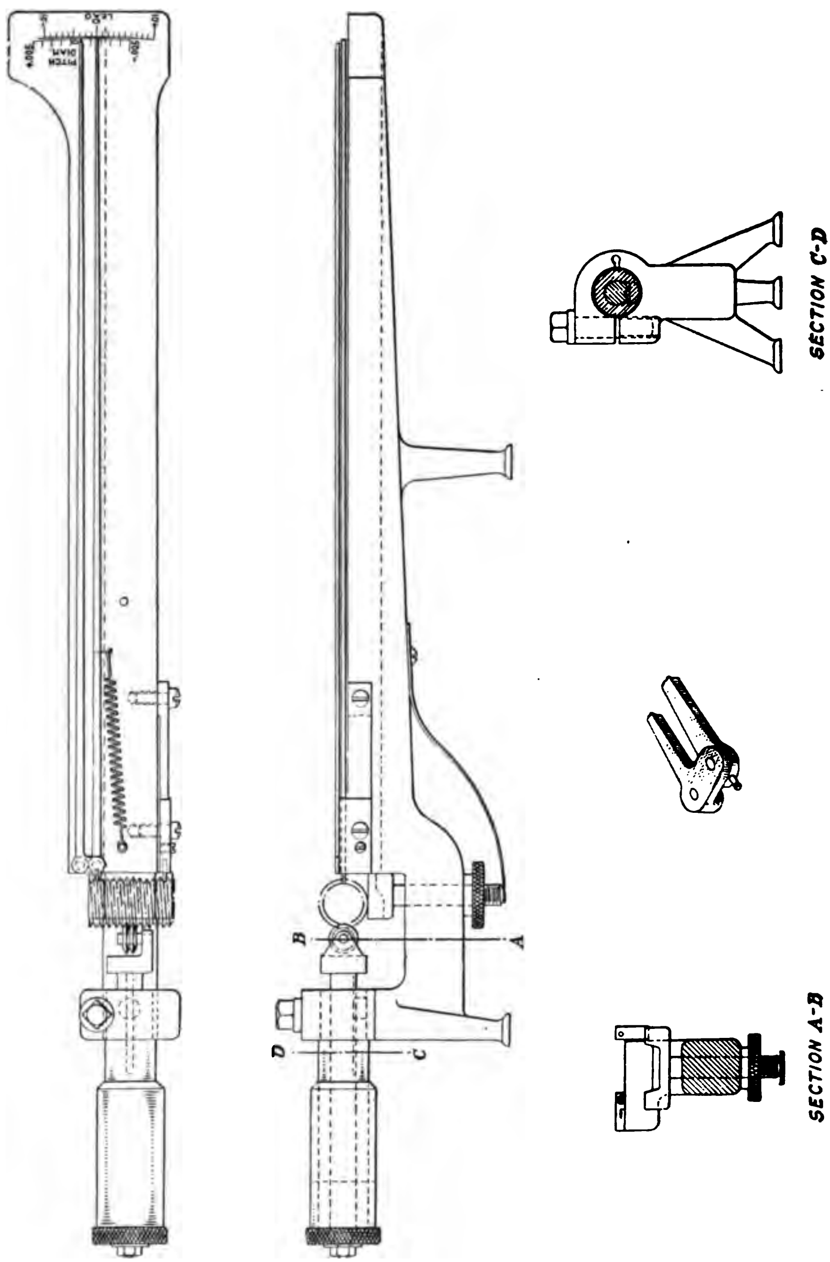

FIG. 9 COMBINED LEAD AND DIAMETER MEASURING GAGE, WITH COMPOUND LEVERS

tapped hole, work which will pass inspection by the gages will interchange in use, and thus the desired result will be attained.

67 Where expensive special tools are not available, good results can be obtained by measuring the diameter with an ordinary *screw-*

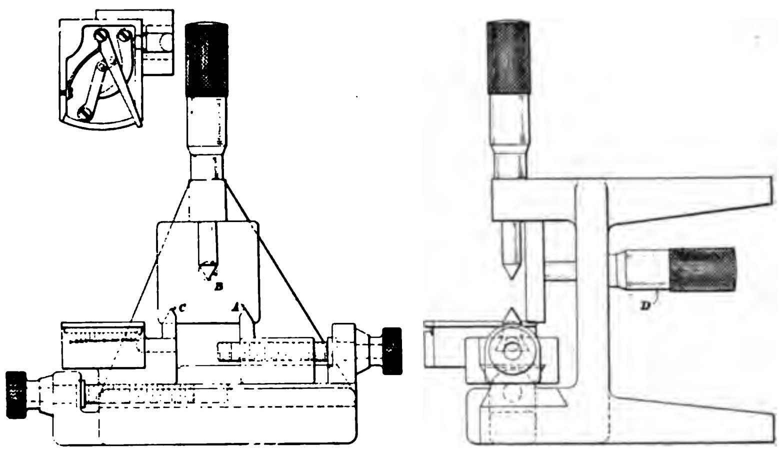

FIG. 10 COMBINED LEAD AND DIAMETER MEASURING GAGE, WITH MICROMETER AND LEVER INDICATOR READINGS

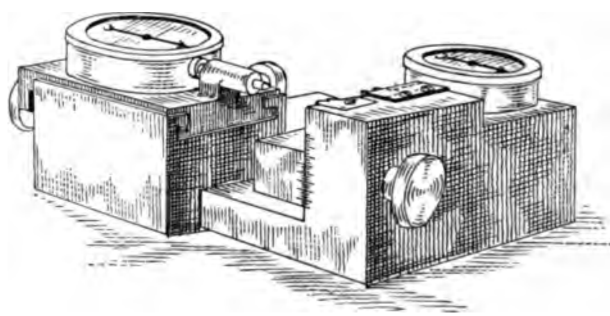

FIG. 11 GAGE WITH DIAL INDICATOR READINGS

thread micrometer shown in Fig. 14, and the lead, with the *lead-measuring indicator*, shown in Fig. 15. By the combined use of gages shown in Figs. 14 and 15, it can readily be determined whether the work will come within the triangular zones on the tables, and thus pass inspection.

68 Another lead measuring instrument is shown in Fig. 16. This uses a commercial test indicator for noting the variations from correct lead.

69 Fig. 17 shows a simple gage which might be provided at small expense where a block of a thickness of about one or one and one-half diameters is tapped at one end for approximately standard pitch diameter and correct lead, while the other end is made thin and tapped under standard pitch diameter so as to be used as a minimum-diameter gage. A screw which will pass through the standard or thicker gage but will not pass through the thin gage may be presumed to be within the required limits of tolerances for variation in both lead and pitch diameter.

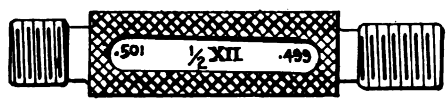

Fig. 12

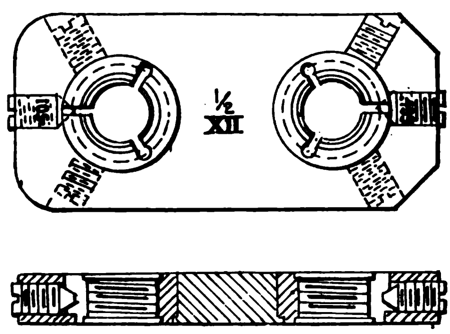

Fig. 13 Plug and Ring Gages

70 Many other designs of measuring instruments have been submitted to the committee, having points of excellence, but it has been thought best here to show simply typical and suggestive designs which can be developed in detail to suit particular needs.

71 Tables 19 and 20 give minimum and maximum diameters for U. S. S. taps, nuts and screws; while Tables 21 and 22 give inspection limits for taps and screws, with the varying allowance in lead for different diameters within the limits specified.

DEFINITIONS AND SYMBOLS

72 The letter symbols given below are those recommended by the U. S. Bureau of Standards:

Allowance. Variation in dimensions to allow for different qualities of fit.

Angle Diameter. Same as pitch or effective diameter.

Angle of Thread, "*A.*" The total or included angle between the sides or slopes of a thread, in a plane passing through the axis of the screw or nut.

Clearance. The space between a screw and a threaded hole.

Clearance Angle. Allowance on the angles or slopes of the thread for screw threads to fit together.

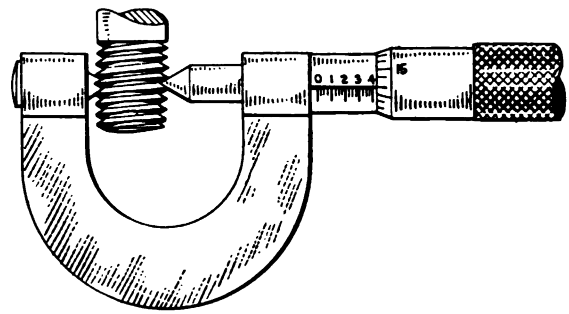

FIG. 14 SCREW THREAD MICROMETER

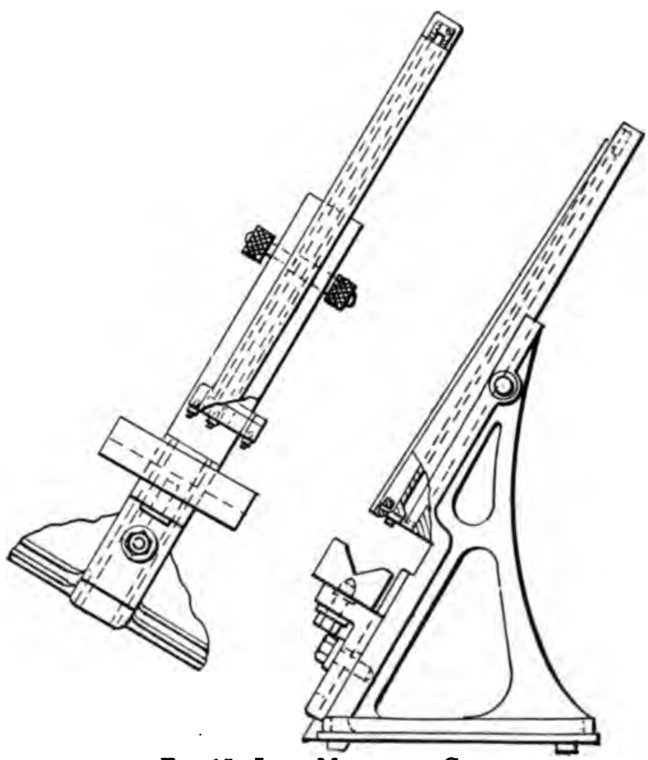

FIG. 15 LEAD MEASURING GAGE

Clearance, Bottom. Allowance or space at bottom of a thread to prevent a bearing at this point and to provide space for dirt.

Clearance, Outside. Allowance between outside diameter of screw and bottom of tapped hole.

Core Diameter, "*K*." English term for the root or bottom diameter of a screw and the small diameter of a nut. In the case of the nut it is measured between the crests of the thread. See Root Diameter.

Crest. English term for the top or most prominent part of a thread, whether on the screw or in the nut.

Effective Diameter, "*E*." English term for pitch diameter and defined as the length of a line drawn through the axis and at right angles to it, measured between the points where the line cuts the slopes of the thread.

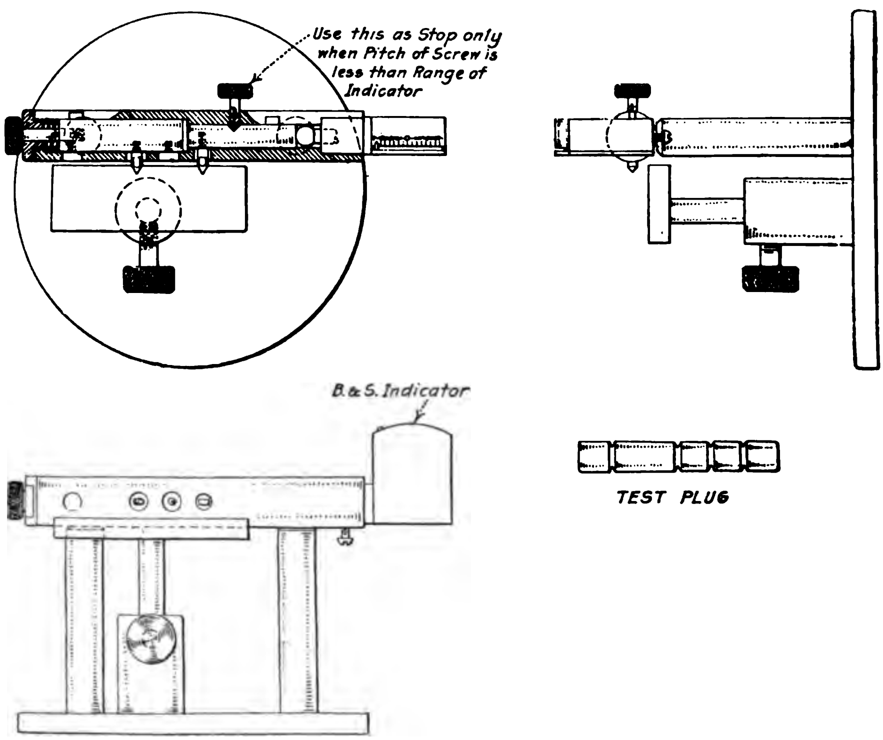

FIG. 16—LEAD MEASURING GAGE WITH INDICATOR READING

External Diameter, "*D*." Same as full diameter or outside diameter.

Finger Fit. Where the screw fits the tapped hole so as to just be screwed in with the fingers.

Flow of Thread. The movement of metal in a screw or nut, or both, when screwed together by force, to fit in spite of an error in lead.

Flute. The groove cut in taps and reamers to form cutting edges and to allow room for chips.

Franklin Institute Thread. The *form* of thread adopted by The Franklin Institute in 1864. It is a 60-deg. angle thread with $\frac{1}{8}$ of the vertical height cut from the top and filled in at the bottom. It is not confined to any special series of pitches.

Full Diameter, "D." English term for outside diameter.

Gage, Check or Checking. Gage for checking or testing other gages.

Gage, Limit. A gage for insuring that any given dimension is within the tolerance laid down for the class of work to be produced.

Gage, Master. A gage which is kept as a standard, solely for comparing reference gage is tested. A copy of the Master Gage.

Gage, Reference. A gage used by the manufacturer and by which the workman's gage is tested. A copy of the Master Gage.

Gage, Shop or Workman's. A gage used by the workman in everyday practice. It is tested by or with the Reference Gage.

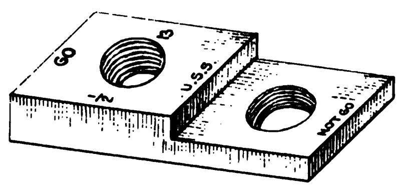

FIG. 17 CHEAP GAGE FOR TEMPORARY UCE

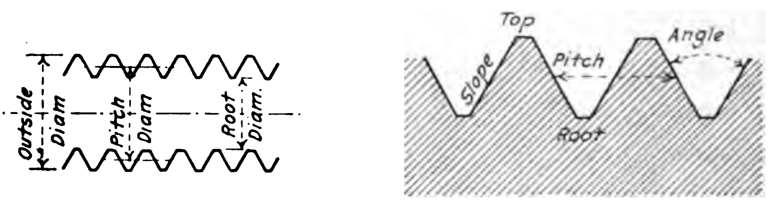

FIG. 18 DIAGRAMS SHOWING ELEMENTS OF THREAD

Gage, Standard. English term for Master Gage.

Land. The space between flutes on a tap or reamer. It includes the cutting edge and the supporting metal behind it.

Lead, "L." The distance a screw advances in one turn. In a single-thread screw this is the same as pitch.

Lead, Normal, "L." Correct lead.

Lead per Inch. The lead multiplied by its reciprocal. For a perfect thread this equals one inch.

Limits. Two sizes expressed by positive dimensions, the larger being termed the maximum, and the smaller the minimum, limit.

Limit Gage. See Gage, Limit.

Outside Diameter, "D." Diameter at the outside of the thread. External or full diameter.

Pitch. The distance from a given point on one thread to a similar point on the next thread, along the axis of the screw. The same as lead for a single thread. The reciprocal of threads per inch.

Pitch Diameter, "E." Same as effective diameter. Also defined as the diameter of a screw at a point midway of the depth of the thread. Equal to the outside diameter less the depth of one thread. This depth equals:

$$\text{For "V"-threads} = \frac{0.866}{\text{Thds. per inch}}. \quad \text{For U. S. S.} = \frac{0.6495}{\text{Thds. per inch}}.$$

TABLE 19 MINIMUM AND MAXIMUM PITCH DIAMETERS OF TAPS AND SCREWS AT PERFECT LEAD FOR U. S. STD.

MEDIUM GRADE FOR GENERAL WORK. ALL DIMENSIONS IN INCHES

TAP				SCREW	
Nom. diam. and threads per inch	Basic pitch diameters	Min. and max. add. to basic pitch diam.	Min. and max. pitch diam.	Max. and min. reduction from basic pitch diam	Max. and min. pitch diam.
1/4—20	0.2175	0.0004	0.2179	0.0000	0.2175
		0.0018	0.2193	0.0018	0.2158
5/16—18	0.2764	0.0005	0.2769	0.0000	0.2764
		0.0020	0.2784	0.0020	0.2744
3/8—16	0.3344	0.0006	0.3350	0.0000	0.3344
		0.0023	0.3367	0.0023	0.3321
7/16—14	0.3911	0.0006	0.3917	0.0000	0.3911
		0.0025	0.3936	0.0025	0.3886
1/2—13	0.4501	0.0007	0.4508	0.0000	0.4501
		0.0027	0.4528	0.0027	0.4474
9/16—12	0.5084	0.0007	0.5091	0.0000	0.5084
		0.0029	0.5113	0.0029	0.5055
5/8—11	0.5660	0.0008	0.5668	0.0000	0.5660
		0.0031	0.5691	0.0031	0.5629
3/4—10	0.6851	0.0009	0.6860	0.0000	0.6851
		0.0034	0.6885	0.0034	0.6817
7/8—9	0.8029	0.0009	0.8038	0.0000	0.8029
		0.0037	0.8066	0.0037	0.7992
1—8	0.9188	0.0010	0.9198	0.0000	0.9188
		0.0040	0.9228	0.0040	0.9148
1 1/8—7	1.0322	0.0011	1.0333	0.0000	1.0322
		0.0043	1.0365	0.0043	1.0279
1 1/4—7	1.1572	0.0011	1.1583	0.0000	1.1572
		0.0045	1.1617	0.0045	1.1527
1 3/8—6	1.2668	0.0012	1.2680	0.0000	1.2668
		0.0048	1.2716	0.0048	1.2620
1 1/2—6	1.3918	0.0013	1.3931	0.0000	1.3918
		0.0050	1.3968	0.0050	1.3868
1 5/8—5 1/2	1.5069	0.0013	1.5082	0.0009	1.5069
		0.0052	1.5121	0.0052	1.5017
1 3/4—5	1.6201	0.0014	1.6215	0.0009	1.6201
		0.0055	1.6256	0.0055	1.6147
1 7/8—5	1.7451	0.0014	1.7465	0.0000	1.7451
		0.0057	1.7508	0.0057	1.7394
2—4 1/2	1.8557	0.0015	1.8572	0.0000	1.8557
		0.0059	1.8616	0.0059	1.8498

TABLE 20 OUTSIDE AND ROOT DIAMETERS FOR TAPS AND SCREWS — U. S. S.T.D.
MEDIUM GRADE FOR GENERAL WORK. ALL DIMENSIONS IN INCHES

TAP OR NUT								SCREW			
Outside Diam.				Root Diam.							
1	2	3	4	5	6	7	8	9	10	11	12
Nominal diam. and threads per inch	Decimal basic outside diam.	Addition to basic O. D.	Min. and max. O. D.	U. S. S. root diam.	Pitch of tool for tap and width across point	New basic root diam.	Min. and max. root diam.	Max. and min. outside diam.	Pitch of tool for thread and width across point	New basic root diam.	Max. and min. root diam.
1/4—20	0.2500	0.0019 0.0057	0.2519 0.2557	0.1850	18 0.0069	0.1863	0.1860 0.1908	0.2500 0.2440	22 0.0057	0.1841	0.1850 0.1792
5/16—18	0.3125	0.0019 0.0061	0.3144 0.3186	0.2403	16 0.0078	0.2418	0.2414 0.2464	0.3125 0.3059	20 0.0062	0.2391	0.2403 0.2342
3/8—16	0.3750	0.0020 0.0065	0.3770 0.3815	0.2936	14 0.0089	0.2955	0.2949 0.3001	0.3750 0.3677	18 0.0069	0.2921	0.2936 0.2871
7/16—14	0.4375	0.0021 0.0070	0.4396 0.4445	0.3447	13 0.0096	0.3459	0.3461 0.3517	0.4375 0.4292	16 0.0078	0.3428	0.3447 0.3377
1/2—13	0.5000	0.0021 0.0073	0.5021 0.5073	0.4001	12 0.0104	0.4015	0.4016 0.4074	0.5000 0.4911	14 0.0089	0.3989	0.4001 0.3928
9/16—12	0.5625	0.0022 0.0076	0.5647 0.5701	0.4542	11 0.0114	0.4559	0.4559 0.4618	0.5625 0.5530	13 0.0096	0.4528	0.4542 0.4466
5/8—11	0.6250	0.0022 0.0080	0.6272 0.6330	0.5069	10 0.0125	0.5089	0.5087 0.5148	0.6250 0.6147	12 0.0104	0.5053	0.5069 0.4990
3/4—10	0.7500	0.0022 0.0085	0.7522 0.7585	0.6201	9 0.0139	0.6225	0.6221 0.6284	0.7500 0.7387	11 0.0114	0.6181	0.6201 0.6118
7/8—9	0.8750	0.0023 0.0090	0.8773 0.8840	0.7307	8 0.0156	0.7337	0.7329 0.7395	0.8750 0.8626	10 0.0125	0.7283	0.7307 0.7219
1—8	1.0000	0.0023 0.0092	1.0023 1.0092	0.8376	7 0.0179	0.8416	0.8401 0.8469	1.0000 0.9861	9 0.0139	0.8346	0.8376 0.8283
1 1/8—7	1.1250	0.0024 0.0105	1.1274 1.1355	0.9394	6 0.0208	0.9444	0.9423 0.9494	1.1250 1.1093	8 0.0156	0.9356	0.9394 0.9294
1 1/4—7	1.2500	0.0024 0.0105	1.2524 1.2605	1.0644	6 0.0208	1.0696	1.0673 1.0744	1.2500 1.2343	8 0.0156	1.0606	1.0644 1.0544
1 3/8—6	1.3750	0.0024 0.0115	1.3774 1.3865	1.1585	5 1/2 0.0227	1.1618	1.1618 1.1694	1.3750 1.3568	7 0.0179	1.1533	1.1585 1.1476
1 1/2—6	1.5000	0.0024 0.0115	1.5024 1.5115	1.2835	5 1/2 0.0227	1.2868	1.2868 1.2944	1.5000 1.4818	7 0.0179	1.2783	1.2835 1.2726
1 5/8—5 1/2	1.6250	0.0025 0.0122	1.6275 1.6372	1.3888	5 0.0250	1.3928	1.3925 1.4003	1.6250 1.6053	6 0.0208	1.3855	1.3888 1.3773
1 3/4—5	1.7500	0.0025 0.0130	1.7525 1.7630	1.4902	4 1/2 0.0278	1.4950	1.4942 1.5024	1.7500 1.7284	5 1/2 0.0227	1.4863	1.4902 1.4780
1 7/8—5	1.8750	0.0025 0.0130	1.8775 1.8880	1.6152	4 1/2 0.0278	1.6200	1.6192 1.6274	1.8750 1.8534	5 1/2 0.0227	1.6113	1.6152 1.6030
2—4 1/2	2.0000	0.0025 0.0139	2.0025 2.0139	1.7113	4 0.0313	1.7173	1.7158 1.7243	2.0000 1.9761	5 0.0250	1.7065	1.7113 1.6984

TABLE 21 INSPECTION LIMITS FOR U. S. S. TAPS

ALL DIMENSIONS IN INCHES

¼-IN. 20 U. S. S. BASIC PITCH DIAM. 0.2175		5/16-IN. 18 U. S. S. BASIC PITCH DIAM. 0.2764		⅜-IN. 16 U. S. S. BASIC PITCH DIAM. 0.3344		½-IN. 13 U. S. S. BASIC PITCH DIAM. 0.4301	
Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.
0.2179	±0.0005	0.2769	±0.0005	0.3350	±0.0006	0.4508	±0.0006
0.2185	±0.0018	0.2775	±0.0016	0.3355	±0.0013	0.4515	±0.0013
0.2190	±0.0029	0.2780	±0.0025	0.3360	±0.0020	0.4520	±0.0019
0.2193	±0.0035	0.2784	±0.0033	0.3365	±0.0028	0.4525	±0.0025
				0.3367	±0.0031	0.4528	±0.0028

⅝-IN. 11 U. S. S. BASIC PITCH DIAM. 0.5660		¾-IN. 10 U. S. S. BASIC PITCH DIAM. 0.6851		⅞-IN. 9 U. S. S. BASIC PITCH DIAM. 0.8029		1-IN. 8 U. S. S. BASIC PITCH DIAM. 0.9188	
Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.
0.5668	±0.0006	0.6860	±0.0005	0.8038	±0.0004	0.9198	±0.0004
0.5675	±0.0012	0.6865	±0.0009	0.8045	±0.0009	0.9205	±0.0008
0.5680	±0.0016	0.6870	±0.0012	0.8050	±0.0012	0.9210	±0.0011
0.5685	±0.0021	0.6875	±0.0016	0.8055	±0.0016	0.9215	±0.0014
0.5690	±0.0026	0.6880	±0.0020	0.8060	±0.0019	0.9220	±0.0017
		0.6885	±0.0024	0.8066	±0.0023	0.9225	±0.0019
						0.9228	±0.0021

1¼-IN. 7 U. S. S. BASIC PITCH DIAM. 1.1572		1½-IN. 6 U. S. S. BASIC PITCH DIAM. 1.3918		1¾-IN. 5 U. S. S. BASIC PITCH DIAM. 1.6201		2-IN. 4½ U. S. S. BASIC PITCH DIAM. 1.8557	
Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.	Actual pitch diam.	Allowable error in lead per in.
1.1583	±0.0003	1.3930	±0.0004	1.6215	±0.0004	1.8572	±0.0004
1.1590	±0.0006	1.3940	±0.0008	1.6225	±0.0007	1.8580	±0.0006
1.1595	±0.0009	1.3950	±0.0011	1.6235	±0.0010	1.8590	±0.0009
1.1600	±0.0011	1.3960	±0.0015	1.6245	±0.0014	1.8600	±0.0012
1.1605	±0.0013	1.3965	±0.0017	1.6255	±0.0017	1.8610	±0.0014
1.1610	±0.0016	1.3968	±0.0018			1.8615	±0.0016
1.1617	±0.0019						

TABLE 22 INSPECTION LIMITS FOR U. S. S. SCREWS

ALL DIMENSIONS IN INCHES

¼-IN. 20 U. S. S. BASIC PITCH DIAM. 0.2175

Actual pitch diam.	Allowable error in lead per in.
0.2170	±0.0012
0.2165	±0.0022
0.2160	±0.0033
0.2157	±0.0040

5/16-IN. 18 U. S. S. BASIC PITCH DIAM. 0.2764

Actual pitch diam.	Allowable error in lead per in.
0.2760	±0.0008
0.2755	±0.0017
0.2750	±0.0027
0.2745	±0.0037

⅜-IN. 16 U. S. S. BASIC PITCH DIAM. 0.3344

Actual pitch diam.	Allowable error in lead per in.
0.3340	±0.0006
0.3335	±0.0014
0.3330	±0.0021
0.3325	±0.0029
0.3321	±0.0035

½-IN. 13 U. S. S. BASIC PITCH DIAM. 0.4501

Actual pitch diam.	Allowable error in lead per in.
0.4495	±0.0007
0.4490	±0.0013
0.4485	±0.0018
0.4480	±0.0024
0.4474	±0.0031

⅝-IN. 11 U. S. S. BASIC PITCH DIAM. 0.5660

Actual pitch diam.	Allowable error in lead per in.
0.5655	±0.0005
0.5650	±0.0009
0.5645	±0.0014
0.5640	±0.0018
0.5635	±0.0023
0.5629	±0.0028

¾-IN. 10 U. S. S. BASIC PITCH DIAM. 0.6851

Actual pitch diam.	Allowable error in lead per in.
0.6845	±0.0005
0.6840	±0.0009
0.6835	±0.0012
0.6830	±0.0016
0.6825	±0.0020
0.6820	±0.0024
0.6817	±0.0026

⅞-IN. 9 U. S. S. BASIC PITCH DIAM. 0.8029

Actual pitch diam.	Allowable error in lead per in.
0.8025	±0.0003
0.8020	±0.0006
0.8015	±0.0009
0.8010	±0.0013
0.8005	±0.0016
0.8000	±0.0020
0.7992	±0.0025

1-IN. 8 U. S. S. BASIC PITCH DIAM. 0.9188

Actual pitch diam.	Allowable error in lead per in.
0.9185	±0.0002
0.9180	±0.0005
0.9175	±0.0008
0.9170	±0.0010
0.9165	±0.0013
0.9160	±0.0016
0.9155	±0.0019
0.9148	±0.0023

1¼-IN. 7 U. S. S. BASIC PITCH DIAM. 1.1572

Actual pitch diam.	Allowable error in lead per in.
1.1565	±0.0003
1.1560	±0.0006
1.1550	±0.0010
1.1540	±0.0015
1.1530	±0.0020
1.1527	±0.0021

1½-IN. 6 U. S. S. BASIC PITCH DIAM. 1.3918

Actual pitch diam.	Allowable error in lead per in.
1.3910	±0.0003
1.3900	±0.0007
1.3890	±0.0011
1.3880	±0.0015
1.3870	±0.0018
1.3868	±0.0019

1¾-IN. 5 U. S. S. BASIC PITCH DIAM. 1.6201

Actual pitch diam.	Allowable error in lead per in.
1.6195	±0.0002
1.6190	±0.0004
1.6180	±0.0007
1.6170	±0.0010
1.6160	±0.0014
1.6150	±0.0017
1.6147	±0.0018

2-IN. 4½ U. S. S. BASIC PITCH DIAM. 1.8557

Actual pitch diam.	Allowable error in lead per in.
1.8550	±0.0002
1.8540	±0.0005
1.8530	±0.0008
1.8520	±0.0011
1.8510	±0.0014
1.8498	±0.0017

Relief. The reduced diameter behind the cutting edge of a tap.

Root. Bottom or smallest diameter of thread, whether in screw or nut.

Root Diameter, "K." The smallest diameter, whether for a screw or in a nut.

Slope of Thread. The angular part which connects the large and small diameters of a thread.

Standard Gage. English term for Master Gage. (See Gage, Master.)

Thread, Modified "V." A form of thread having a 60-deg. angle and such that if carried to a sharp point it would measure to the nominal size, but with the top or bottom, or both, modified usually by being flattened according to conditions or individual ideas.

Threads, U. S. S. The standard adopted by the United States Government, which uses the Franklin Institute form of thread with a definite pitch for each diameter. (See Franklin Institute Thread.))

Thread, "V." A form of thread having a 60-deg. angle and sharp at top and bottom. Impossible in practice and always more or less modified, whether intentionally or not.

Thread, Whitworth. A thread having a 55-deg. angle and a rounded top and bottom. The proportions are:

$$\text{Depth} = \frac{0.640327}{\text{Thds. per inch}}. \quad \text{Radius of top and bottom} = \frac{0.37329}{\text{Thds. per inch}}.$$

Thread Micrometer. A micrometer caliper with special points for measuring the pitch or angle diameter of the screw.

Thread per Inch, "n." Number of threads in one inch of length.

Tolerance. The allowable variation in size, equal to the difference between the minimum and maximum limits.

Turns per Inch, "N." The number of turns required to advance one inch. Equal to the threads per inch of a single-threaded screw.

Wrench Fit. Where the screw fits the tapped hole so tightly as to require wrench to screw into place. Used for cylinder studs in steam engines and for similar work.

Respectfully submitted,

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No. 1780

REPORT OF RESEARCH SUB-COMMITTEE ON BEARING METALS

[THE MICROCHARACTER AT WORK]

THE work done by the Sub-Committee on Bearing Metals has been along the lines indicated in our report presented at the Detroit Meeting,[1] with which we still find ourselves in complete accord. Our investigations have been somewhat broadened to include both the non-ferrous and the ferrous alloys to such an extent that we now offer a *Tentative Scale of Microhardness.* Our instrument the Microcharacter has also been further improved and refined.

2 The importance of studying the physical properties of the individual crystals of the bearing alloys was very forcibly demonstrated by one of our early experiments, as illustrated by the photomicrographs shown in Figs. 1 to 4, inclusive. Fig. 1 shows a small part of the surface of a bearing bronze, a copper-tin alloy, which had first been ground and polished and then strongly etched with ammonium persulphate. This left the copper-tin delta crystal unaffected and in pronounced relief, while the portions containing a higher percentage of copper were etched to proportionately greater depths, the deeper surfaces corresponding to the alpha and the intermediate to the beta crystals. A small piece of cold-rolled steel shafting was then ground and highly polished — a steel containing 0.12 per cent of carbon, and having a Brinell hardness of 175 and a scleroscope hardness of 24. The etched bronze surface was placed in a crosswise position upon this polished steel surface and then subjected to a pressure of 50,000 lb. per sq. in. of contact area. Figs. 2 and 3 show the edges of the steel surface where the bronze had overlapped. They both show a distinct indentation made by the delta crystal into the steel. The larger part of the steel surface, say, 95 per cent, remained

[1] June 1919.

Received by the Council, November 17, 1920, and ordered printed. Presented at the Annual Meeting, December 1920, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

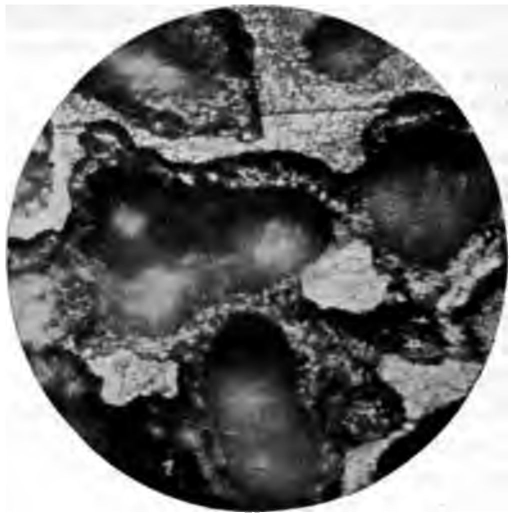

Fig. 1 Photomicrograph of an Etched Bronze Surface. 300 Magnifications

Fig. 2 Photomicrograph of Surface of Steel with the Copper-Tin Delta Crystal Impressed upon It. 300 Magnifications

Fig. 3 Photomicrograph of Surface of Steel with the Copper-Tin Delta Crystal Impressed upon It. 300 Magnifications

Fig. 4 Photomicrograph of Same Surface as Fig. 1 after Being Subjected to a Pressure of 50,000 lb. per sq. in. 300 Magnifications

entirely unaffected and retained its original polish, which seemed strange in view of the heavy load to which these surfaces had been subjected. But upon examining the condition of the bronze surface, matters seemed to be accounted for. The softer or plastic crystals of the bronze had allowed the harder ones to become embedded in them, and thus as shown in Fig. 4 the surface originally shown in

FIG. 5 PHOTOMICROGRAPH OF STEEL SURFACE SHOWING END OF MICRO-CUT. 2000 MAGNIFICATIONS

Fig. 1 has the order of the surface crystals completely reversed, that is, the alpha crystal which had been deeply etched out is now in bold relief, as shown by the surfaces, the areas in focus; and the delta crystal which was originally in relief is now distinctly intaglio.

FIG. 6 SHADOWGRAPH SHOWING THE POINT OF FINEST CAMBRIC NEEDLE—SHARPS NO. 12. 2000 MAGNIFICATIONS

This of course is possible only in an alloy whose various constituents possess varying degrees of hardness. The fact that the bearing bronze had constituent elements harder than the steel shaft seemed remarkable.

3 The importance of a knowledge of the microcharacteristics of alloys is not new. It has been fully appreciated by many of the

early investigators who first applied the use of the microscope to the study of alloys, as is shown in their early work. The only means

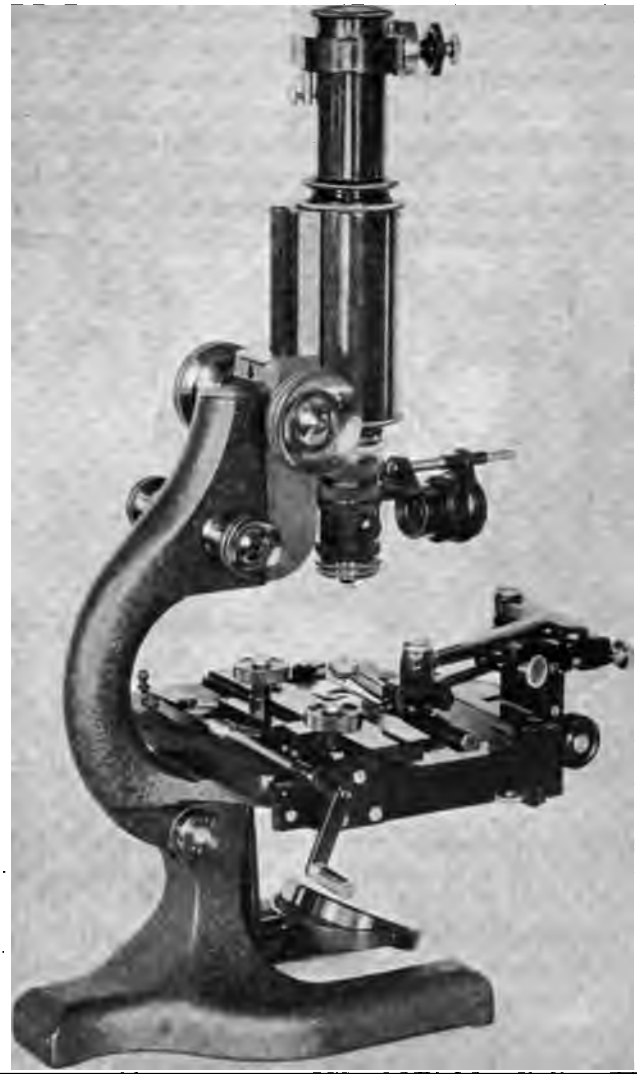

FIG. 7 MICROSCOPE COMPLETE WITH MICROCHARACTER ATTACHED

for this study until now, however, has been the needle point, and when this is compared with the finest of the sapphire points, which have been successfully ground for our latest instrument, the contrast becomes striking. Fig. 5 shows the end of a micro-cut in soft steel with a sapphire point under a magnification of 2000 diameters, while Fig. 6 shows under exactly the same magnification a shadowgraph of the point of the finest cambric needle, Sharps No. 12. A comparison of the two shows that a refinement of at least a thousand-fold has been effected. In addition, the hardness of the sapphire as compared with steel offers a means for testing the hardness of the individual crystals of which the steel needle is composed. The fact that these sapphire points have been accurately ground and mounted according to definite specifications and can now be perfectly duplicated, makes

Fig. 8 The Microcharacter

it possible to supply various investigators with a standard instrument for doing our work in a manner that permits direct comparison of results.

4 A large part of the work done by the Committee up to the present time has been that necessary for the perfecting of the microcharacter and the preparation of a proposed hardness scale. Fig. 7 shows the instrument complete, mounted upon the stage of an ordinary microscope, the latter being equipped for work with a vertical illuminator with condensing lenses and an ocular provided with a movable micrometer scale. Fig. 8 shows the microcharacter separate from the microscope with the jewel suspension bracket swung out. At the left is seen the worm gear for the micrometer feed, the handle

of which is rotated when making a cut at the rate of, say, 30 r.p.m., which gives a very slow and continuous feed to the jewel point. Fig. 9 shows the jewel suspension. The jewel is mounted in a delicate steel spring directly beneath the point of application of the weight; it is suspended slightly above its center of gravity on sapphire jewels, and the left end is provided with a spirit level.

5 The distinctive features of our instrument are: first, the extreme refinement of the shape and proportions of the jewel point; second, the special advantages due to the particular shape of the point; third, the micrometer feed for the jewel point; and fourth, the elastic mounting of the jewel point. We have found that the material for the jewel point as given in our first report,[1] namely, the artificial white sapphire (fused Al_2O_3), is very satisfactory and seems to meet all requirements for hardness and toughness. The

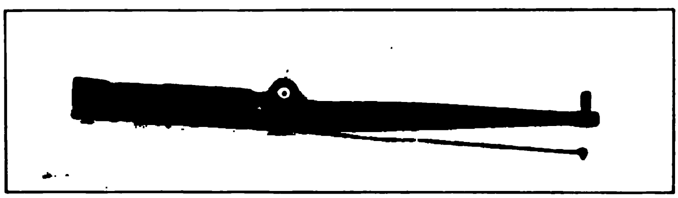

FIG. 9 JEWEL SUSPENSION OF MICROCHARACTER

shape of the point, a corner of a cube, accurately mounted so that the three facets make equal angles with the test surface, with one edge of the cube advancing in direct line of motion, seems to possess special advantages. The advance edge of the cube and the line of cut form an angle of 35.26 deg., the *angle of incision*, which insures that the normal pressure upon a crystal is always greater than the transverse pull, thus completely eliminating the objectionable tearing effects of a sharp conical point; moreover, we have thus far been entirely unable to grind a round conical point to the same degree of accuracy as one made by grinding three intersecting plain surfaces. Fig. 10 shows two views of this jewel point: an end view (1) showing the three facets ground at an angle of 54.73 deg. with the axis, each facet being ground after rotating the jewel upon its axis 120 deg.; and a side view (2), the lower edge at the right being the advance or cutting edge. To start with, the jewel is a small cylinder, say, 0.022 in. in diameter and $\frac{3}{32}$ in. long, and the point is ground and

[1] MECHANICAL ENGINEERING, January 1919, p. 71.

tested before the jewel is mounted. The mounting is done by accurately clamping both the jewel and the jewel spring in a fixture while in the field of the microscope. A heavy solution of specially refined shellac is then applied to the point on the lower side of the spring and permitted to harden. The spring is then removed from the jig and a small amount of this shellac is applied to the jewel on the upper side of the spring. After this has hardened the excessive top length of the jewel is ground off in order to reduce its weight to an absolute minimum. Alcohol should therefore never be used in cleaning the point, xyline being the most desirable solvent to employ. Early in our work it was found that the jewel point should move at a slow and constant rate in order to insure corroborative results. This requirement has been met by means of a worm gear. The rate of travel should be so slow that no additional indentation

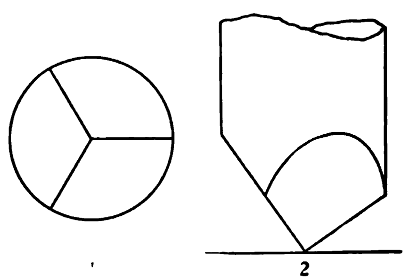

FIG. 10 DETAILS OF JEWEL POINT

is effected by stopping. At the same time it was shown that the cutting point should be held in such a manner that the point while passing suddenly from one crystal into another, could instantly take a new position without having to overcome excessive inertia of moving parts. Accordingly the jewel itself is reduced in weight as much as possible and is mounted in the end of an elastic steel blade which tapers evenly toward the jewel point.

6 It was found desirable to lubricate the test surfaces, and watch or porpoise-jaw oil was at first used for this purpose. It was found, however, to be difficult to completely remove this oil, and with the application of immersion oil, which is not miscible with watch oil, optical difficulties were encountered, since all measurements are made with the high-power immersion objectives. The immersion oil used is the standard optical cedar oil having a specific gravity of 0.976 at 15 deg. cent. The same oil may now be left upon the surface while making the necessary measurements, after which the

oil may be left to dry upon the specimen and so form an effective coating against corrosion. The jewel point should be kept clean and should be cleaned immediately after use, since a small amount of oil dried on alone or in connection with other material may produce very erratic and unaccountable results, even though this dried-on material be so small as to be barely visible. We have found that elderberry pith wetted with xylol is a very desirable material for

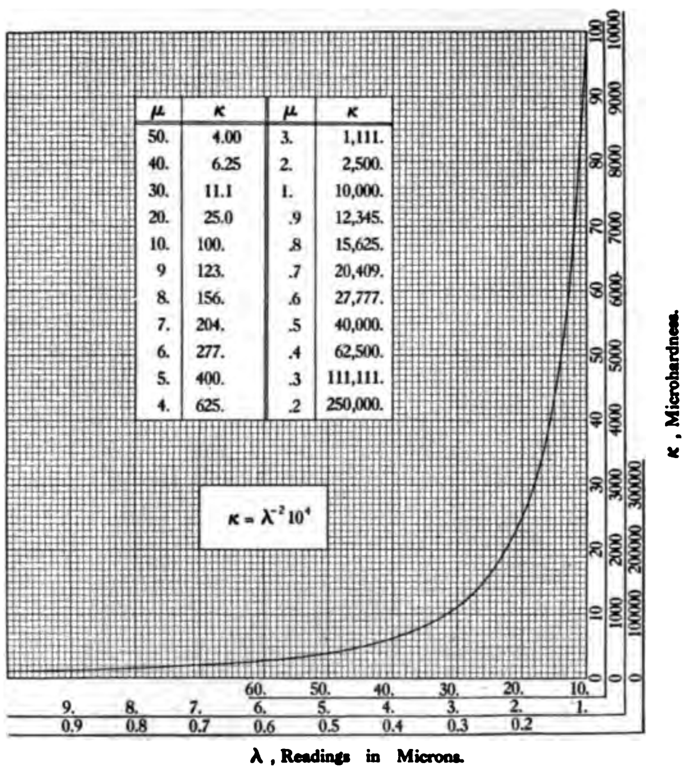

FIG. 11 DIAGRAM OF SCALE OF MICROHARDNESS

cleaning the jewel point. This pith is also very suitable for wiping off the test surfaces after a cut has been made, since a freshly cut surface of this material will not scratch or mar the test surfaces and will carry away the loose chips made by the cut.

7 Many crystals of a known definite chemical composition could be tested in bulk without resorting to the extreme refinements here entered into, but all crystals formed due to a varying degree of

Fig. 12 Photomicrograph Showing Micro-Cut in Two Adjacent Grains Having Same Hardness. Pure Copper. 800 Magnifications

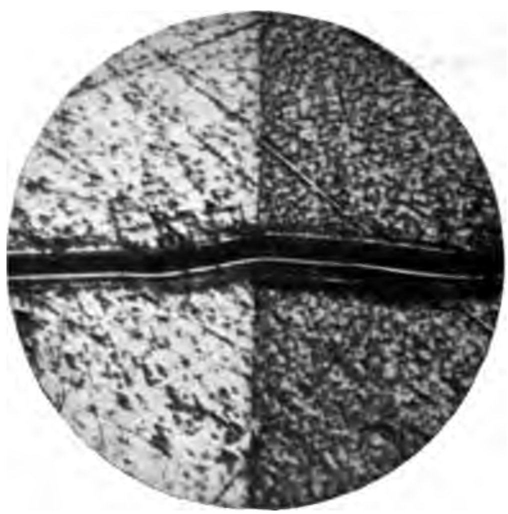

Fig. 13 Photomicrograph of Micro-Cut on Same Crystals as Those of Fig. 12, Showing Difference in Hardness. 800 Magnifications

solubility of constituents under varying temperature, together with the various eutectic formations in general, demand the utmost refinement in this work.

8 A three-gram weight seems to be the lightest that will make a sufficient impression upon the hardest crystals that have to be taken into consideration in our work, and at the same time give a reading the measurement of which shall be possible and within the limits of the modern microscope.

9 The proposed Scale of Microhardness is shown in Fig. 11. It is a curve plotted on two rectangular coördinates; the readings

Fig. 14 Photomicrograph of Micro-Cut Showing Difference of Hardness of Bronze and Fused Tin Oxide. 1000 Magnifications

of the widths of cuts are given on the X-axis and the corresponding degrees of hardness are given on the Y-axis. The micron, a thousandth of a millimeter and a unit commonly used in physics, seems for our purpose to be a convenient unit of length and as such is usually represented by the character μ. For the reason that the various cross-sections of micro-cuts in the different materials are symmetrical and have areas proportional to the squares of their widths, these squares should be taken into consideration; and for the reason that the material which has the narrower micro-cut is the harder, the reciprocals of the squares should be used. Now, if we let κ be the microhardness and λ the reading for the different widths of micro-cuts, in terms of microns, and if we then multiply the reciprocal of the square by 10 raised to a convenient power, we have $\kappa = \lambda^{-2}\ 10^4$. This formula has been plotted and is shown in Fig. 11. We believe that in this work we have been the first to succeed in establishing a rational scale of hardness, one that expresses in definite units the relation of the hardness between different materials.

10 The hardness of some materials varies considerably in different directions. For example, we find this property illustrated in Figs. 12 and 13, both of which show a micro-cut across the surface

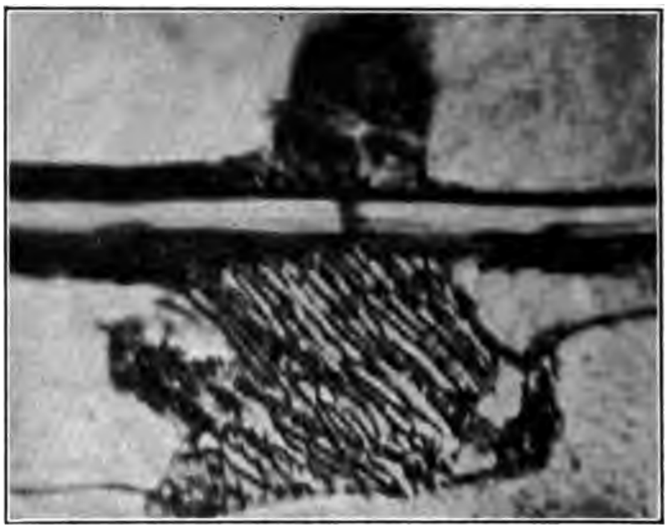

FIG. 15 PHOTOMICROGRAPH OF MICRO-CUT IN SOFT STEEL SHOWING HARDNESS OF FERRITE AND PEARLITE. 1500 MAGNIFICATIONS

of chemically pure copper. In the first we have practically a uniform width of cut, although the cut is made across the border line of two adjacent grains. In Fig. 13 we have a cut in a different direction at a different point, of the same two grains, and we have an appreciable difference in hardness of the two grains as shown by the width of micro-cut. This is due to the different crystalline orientation in the grains.

11 Fig. 14 shows a micro-cut across a portion of a fused tin oxide crystal embedded in bronze. The difference in the width of the cut indicates a difference in hardness between the two materials. Fig. 15 shows a micro-cut over the surface of a soft steel,

passing through a pearlite crystal. The square of the width of the cut at the right shows the relative hardness of the ferrite crystal as compared to the narrow part of the cut, that of pearlite.

12 Fig. 16 shows a micro-cut across a martensite structure in hardened tool steel. "It is undoubtedly the hardest constituent of hardened steel."[1]

13 From what has gone before it must be evident that our instrument is adapted for a very wide range of testing, including not only the various bearing alloys, but also all the steels suitable

FIG. 16 PHOTOMICROGRAPH OF MICRO-CUT IN HARDENED STEEL SHOWING MICRO-CUT IN THE MARTENSITE FORMATION. 2000 MAGNIFICATIONS

for journals, whether hardened or soft. We believe that we have reached the point where the general membership of our Society can aid in our work, namely, in referring to our attention old bearings, together with their journals, which have given exceptionally satisfactory service for an unusually long period, and in lending us their assistance in securing such material for our study.

C. H. BIERBAUM, *Chairman*
J. A. CAPP
H. DIEDERICHS

Sub-Committee on Bearing Metals

[1] Introduction to Physical Metallurgy, Rosenhain.

No. 1781

NECROLOGY

JOHN R. ALLEN

John R. Allen, director of the Bureau of Research of the American Society of Heating and Ventilating Engineers in coöperation with the U. S. Bureau of Mines, and vice-president of The American Society of Mechanical Engineers, died on October 26, 1920, of pneumonia.

Professor Allen was born in Milwaukee on July 23, 1869, and received his preparatory education at Milwaukee and at the Ann Arbor, Mich., high school. In 1892 he was graduated from the University of Michigan with the degree of B.S. (M.E.), receiving his master's degree from the same institution in 1896. For two years he was connected with the L. K. Comstock Construction Co. From 1896 to 1911 he was associated with the University of Michigan, rising from an instructorship to a full professorship. In 1912 he was called to Robert College in Constantinople, Turkey, as dean of the engineering department. Two years later he returned to the University of Michigan as head of the mechanical engineering department, being so connected until 1917, when he became dean of the college of engineering and architecture of the University of Minnesota. In August 1919, Professor Allen accepted the directorship of the Bureau of Research of the American Society of Heating and Ventilating Engineers, with headquarters at the U. S. Bureau of Mines, Pittsburgh.

Professor Allen was extremely active in the work of engineering societies, having been a past-president of the American Society of Heating and Ventilating Engineers and of the Michigan Engineering Society. He was an honorary member of the National District Heating Association, a member of the British Institute of Heating and Ventilating Engineers and of the Society for the Promotion of Engineering Education. He was also a member of the honorary societies, Tau Beta Pi and Sigma Xi. He became a member of our Society in 1894, and at the time of his death held the office of vice-president of the Society.

Professor Allen has been a contributor of important technical papers to these societies, papers which have proved to be reference works of great value. He was the author of textbooks, not only on the subject of steam engines, which he taught, but also on heating and ventilation. His book on the latter subject is recognized throughout the country as a standard. His long connection with the University of Michigan gave him a wide acquaintance and he was called for consulting work to all parts of the world. Notable among his engineering activities during the period of his connection with the University of Michigan were his expeditions to the unexplored districts of western Mexico to report upon the existence of rubber-producing trees; to the Rocky Mountains to report on water-power projects, and his engineering expeditions into northern Michigan and southern Canada.

A great lover of outdoor life, Professor Allen was the owner of two beautiful farms in the outskirts of Ann Arbor, one of which was owned jointly with his adopted son. He brought the boy back with him from his Mexican expedition and adopted him that he might have the opportunity of coming to this country and of being educated here. This boy, who is a native Mexican Indian, has of late years operated both farms, and it was Professor Allen's great pleasure to spend much of his leisure time in farming.

He leaves a wife and daughter residing in Pittsburgh, his daughter attending the Margaret Morrison school of Carnegie Institute.

From Dean M. E. Cooley, a very close friend of Professor Allen's, comes the following appreciation:

Prof. John R. Allen's death was untimely. It came in the midst of his most important life work, apart from his teaching days. He was intensely interested in his heating and ventilating researches, and was trying to establish fundamental laws governing the flow of heat under a variety of practical conditions. A number of colleges were coöperating with him. A year or two more would have seen certain benchmarks firmly established. His death was indeed untimely.

Professor Allen was a man of charming personality. He had a host of friends. In the honor Society of Michigamua here at Michigan his Indian name was "Man of Many Friends." He sat in well at committee meetings and conferences. Always careful of the feelings of others and respectful, difficult situations were overcome and his own views, even when radically different, were well received. He took advice well and was himself a good adviser. He was a splendid teacher. His students respected and admired him. He was a good influence on the young men who sat under him. He was one about whom the older alumni inquired years after their college days. That fact alone marked him with especial honor in the teaching world.

Professor Allen was a well-poised man. Unpleasant things did not upset him. He never lost his temper. His rule was to forget unpleasant things — at least to ignore them. If he could not get away from them in one locality he would move to another. Nothing seemed to ruffle him. At least he did not show it. And yet he was sensitive and easily hurt, but always ready to forgive and start over again.

Professor Allen was resourceful and adaptable. He got on well. In foreign countries he had the same respect and devotion from those with whom he came into contact as at home. He was quick to learn the peculiarities of men and to act accordingly. Rarely did anyone get the better of him for long. His construction gangs frequently embraced Italians, Greeks, Turks, Bulgarians, Armenians. But whatever their nationality, men would work for him cheerfully and well.

Professor Allen loved companionship. He enjoyed his friends and could not do too much for them. He was a good banqueter and could always be relied on for some post-prandial remarks. He enjoyed speaking in public and was more than generous in responding to invitations to address gatherings. His experiences in Mexico when investigating rubber-producing trees, and in Turkey, were favorite non-technical subjects and he made them very attractive with his lantern slides.

The following letter has been received from Prof. O. P. Hood, chief mechanical engineer of the U. S. Bureau of Mines, Washington:

Wherever John Allen made contact with people or work, there will be a keen sense of loss at his going.

Dean Allen came to Pittsburgh in a scientific adventure, the coöperative agreement between the American Society of Heating and Ventilating Engineers and the Bureau of Mines being new and untried as a method of furthering science and technology. His selection to head the work proved so happy that the Bureau had no fears whatever for the character of the output. He dropped into the organization with the greatest accommodation, fitting himself to the situation so graciously as to win instant appreciation from all the members of the Bureau of Mines with whom he was brought in contact. Kindly and helpful in every situation, he was a stimulus to good work. The installation of equipment was just being completed and we all looked forward to an early output of valuable information..

The sense of personal loss to members of the Bureau will be slow to fade.

SAMUEL F. BAGG

Samuel F. Bagg was born in Utica, N. Y., in 1849. He received his early education in the Utica schools and later attended Hamilton College from which he received his A.B. and B.S. degrees. From 1871 to 1877 he was connected with the Utica Steam Engineering Works, where he served his apprenticeship and gained his shop experience. In 1878 he became associated with the Watertown Steam Engine Co., Watertown, N. Y. He was with this firm for about thirty years and during this period held the offices of secretary, treasurer, superintendent and general manager, respectively. He resigned from the company to become district agent with the Fidelity & Casualty Co., Troy, N. Y., which position he held at the time of his death, January 1, 1920.

Mr. Bagg became a member of the Society in 1896.

RALPH BAGGALEY

Ralph Baggaley, life associate of the Society since 1891, died on September 23, 1915. Mr. Baggaley was born in Allegheny, Pa., on December 26, 1846. He attended school at the Sewickley and Kenwood Academies, New Brighton, Pa., and for three years studied in Dresden, Germany. Upon his return he was employed by Ballman & Co., manufacturers of water chilled rolls.

A year and a half later the business was liquidated and a new firm was organized under the name of Baggaley, Young & Co., and the foundry and machine business continued. While acting as salesman for this concern, Mr. Baggaley met George Westinghouse who interested him in his idea of using compressed air for railroad brakes. For over a year they worked together on this project, building the necessary equipment in the works of Baggaley, Young & Co. At the end of that period patents were applied for, and a company with a nominal capital of $500,000 was formed, Mr. Westinghouse retaining $200,000 and Mr. Baggaley $100,000.

The Pennsylvania Railroad officials, though interested in the invention, would not give it a test, so stock interests in the company were given to F. L. Jewett, president, W. W. Card, superintendent, and G. D. Whitcomb, purchasing agent of the old Pan Handle Railroad and permission was thus obtained to make tests on the Burgestttown accomodation, provided it cost the railroad absolutely nothing. The train was equipped, the trial was successful and from that time and for twelve years thereafter Mr. Baggaley was manager of the Westinghouse Air Brake Co. He was also manager of the Union, Switch & Signal Co. and for eight years manager of the Westinghouse Machine Co.

In 1875 Mr. Baggaley purchased the *Evening Telegraph* of Pittsburgh which he controlled for seven years, when it was consolidated with the *Evening Chronicle* forming the *Chronicle-Telegraph* of today.

Mr. Baggaley was a director in twenty-eight companies. He organized the Arthurs Coal & Lumber Co; he was president of the Lake Superior Iron Co. and took an active part in the organization and building of the Pittsburgh & Lake Erie Railroad.

He purchased coal land in the Connersville field and built the Puritan Coke Works, now leased to the H. C. Frick Co. at the town of Baggaley, Pa. He was interested with Franklin B. Gowen in the project of building the South

Pennsylvania Railroad. He also took over the management of the U. S. Glass Co. for one year and brought it safely through a trying period.

Through his study of operating his Gold Hill property in the Cascade Mountains he became interested in the metallurgy of copper, silver and gold and formed the Montana Copper Co., now the East Butte Co., building the only smelter outside of the Anaconda property in the Butte district. He was the first one to build and operate successfully a basic lined smelter, the patent on which was assigned to the Pittsburgh & Montana Copper Co.

Mr. Baggaley was a life member of the Art Society of Pittsburgh, the American Association for the Advancement of Science, the National Geographic Society, the American Institute of Mining and Metallurgical Engineers and the Strollers Club of New York. He also belonged to the Duquesne and Pittsburgh Clubs.

FRANK HARVEY BALL

In the death of Frank H. Ball in Detroit on November 12, 1920, the mechanical and engineering world lost one of its most active members, as practically all of his life was devoted to the invention and manufacture of useful mechanical devices.

Mr. Ball was born on May 21, 1847, in Oberlin, Ohio, and was educated in the grammar and high schools of Buffalo, N. Y. After some early experiences in a country sawmill on his father's farm he entered into the production of steam engines for drilling and operating oil wells. He brought out the well-known Ball automatic high-speed engine and founded the Ball Engine Co., of Erie, Pa., of which he was general manager from 1880 to 1890. He then formed the Ball & Wood Co., Elizabeth, N. J., acting as general manager of this firm until 1895, when he became general manager of the American Engine Co., Bound Brook, N. J., producing the American Ball engine.

Retiring from the steam-engine business several years ago, he and his youngest son invented and put on the market the Ball & Ball carburetor for gasoline engines. In 1913 he became manager of the carburetor department of the Penberthy Injector Co., Detroit, Mich., a position which he held at the time of his death.

Mr. Ball was not only a successful engineer but also an enthusiastic yachtsman. From his boyhood days on the Niagara River he always owned some kind of a sailing craft and was the winner of many sailing races in Buffalo, Erie and Cleveland. For several years his sailing houseboat, the *Sommerheim*, aroused great enthusiasm and interest in Great South Bay, L. I. In Erie, his devotion to yachting resulted in the formation of the Erie Yacht Club, of which he was made the first and only honorary member.

Mr. Ball became a member of the Society in 1883. He was one of its managers from 1888 to 1891 and from 1894 to 1896 a vice-president. He also belonged to the Society of Automotive Engineers, in which organization he took an active interest.

EVARTS SHANKIN BARNUM

Evarts S. Barnum, of the C. M. Basford Co., died at his home in Ridgewood, N. J., on February 3, 1920. Mr. Barnum was born in Louisville, Ky., in July, 1883. He was educated at Purdue University, being graduated in 1906.

His entire business life was connected with railroad work. Immediately upon his graduation from college he entered the service of the Pennsylvania Lines West as apprentice, and worked successively as apprentice, machinist, foreman, general foreman, roundhouse foreman and motive-power inspector. Leaving the railroad in 1917 he joined the staff of the *Railway Age* as associate editor, later becoming associated with the C. M. Basford Co. in charge of the copy department.

Mr. Barnum became a member of the Society in 1918.

WILLIAM M. BARR

William M. Barr, who joined the Society in the year of its foundation, 1880, died on July 27, 1920. Mr. Barr was born in November, 1842, in Muncy, Pa. He received the degree of mechanical engineering, to which profession he devoted his life. Mr. Barr served in the Navy throughout the Spanish and Civil Wars. He is well known in the world of technical literature for his works on Pumping Machinery, Boilers and Furnaces, Combustion of Coal, etc. His chief work is Barr's Industrial Engineering, a Handbook of Useful Information for managers, engineers, superintendents, designers, draftsmen and others engaged in constructive work, Part I of which was recently issued. Part II is in preparation for early publication.

Mr. Barr was also a member of the American Society of Naval Engineers.

HENRY B. BARTLETT

Henry B. Bartlett was born in December, 1856, in Carbondale, Pa. He began his mechanical experience at the old locomotive works in Paterson, N. J., in 1872 and completed his trade at the Farrel Foundry & Machine Co., Waterbury, Conn. Since that time Mr. Bartlett was connected with some of the largest machine interests of the country as mechanical expert and engineer.

Through his skill and inventions the Mergenthaler Linotype Co. of Baltimore was brought to a manufacturing basis, making possible the use of linotypes in newspaper work. He was also connected with the Dietrich & Harvey Co., Baltimore, in the manufacture of their open-side planer. Later he was with the Bullard Machine Co., Bridgeport, Conn. For two years he was superintendent of the Williams Typewriter Co., Derby, Conn., afterwards becoming superintendent of the small-tool department of the Pratt & Whitney Co., Hartford, Conn.

In 1896 Mr. Bartlett went to Berlin, Germany, as general manager of the Ludwig Loewe Co., the largest manufacturers of shop tools and machines in Germany. While there he became advisory engineer to the Typograph Gesellschaft. In 1902 he returned to the United States, becoming a teacher of tool and machine making at the Hebrew Technical Institute of New York, holding this position for eleven years.

At the time of his death, January 18, 1920, Mr. Bartlett was consulting engineer for B. T. Perkins & Sons and their allied paper interests in Holyoke, Mass.

He was a member of the Engineering Society of Western Massachusetts. He became a member of our Society in 1918.

CHARLES SAMUEL BAVIER

Charles S. Bavier, for the past twenty years chief engineer of the Metropolitan Life Insurance Co. Building, New York City, died on November 27, 1920. Mr. Bavier was born in Jersey City, N. J., on March 4, 1850. He served his apprenticeship with the Oakville Engine & Foundry Co., from 1870 to 1874. He then became assistant to the Chief Engineer of Public Works, Buffalo, N. Y. From 1879 to 1886 he was connected with the following firms in the capacity of master mechanic: W. W. Goodman Meter Co., Philadelphia, Pa., American Meter Co., N. Y., J. L. & W. L. Hastings, contractors, Pittsburgh, and the Louisiana Sugar Refining Co. In 1886 he became superintendent of the latter firm and two years later took charge of the design and erection of the power plant for the New Orleans Cotton Exchange. In 1900 he became chief engineer of the Metropolitan Life Insurance Co. Building.

Mr. Bavier was the inventor of a number of mechanical appliances. He was a member of the American Society of Heating and Ventilating Engineers, the Stationary Engineers, and the Phœnix Society. He became an associate of The American Society of Mechanical Engineers in 1907.

COOLIDGE JOHNSON BENNETT

Coolidge J. Bennett was born on June 9, 1885, in Lowell, Mass. When about seven years of age he was taken to Milltown, N. B., Canada, where his father held the position of mechanical superintendent of the St. Croix Cotton Mills. He was educated in the Milltown schools and during his vacations worked in the machine shop of the mill.

At the age of fourteen he returned to Lowell and started work for the Hamilton Manufacturing Co., where he received a seven-years training as an all-round machinist. During this period he attended evening classes at the Lowell Textile School, the Y. M. C. A. Trade School, Hawley's Steam Engineering School and McNabb's Steam Engineering School, spending seven years in studying to become a mechanical engineer. He held Massachusetts licenses both as a first-class engineer and a first-class electrician.

In 1910 he became connected with Slater & Sons' Woolen Mills, Farnumsville, Mass. Two years later he became engineer in charge of Armour & Co., Lowell. From 1914 to 1915 he held the position of master mechanic with the Federal Shoe Co., Lowell, when he resigned to become assistant chief engineer at the Stone & Webster plant at Canton, Mass. About six months later he became associated with the Bigelow Hartford Carpet Co., Lowell, and at the time of his death, February 24, 1920, held the position of mechanical superintendent of that concern.

Mr. Bennett was a member of the National Association of Stationary Engineers and belonged to a number of fraternal organizations. He became an associate-member of the Society in 1919.

PIERREPONT BIGELOW

Pierrepont Bigelow, treasurer of the Bigelow Co., New Haven, Conn., died on January 27, 1920. Mr. Bigelow was born in August, 1888, in New Haven. He received his early education at Hill School and then attended Sheffield Scientific School, Yale University, from which he was graduated in 1910. Upon leaving college he became connected with the Bigelow Co., serving his apprenticeship in their works. Later he was in charge of the engineering and production work of the plant and at the time of his death held the position of treasurer.

Mr. Bigelow became a junior member of the Society in 1914. He was a member of the Yale Club in New York and the Graduates Club in New Haven.

CHARLES ETHAN BILLINGS

Charles Ethan Billings, chairman of the Board of Directors and former president of the Billings & Spencer Manufacturing Co., Hartford, Conn., died on June 4, 1920, after a long illness.

Mr. Billings was born in Wethersfield, Vt., on December 5, 1835. After receiving a common-school education, he served a three-year apprenticeship in the gun department of the machine works of the Robbins & Lawrence Co., Windsor, Vt. In 1856 he entered the employ of Colt's Patent Firearms Manufacturing Co. as a toolmaker and die sinker. During the early stages of the Civil War he was employed in the gun factories of E. Remington & Sons, Ilion, N. Y.

From 1863 to the close of the war, Mr. Billings was making drop forgings for the U. S. Government. He was one of the first men in the country to use drop hammers in the manufacture of arms, and it was he who perfected the forgings that were universally used in the manufacture of pistols for many years afterward.

When the war was at its height and many large contracts for pistols were coming in from the Government, it was found necessary often to run the shops at night. The continual pounding of the drop hammers was a source of great disturbance to the surrounding neighbors and they voiced their indignation in no uncertain terms. It was not long before a delegation composed of the mayor and some prominent citizens waited upon Mr. Billings to complain of the noises that issued nightly from the shop and to demand that they cease. Mr. Billings stated briefly that he had orders to run his shop night and day until the contract was completed. "Orders!" exclaimed the mayor. "Be pleased to understand, sir, that all orders in this town come from me! I should like to ask you, sir, who gave you these orders! " "My orders," replied Mr. Billings, "come from Abraham Lincoln." The noisy drop hammers continued their work until the peace of Appomattox.

At the close of the Civil War, Mr. Billings returned to Hartford, where he became superintendent of the Weed Sewing Machine Co. for three years. In 1869 the Billings & Spencer Co. was organized, C. M. Spencer being the other member of the firm. The company was incorporated to manufacture drop forgings and develop various improvements and inventions in the numerous small parts of machinery which, through the inventive genius of Mr. Billings, became of great value to the world. He was the inventor of a score of articles now in general use, such as drills, chucks, pocket knives, wrenches, etc.

Mr. Billings was a life member of the Society, and one of the first to join the organization, which he did in 1880. He served as vice-president and president from 1893 to 1895.

He was also very active in the civic affairs of Hartford, where for twelve years he was president of the Board of Fire Commissioners. For several terms he served as councilman and as alderman. He was president of the State Savings Bank of Hartford and trustee of the Hartford Trust Co. He leaves a wife, two sons and one daughter.

ALLAN S. BIXBY

Allan S. Bixby, general manager of the National Malleable Castings Co., Indianapolis, Ind., died on June 12, 1920. Mr. Bixby was born in Philadelphia, Pa., in 1870. He was graduated from Rose Polytechnic Institute in 1892 and started work with the Ewart Manufacturing Co. of Indianapolis, now one of the branches of the Link-Belt Co. With one or two unimportant interruptions he continued with that company until 1902, when he resigned his position as superintendent to associate himself in a similar capacity with the Indianapolis works of the National Malleable Castings Co.

Mr. Bixby's earlier work was primarily in the design and construction of automatic tools and similar equipment. In later years his attention was diverted toward plant management, industrial relations, etc., in which field he did noteworthy work just before and during the War.

In 1919 he was awarded the degree of M.E. by his Alma Mater on his professional record. He became a member of the Society in 1907. He was also a member of a number of local organizations, including the Indianapolis Chamber of Commerce, to the Industries Committee of which he gave much time and attention.

JOHN ALFRED BRASHEAR

Ending a long and active career of notable achievement, Dr. John A Brashear, famed throughout the world as a scientist and maker of precise astronomical instruments, and known to a multitude of devoted friends as "Uncle John," died at his home in Pittsburgh on April 8, in his eightieth year.

During his later years he was the recipient of several honorary degrees, but when asked at one time what degree he would prefer, he replied, "Doctor of Humanity," which fitly expresses the crowning characteristic of his life, evident in all his relations with his fellow-men. It would be difficult to recall the name of any person in recent years who has received the outward expression of real devotion from so many men, women and children, of all stations of life and of all nationalities, as did John A. Brashear. On several anniversaries he received testimonials of appreciation and respect, literally by the thousand, and it is a source of gratification that he could have lived to receive such abundant recognition of the value attached to a life of service.

He was born at Brownsville, Pa., November 24, 1840, attended the Brownsville common school, learned patternmaking, and at the age of 22 was established, apparently for life, as a millwright in a Pittsburgh rolling mill. He remained in millwork for 20 years, after which he engaged in his life work of astronomical and physical instrument making.

From his earliest boyhood the study of astronomy had been his hobby. A traveling astronomer exhibiting the wonders of the heavens to passersby at five cents a look gave to John Brashear, then a lad of seven, his first glimpse of the wonders of the heavens. This made a deep impression on his young mind and he began to dream and study about the stars. He longed for a telescope of his own, and when he became married early in his industrial life he and his wife set out to make one. They fitted a workshop with a small engine and lathe at their humble home on the south side of the city where they made the tubes and ground the lenses, often working until midnight in absorbed joy in the work after a long, hard day at the mill and household tasks. Their first glass took three years to grind. Dissatisfied with its limitations, they set to work on a larger one only to have it break in the silvering process after two years of patient grinding and correcting.

The next night Brashear returned from the mill completely discouraged. His wife was not in the kitchen, so he wandered out into the workshop, where he found a fire under the boiler, steam up and a fresh piece of glass in the lathe. "We'll make a new one," his wife said, and they did. This proved to be the turning point in Brashear's life and was the beginning of his really great work in the field of science. Through the interest which he attracted from Langley, of flying-machine fame, then head of the Allegheny Observatory, he was enabled to move to Allegheny and set up a shop for lens grinding — the beginning of the firm of John A. Brashear Company, Ltd., known today wherever astronomical instruments are used. Here have been made many of the most important optical parts in the world. It has been the policy of the firm to have no patents and no secrets. Whatever it accomplished it has given freely to the world. His son-in-law, Mr. James B. McDowell, joined him in the work and to him Dr. Brashear gave a large share of credit for the success of the undertaking.

Of the products of the Brashear Company reference should be made to the following:

A 72-in. parabolic mirror, Dominion Observatory, Victoria, Canada.

Two 37-in. parabolic mirrors for Lick Observatory, Mt. Hamilton, Calif., and University of Michigan, Ann Arbor, Mich.; and one 30-in. parabolic mirror for Allegheny Observatory, Pittsburgh, Pa.

Several plain and parabolic mirrors and object glasses ranging from 18 in. to 30 in. in diameter for Yale University, New Haven, Conn.; Alleghemy Observatory, Pittsburgh, Pa.; Swarthmore College, Swarthmore, Pa.; Chabot Observatory, Oakland, Calif.; Dominion Observatory, Victoria, Canada; and University of Pennsylvania, Philadelphia, Pa.

Some twenty-five object glasses, ranging from 9 in. to 18 in. in diameter, for various other observatories throughout the world.

A 16-in. photographic doublet for the Königstahl Observatory, Heidelberg, Germany, and a 10-in. photographic doublet for Bruce Telescope, Yerkes Observatory, Williams Bay, Wis.

Perhaps his most important achievement was in connection with the design and development of the spectroscope for astronomical uses. Many of the spectroscopes in the principal observatories in the world have been made in the Brashear workshop, including that for the 38-in. telescope of the Lick Observatory. The excellence of the work done by Professor Keeler at this observatory is freely attributed to Dr. Brashear's skill and genius.

In 1884 Dr. Brashear was commissioned by Professor Rowland of Johns Hopkins University to make the speculum metal plates for his diffraction gratings. Mr. Ambrose Swasey, in his presidential address before the A.S.M.E. in 1904, referred to the extraordinary accuracy of these plates and of others made by Dr. Brashear for Professor Michelson in connection with the determining of the standards of length. Mr. Swasey said:

The Rowland engine was made especially for ruling diffraction gratings of speculum metal, and with it a metal surface has been ruled with 160,000 lines, there being about 29,000 to the inch, and as many as 43,000 lines to the inch have been ruled.

The gratings mostly used have from 14,000 to 20,000 lines to the inch, and with such exactness is the cutting tool moved by the screw that the greatest error in the ruling does not exceed one millionth of an inch.

The production of these gratings, which has enabled the physicist in his study of the spectrum to enter fields of research before unknown, has not only called for the highest degree of perfection ever attained in the spacing of linear distances, but it has also called for a refinement most difficult in the optical surfaces upon which the lines are ruled. To Mr. Brashear was given the problem of producing such surfaces, and notwithstanding the many difficulties encountered in working and refining the speculum metal plates he has made many hundred plates with surfaces either flat or curved with an error not to exceed one-tenth of a wave length of light, or one four-hundred-thousandth of an inch.

As the established standards of length, which are the yard of Great Britain and the meter of France, are made of metal, and liable to destruction or damage, Professor Michelson conceived the idea of determining the lengths of these standards in wave lengths of light, which would be a basis of value unalterable and indestructible.

For the purpose of carrying out these experiments, the interferometer was constructed, an instrument which required the highest order of workmanship and the greatest skill of the optician. Again Mr. Brashear proved to be equal to the occasion, and made for the instrument a series of refracting plates, the surfaces of which were flat within one twentieth of a wave length of light, with sides parallel within one second. This was the most difficult work ever attempted in the refinement of optical surfaces.

No event in Dr. Brashear's life gave him greater satisfaction, nor elicited from him greater enthusiasm, than the completion of the 72-in. glass for the Dominion Observatory, and its safe arrival at Victoria. The disk of glass was cast in Belgium and weighed when completed for shipment from Pittsburgh 2¼ tons. The final grinding and polishing was done by hand in the basement of the Brashear workshop, where temperature conditions could be kept constant and where the great tube with its double walls was located for making the final tests. The correcting of the surface had to be done by hand by local polishing. After working on the glass for a few hours it had to be lifted to the testing tube where it remained for 10 hours to attain uniform temperature. When the glass finally arrived at Victoria in its special car Dr. Brashear wrote: "The glass reached its destination in less than six days after I bid it good-bye on the car in Pittsburgh, a trip of 2200 miles. You bet Uncle John and his son-in-law and some others are happy over this news, after three years and eight months." The Dominion Observatory was dedicated on June 11, 1918.

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

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

Dr. Brashear was elected to membership in The American Society of Mechanical Engineers in 1891 and was made an honorary member in 1908. He was a manager of the Society from 1899 to 1902. In 1911 he was elected one of the Society's representatives on the John Fritz Medal Board to serve for four years. He was president of the Society for the year 1915, during which time he twice toured the country, visiting the sections and speaking before many engineering and other organizations. His magnetic personality made him always in demand as a speaker. Later he addressed large audiences of the Society on Photography of the Stars and once on The Science of the Beautiful in Commonplace Things.

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

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

A few years ago he was voted Pennsylvania's foremost citizen by a committee of prominent men appointed by the governor of the state. Governor Brumbaugh had been asked by the officials of the Panama-Pacific Exposition to name the most distinguished citizen of Pennsylvania. Having consulted men of prominence throughout the state, the Governor unhesitatingly named John Brashear. When word of this honor was brought to "Uncle John," he was at work in his laboratory and his only comment was: "I guess you're joking." Then he went on with his work.

Dr. Brashear's seventy-fifth birthday was celebrated with a dinner that will long be remembered in Pittsburgh. Alexander Graham Bell, Charles M. Schwab, Henry Clay Frick, Rear-Admiral Robert E. Peary and many other men of high achievement joined with Uncle John's neighbors to pay him homage. One thousand people sat down at dinner and the table was piled high with telegrams of congratulation — from the president of the United States, from men of science all over the world and from thousands of his less famous but as well-loved nieghbors. That night Dr. Brashear's friends made up a purse of fifty thousand dollars, the interest of the fund to go to him as long as he lived and after that "to carry on, but not to take the place of," the work he had begun.

Scarcely less notable was a public reception tended him at Pittsburgh on the eve of his seventy-sixth birthday and of his departure for an extended trip to the Orient, with Messrs. Ambrose Swasey and John R. Freeman, Past-Presidents A.S.M.E. Concerning this, the following extract from a letter written at the time by his secretary, Miss Martha C. Hoyt, gives a just estimate of the affection in which he was held by the people of his own city.

It was a truly wonderful sight such as Pittsburgh will probably never see again. In the receiving line were the Jew and the Christian, Protestant and Catholic, well-known public men and private citizens, the rich and the poor. There were even people whom the doctor has rescued from privation. In the line which came to greet him were the most prominent educators, social and financial leaders, down to the dirty little newsboy with his papers under his arm, who conducted himself during his progress down the receiving line with as much dignity as the more experienced members of society. There was a deaf and dumb man who knew him when a young man and brought his congratulations and best wishes written on a paper so he would not take too much of Dr. Brashear's time presenting them, and an Italian laborer who was attending night school in the neighborhood and came running in to say what his teacher had taught him for the occasion: "I wish you many happy returns of the day." He was radiant with his success of being understood. There were delegations from the Carnegie Technical Schools, the University of Pittsburgh, the School for the Blind, the Newsboys Home, and the Loendi Club, a group of colored men in whom he has been interested and who have had their share of his loving service.

Among the many benevolent acts which were typical of the life of Dr. Brashear was the founding three years ago of the Brashear Settlement Association which has its headquarters in the old Brashear home, where the doctor spent his younger years. He suggested that the house should be called "Inspiration," for the humble old-fashioned structure with the rude shop in the rear would point out to these lads that from humble beginnings one may rise to greatness through hard work.

Upon the last celebration of the anniversary of his birth in November, at his request, money which would have been used by his numerous friends to purchase gifts for him was given to the Brashear Settlement to enable it to further the work which it had instituted. It was Dr. Brashear's wish that the work should live after his death, and acting upon this the officers of the Association will endeavor to make it a permanent tribute to his name. On this same anniversary a "birthday shower" of postal cards was arranged by the officers of the Brashear Settlement House and it was planned and carried out for thirty thousand Pittsburghers to send cards of greeting to Uncle John. At this time Dr. Brashear was confined to his home by illness.

Memorial Hall, Pittsburgh, was the scene of Dr. Brashear's funeral services, which were simple and carried out in accordance with his wishes expressed shortly before his death. Thousands from all walks of life were present to show the love they had for "Uncle John." Not only was the great hall filled, with hundreds standing, but many hundreds more for whom there was no room lingered outside. The ashes of Dr. Brashear, together with those of his wife Phoebe Stewart, who died a few years ago, will rest in the base of the great telescope erected in Allegheny Observatory. The inscription, written by Dr. Brashear, reads, "We have loved the stars too fondly to be fearful of the night."

LOUIS L. BRINSMADE

Louis L. Brinsmade was born in Elmira, N. Y., in 1875. He received his early education abroad. He returned to this country and entered Washington University, from which he was graduated in 1896. The following year he received his M.M.E. from Cornell University. For short periods he was connected with the St. Louis Water Department, the Bell Telephone Co. and Westhinghouse, Church, Kerr & Co. He obtained his shop experience with the Missouri Car Railway Co. He then returned to Westinghouse, Church, Kerr & Co., where he was located for about six years in their engineering department. He resigned from this position to take over the management of the sales and construction of the New York office of the Westinghouse Machine Co. He was with this firm as manager of the New York office until 1915, when he entered business for himself in New York City. He died on June 16, 1920.

Mr. Brinsmade became a junior member of the Society in 1897 and was promoted to full membership in 1906. He also belonged to a number of social clubs in the city.

JOSEPH J. BUTCHER

Joseph J. Butcher, consulting engineer, Victoria, B. C., Canada, and a member of the Society since 1892, died in 1920. Mr. Butcher was born on August 30, 1849, in Liverpool, England. He attended high school and the Liverpool Institute and served his apprenticeship with the Smith & Carvers Co. From 1872 to 1876 he was employed in the drafting rooms of Fawcett, Preston & Co., and of Clark, Chapman & Gurney. His shop experience was gained with the same firms and later he became superintendent of machinery for the companies. From 1882 to 1887 he was a partner in the firm of Norman C. Cookson and Thomas Gray, Newcastle-on-Tyne, where his work was chiefly experimental, dealing with gas engines and gas apparatus.

Mr. Butcher's first position in this country was as engineer with the Abendroth & Root Manufacturing Co., New York City. In 1892 he became associated with the Guiler-Gordon Engineering Co., Boston, Mass., as consulting engineer. At the time of his death Mr. Butcher was engaged in consulting work in Victoria, B. C., Canada.

CHARLES B. CALDER

Charles B. Calder, vice-president and general manager of the Toledo Shipbuilding Co., Toledo, Ohio, died on January 23, 1920. Mr. Calder was born on January 15, 1853, in Antwerp, N. Y. In 1866 his father went into partnership in a shingle-mill venture and the son ran the engine in the mill until 1872. In that year he shipped as cook on a little schooner and later obtained a second engineer's license. In the following years he shipped on many vessels and in various capacities, spending all his spare time in studying for advancement.

In 1891 he accepted the position of chief engineer with the Menomee Transportation Company's steamers, and in 1894 became superintendent of the Dry Dock Engine Works, Detroit, Mich., holding this position until 1899, when he was appointed general superintendent of the Detroit Shipbuilding Co. In the fall of 1905 he resigned to become one of the incorporators of the Toledo Shipbuilding Co., of which he was made general manager. In 1909 he was elected to the position of vice-president.

Mr. Calder was a member of the American Society of Naval Engineers and of the American Society of Naval Architects and Marine Engineers. He also belonged to several clubs and fraternal associations. He became a member of our Society in 1895.

JOSEPH J. CARLOTTI

Joseph J. Carlotti, equipment engineer for J. E. Musselman, consulting engineer, New York, died on May 23, 1920. Mr. Carlotti was born in 1892 in New York City and was educated in private schools. He entered the employ of J. E. Musselman as junior draftsman and received rapid advancement. As chief draftsman he was responsible for the design of the electrical and mechanical equipment of many of the public buildings in the city. During the War he was assistant to the chief of the electrical and mechanical section of the U. S. Navy Yard in Brooklyn, N. Y., returning to his former position in 1919.

Mr. Carlotti became an associate member of the Society in 1919.

ALBERT P. CHAPMAN

Albert P. Chapman was born in Springfield, Mass., on February 5, 1883. He attended the Springfield schools and later entered Worcester Polytechnic Institute, receiving his B.S. degree in 1907; the following year he took post-graduate work at the school and received his E.E. degree.

Mr. Chapman was first employed by the General Electric Co., Schenectady, N. Y., where he had charge of various shop tests on motors and generators. For a few months in 1909 he worked for H. H. Richards & Co., Hartford, as assistant to the head draftsman on the design and layout of a complete atlas of the city. He left that firm to take a position with the Public Service Railway Co., Newark, N. J., as cadet engineer in all branches of the mechanical department, remaining there until 1910, when he entered the electrical engineering department of the Boston Elevated Railway Co., as electrical assistant in the designing, laying out and supervising of the construction of a. c. and d. c. feeder lines. From Boston Mr. Chapman went to Lowell, Mass., to become electrical engineer at the Boott Cotton Mills, in charge of all construction and maintenance work. In October, 1914, he became connected with the Ludlow Manufacturing Associates, Ludlow, Mass., as electrical engineer; in 1917 he became superintendent of power and repairs, which position he held at the time of his death, March 18, 1920.

Mr. Chapman became an associate member of the Society in 1917. He was also a member of the American Institute of Electrical Engineers and the Engineering Society of Western Massachusetts.

FRED PERCY CLEVELAND

Fred P. Cleveland, secretary and treasurer of B. F. Perkins & Sons, Holyoke, Mass., died at his home in that city on February 10, 1920, after a brief illness of influenza followed by pneumonia.

Mr. Cleveland was born in Holyoke, Mass., on January 20, 1885, and was educated in the schools of that city. He later attended Cornell University, from which he was graduated in 1906 with the degree of M.E. Immediately upon graduation he became associated with B. F. Perkins & Sons, Inc., and for several years has been the active manager of the company. Under his management the business developed greatly, necessitating the construction of a new plant in Willimansett. Mr. Cleveland was very much interested in this project, and his unceasing work to obtain its accomplishment without doubt weakened his vitality.

Mr. Cleveland was a member of the Engineering Society of Western Massachusetts. He became an associate member of our Society in 1914.

DEAN BERNARD COBB

Dean B. Cobb, head of the drafting department and in charge of the evening electrical course of the Boardman Trade School, New Haven, Conn., died on April 1, 1920. Mr. Cobb was born on June 11, 1891, in West Haven, Conn. He was educated in the West Haven grammar and high schools and later took courses in mechanical drawing and electrical engineering in night and correspondence schools. From 1909 to 1919 Mr. Cobb was connected with the following firms, all in New Haven: New York, New Haven & Hartford Railroad as electrical draftsman; the Acme Wire Co. as chief electrician; the Southern New England Telephone Co. as chief draftsman; Westcott & Mapes Co. as assistant mechanical engineer and the Greist Manufacturing Co. as plant engineer. In 1919 Mr. Cobb became associated with the Boardman Trade School, with which he was connected at the time of his death.

Mr. Cobb was a member of the American Institute of Electrical Engineers. He belonged also to several fraternal organizations. He became a junior member of our Society in 1919.

MITCHELL COFFIN

Mitchell Coffin, member of the firm of S. M. Ryder & Son, Niagara Falls, N. Y., died on February 6, 1920. Mr. Coffin was born on September 27, 1889, in Brooklyn, N. Y. He was graduated from the Massachusetts Institute of Technology in 1912, receiving his B.S. in mechanical engineering. From graduation until the time of his death, he was a member of the firm of S. M. Ryder & Son, engaged in the designing and manufacture of machinery.

Mr. Coffin became a junior member of the Society in 1913.

JOHN JOSEPH CONVERY

John J. Convery was born on February 23, 1871, in Worcester, Mass., and received his early education in the schools of that city. He followed the trade of machinist until he was twenty-three years old, when he took up the study of mechanical engineering.

He secured employment with the Washburn & Moen Co., Worcester, as a machinist. Showing his ability at designing, he was transferred to the drawing room of that firm. His task was designing improvements on machines for the manufacture of all kinds of springs. While in this department, he started his work of designing a grinding apparatus which insured more accuracy and a greater output. He designed the first spring-coiling machine to coil and square the base coil of a spring in the same operation. He also designed the first automatic spring-cutting machine. Through these and many other machines and appliances of his design, the Washburn & Moen Co., now the American Steel & Wire Co., took a leading place in the spring market of 1902.

To obtain experience in rolling-mill design, Mr. Convery went to Pittsburg, where for five years he was connected with the Heyl & Patterson Co. and the National Rolling Mill. He then became associated with the Midvale Steel Co., Philadelphia. In 1910 he designed an improved automobile-tire machine for wrapping the tire on the carcass automatically, an accomplishment thought impossible up to that time. He assigned his interests in these machines to the

Goodyear Tire Co., and supervised their construction. About 1912 he opened offices in New York City as consulting engineer and was engaged in this branch of the profession at the time of his death, April 17, 1920.

Mr. Convery became a member of the Society in 1916.

ALFRED W. COOKSON

Alfred W. Cookson was born on September 13, 1868, in Birmingham, England, and was educated in the schools of that city. He served an apprenticeship to James Archdale & Co., Birmingham, builders of machine tools and special machines. Upon coming to this country he was employed by the Baldwin Locomotive Works, Philadelphia. He also worked for a number of years in Cramp's Shipyard, Philadelphia. Later he went to Hornell, N. Y., where he acted in a consulting capacity to firms interested in silk-knitting and silk-weaving machines. For four years he was associated with the Willys-Morrow Co., Elmira, N. Y., as maintenance and construction engineer. Because of failing health he was obliged to give up active work. He died on January 11, 1920.

Mr. Cookson became an associate of the Society in 1917.

WILLIAM WALTER DINWIDDIE

William W. Dinwiddie, research and experimental engineer, Laboratory of Thomas A. Edison, Inc., Orange, N. J., died on October 6, 1920. Mr. Dinwiddie was born on November 4, 1876, in Greenwood, Va. He received his early education in the home schools and later attended the University of Virginia, from which he received the degrees of A.B. and M.E. For several years he was connected with the U. S. Naval Observatory as computer and during this period he was a member of eclipse expeditions to Georgia (1900), Sumatra (1901) and Africa (1905). From 1906 to 1908 he was connected with the Ames-Bonner Co., Toledo, Ohio, as superintendent. At the close of that period he became general manager for the Alvan Clark & Sons Corporation, Cambridgeport, Mass., manufacturers of astronomical instruments. For the past eight years Mr. Dinwiddie was associated with the Thomas A. Edison Co., Inc., first as manager of the disk record division and subsequently as research and experimental engineer.

Mr. Dinwiddie was a member of the masonic order. He became a member of the Society in 1915.

JOHN F. AND HORACE E. DODGE

Dying within a year of each other, these two brothers. who had established the well-known automobile firm of Dodge Brothers in Detroit, Mich., seemed to prove their inability to live one without the other.

Born three years apart, John being the elder, no two men of the same blood could have been more alike. Theirs was a joint leadership through life. From high school in Niles, Mich., their birthplace, they entered their father's machine shop. Their father was a mechanic of the old school and it required no driving on his part to make his boys good mechanics. Their boyhood was commonplace, but from the first it was apparent that whatever career was to be theirs, it would be Dodge brothers always, and so it was.

Always ambitious, the brothers worked first in the Niles and later in the Port Huron machine shops of their father, but the opportunity offered seemed so limited that in 1886 they went to Detroit and became connected with the Murphy Engine Co., working in the shop. Later they went to Windsor and became machinists in the Dominion Typograph Co. plant. In time they were able to lease this business. Selling out to a Canadian bicycle concern, the brothers returned to Detroit to open a machine shop. Their business developed slowly but they persevered. When the Olds Motor Works gave them a contract for 3000 gear sets which they handled successfully, it proved to be the real beginning upon which the brothers reared their structure. Their work came to Mr. Ford's attention, who, at that time, was still dreaming of the popular-priced automobile in every home, and it does not seem unreasonable to say that the workmanship and honesty of purpose of the Dodge brothers helped to lay the foundation of his success.

When they went into business for themselves, they were prepared but with a preparation of their own making, and their success was, in a measure, due to their marked ability to pick working associates and not mere subordinates.

In their philanthropies there was an element of unspoken agreement. John F.'s charities gravitated to the individual, while those of Horace were of a more organized character. Both brothers had a profound dislike of ostentation and consequently their many gifts and charities will probably remain unknown.

John F. Dodge died in New York in January, 1920, of pneumonia and Horace E. Dodge died in Florida in December of the same year after an illness which dated back to his brother's death. Both brothers were members of the Society, joining in 1914 and 1915 respectively.

PHILIP DROSS

Philip Dross, vice-president and general manager of the Marty Foundry Co., Meridian, Miss., died in June 1919. Mr. Dross was born in March 1882 in Belleville, Texas. He was graduated from the Texas Agricultural and Mechanical College in 1902 receiving both his B. S. and M.E. degrees. The following year he took a special course in the Georgia School of Technology, Atlanta, Ga. From 1903 to 1908 he was connected with the Alabama & Vicksburg Railway Co., the Vicksport, Shreveport & Pacific Railway Co. and the New Orleans & Northeastern Railway Co. first as chief draftsman in the mechanical department and later in charge of the motive power and rolling stock.

In 1908 Mr. Dross founded and organized the Marty Foundry Co. and was himself directly in charge of all production and molding equipment, the designing and construction of conveying equipment and boiler installation, steel structures, tank towers, freight cars, etc. In 1912 the company was incorporated and he became vice-president and general manager in charge of all sales and production.

Mr. Dross became an associate member of the Society in 1917.

SAMUEL H. EDWARDS

Samuel H. Edwards, manager of the Tide Water Oil Co., Bayonne, N. J., died on August 4, 1920, of heart trouble. Mr. Edwards was born in Pottsville, Pa., on January 12, 1858. He was educated in the grammar schools of Albany,

N. Y., and at an early age started work for his father, who was a pioneer oil refiner in that city. He later went to Williamsport, Pa., where a refinery was erected of which his father was superintendent.

Mr. Edwards was in the service of the Tide Water Oil Company interests for 38 years. He entered the employ of the Chester Oil Company in 1882, and was sent to take charge of the can and case department of Lombard, Ayres & Company at Bayonne, N. J., in 1888. Later that same year the Chester Oil Co., the Lombard Ayres Co., and several other oil companies were merged in one under the name of Tide Water Oil Company. In 1890 he was sent to superintend the erection and operation of the iron-ore mines for the Benson Mines Company, New York. In 1897 he returned to the Tide Water Refinery as mechanical superintendent. In 1901 he was sent to superintend the shipyard of Townsend and Downey Shipbuilding Company at Shooters Island, N. Y. In 1903 he again returned to the Tide Water Refinery, resuming the position of mechanical superintendent. In 1906 he started the building of oil-pipe lines and pump stations for the Tide Water Pipe Company from Bradford, Pa., to Stoy, Ill. He finished this work in 1909, returning to the Refinery at Bayonne. Upon the death of his brother Frank W. Edwards in 1911, he was appointed general superintendent of the Bayonne Plant, being advanced in 1917 to position of manager. He was also a member of the Board of Directors of the Company.

Mr. Edwards became a member of the Society in 1912. He belonged to a number of clubs and was interested in the civic affairs of Bayonne.

PRESLEY J. FICKINGER

Presley J. Fickinger, chief engineer of the Pittsburgh & Conneaut Dock Co., Conneaut, Ohio, died on December 11, 1920. Mr. Fickinger was born on January 12, 1858, in Girard, Pa. It was in Kingsville, Ohio, however, that Mr. Fickinger gained his elementary education, later receiving further mechanical training in Syracuse, N. Y.

For four years he worked with the Phœnix Iron Works at Ashtabula, Ohio, leaving to become connected with the Straight Line Engine Co., Syracuse, N. Y., where he became closely associated with Prof. John E. Sweet, Past-president Am.Soc.M.E. Mr. Fickinger's entire business life was linked with the steel industry. Twenty years ago he assumed the position of master mechanic on the local docks in Conneaut. He became an authority on mechanics and advanced rapidly through the ranks till he became chief engineer of the Pittsburgh & Conneaut Dock Co. Mr. Fickinger was associated with George Hulett of Cleveland in the perfecting of the first automatic ore-unloading machinery, and at his death held an interest in the Hulett patents. He also held several patent rights in his own name.

Mr. Fickinger became a member of The American Society of Mechanical Engineers in 1888. He also belonged to several clubs and organizations in Conneaut and elsewhere, including the masonic fraternity.

HOBART D. FRARY

Hobart D. Frary, assistant professor of steam and gas engineering, University of Wisconsin, died on August 15, 1920. Professor Frary was born in Minneapolis, Minn., on April 28, 1887. He was educated in the Minneapolis schools and received his M.E. from the University of Minnesota in 1908 and his M.S. in 1909. He studied in the University of Göttingen in 1913–1914 and in 1918 received his Ph.D from the University of Illinois. Professor Frary had held the positions of instructor in engineering mathematics, University of Minnesota, mechanical engineer and magnetic observer, yacht *Carnegie*, Carnegie Institution; instructor in mathematics, University of Iowa, and engineer in the Forest Products Laboratory. He became connected with the University of Wisconsin in 1919.

Professor Frary was a member of the honorary societies of Tau Beta Pi and Sigma Xi. He became an associate-member of our Society in January, 1920.

GEORGE GILES

George Giles, general manager and director of the Vancouver Engineering Works, Ltd., Vancouver, B. C., died on April 4, 1920. Mr. Giles was born on July 8, 1869, in Walthamstow, London, England. When a mere boy he left England and went to Canada and there he secured employment with the Reid & Currie Co., and their successors, the Vulcan Iron Works, Westminster, B. C., as "improver" in the foundry and machine shops. He worked up to the position of assistant manager. In 1898 he became connected with Armstrong & Morrison, Ltd., Vancouver, as assistant to the superintendent. Two years later this firm was bought by the Vancouver Engineering Works and Mr. Giles became assistant to the manager with full charge of estimating, costs and production. In 1906 he was promoted to the position of works manager and in 1915 became general manager and director of the firm.

Mr. Giles belonged to several clubs and fraternal organizations. He was also the honorary president of the Metal Trades Employers' Association of British Columbia, and a member of the American Foundrymen's Association. He became a member of our Society in 1908.

NATHANIEL OLIVER GOLDSMITH

Nathaniel Oliver Goldsmith, assistant to the president and in charge of operations of the Weir Frog Co., Cincinnati, Ohio, died on February 16, 1920 Mr. Goldsmith was born in Cincinnati on February 27, 1860. He was educated in the schools of that city and Lehigh University, receiving his M.E. degree from the latter in 1883. Upon graduation he entered the drafting room of the Weir Frog Co., with which concern he was connected for thirty-seven years, and through successive promotions attained the position which he held at the time of his death.

Mr. Goldsmith belonged to several clubs in Cincinnati. He became a member of the Society in 1900.

ALEXANDER MCDONALD GRAVER

Alexander M. Graver, vice-president and assistant general manager of the Graver Corporation, East Chicago, Ind., died on January 31, 1920, of pneumonia. Mr. Graver was born on May 15, 1883, in Allegheny, Pa. He attended the University of Michigan, where he received the degree of B.S. in 1905 and

M.S. in mechanical engineering in 1906. Upon graduation he became actively associated with the Graver Corporation. His first duties dealt with engineering on plant equipment. Later he became purchasing agent for the firm, and then sales manager, which position he held until January 1, 1920, when he was appointed vice-president and assistant general manager.

Mr. Graver was a member of several fraternal organizations. He became a junior member of the Society in 1908.

JOSEPH W. HARTMAN

Joseph W. Hartman was born in Lyons, N. Y., on April 8, 1890. He received his education at the Lyons Union School and was graduated in 1912. He then entered Rensselaer Polytechnic Institute, from which he received his M.E. degree in 1916. His first professional work was with the Cambria Steel Co. at Johnstown, Pa., where he spent one year. He was with the Corning Cut Glass Co. for three years and in May, 1920, entered the employ of the Gleason Works, Rochester, N. Y., where he was located at the time of his death.

Mr. Hartman, who became a junior member of the Society in 1917, met death very suddenly by drowning in Sodus Bay, N. Y., on July 3, 1920.

CARL JEFFREY HEATH

Carl Jeffrey Heath, who became a junior member of the Society in 1919. died on April 4, 1920. Mr. Heath was born in Winsted, Conn., on April 2, 1895, He was educated at Stockbridge high school, Andover Academy and the Massachusetts Institute of Technology, receiving from the latter his B.S. degree in 1917. Upon graduation he became connected with the Boston office of R. Hoe & Co. as salesman. He resigned from this position in 1918 to enter the United States Army. Upon his discharge in 1919 he became associated with the General Electric Co., Pittsfield, Mass., in the motor-engineering department, where he was located at the time of his death.

ERNEST GEORGE HENDERSON

Ernest G. Henderson, general manager and president of the Canadian Salt Co., Ltd., Windsor, Ont., died on October 20, 1920. Mr. Henderson was born in Holywood, County Down, Ireland, on September 14, 1858. He was educated at the Hillbrook School and College, Belfast, and was apprenticed for four years as pupil to the engineer of the Belfast and County Down Railway. He was then appointed assistant engineer, Ulster Division, Great Northern Railway of Ireland. In 1883 he went to Canada and engaged in surveys and construction of the main line of the C. P. R. Later he was appointed assistant engineer, C. P. R., Toronto, and then resident engineer, C. P. R., London, Ont. He resigned in 1883 to erect the plant of the Canadian Salt Co., then known as the Windsor Salt Co., of which concern he became vice-president and general manager, and finally president in 1917.

Mr. Henderson was a member of the Civil Engineers of Ireland, the Canadian Society of Civil Engineers, the American Chemical Society, the Society of Chemical Industry and ex-president of the Canadian Manufacturer's Association. He became a member of our Society in 1916.

JAMES AMORY HERRICK

It is with especial regret that we are obliged to record the death of James Amory Herrick, one of the original members of the American Society of Mechanical Engineers, which he joined in 1880.

James Amory Herrick was born at Nashua, N. H., on January 17, 1850, and died at Merrimack, N. H., on September 10, 1920, in his seventy-first year.

His parents were the late Moses Augustus Herrick and the late Jane (Riply) Hubbard, formerly of Winchester, Mass. He was of Puritan-Pilgrim stock, his ancestry going back to Henry and Editha (Laskin) Herrick, of Salem, Mass., who came to this country from England in 1624, while on his mother's side he was directly descended from John and Priscilla (Molines) Alden, passengers on the *Mayflower*, who landed at Plymouth, Mass., in 1620.

His education was obtained in the public schools and the Massachusetts Institute of Technology, graduating in 1872 and taking a post graduate course in 1873.

Mr. Herrick's first work was at the former Nashua Iron and Steel Co., where he built the first 15-ton open-hearth furnace in this country in place of the original 5-ton furnace at these works. He also installed and used the first gas producer (Siemmens) which was imported to this country from England by his father, then treasurer of the above company, employing the principle of the direct gasification of low-grade coal to melt the steel and continued in charge of the production of steel for this company until 1879.

He then went to Pittsburgh, Pa., designed and erected a 17-ton open-hearth steel plant, together with a 17-ton hammer plant, for Park Brothers and Co., also completed an open-hearth steel plant and rolling mills, designed by Alexander L. Holley, New York, in which were installed the first mechanically operated furnaces of the Pernot System in this country.

In 1882 he joined the engineering force of H. A. Gadsen and Co., New York, and designed and erected the Standard Steel Castings Co.'s factory at Chester, Pa., which is now the largest plant of the American Steel Foundries Co. in the Eastern States. In 1884 he was sent abroad to erect some special furnaces at London, England, and to visit and collate information from many steel plants in Great Britain and on the continent.

During 1885 he returned to New York and in 1886 succeeded the firm of Gadsen and Co., remaining in business under his own name, also being associated with H. L. Dixson and Co., Pittsburg, Pa., and David Townsend, Philadelphia, Pa., and making his headquarters at New York under the style of a mechanical and gas engineer. He retired from business about two years ago, at which time he sold his patents and business, his health having failed, to the Smith Gas Engineering Co., of Dayton, Ohio.

Among the plants that Mr. Herrick designed and erected were the John Illingworth Steel Co., Philadelphia, Pa., Diamond Drill and Machine Co., Birdsboro, Pa., Congdon Brake Shoe Co., Chicago, Ill., National Steel Casting Co., New Haven, Conn., Republic Enameling and Stamping Co., Buffalo, N. Y., Allison Tube Co., Philadelphia, Pa., El Paso Steel Casting Co., El Paso, Tex.,

N. E. Steel Casting Co., Everett, Mass., Longview Steel Casting Co., Longview, Tex., Century Steel Co., Poughkeepsie, N. Y., Lloyd's Tube Mills, London, England, and Mariopol Tube and Furnace Mills, Mariopol, Russia.

Mr. Herrick also specialized in the Herrick Gas Producers, both static and mechanical, for use with both anthracite and bituminous coal, furnaces both open-hearth and regenerative, gas- and air-reversing valves and various accessories and has made many important installations of these specialties, one item alone including over 500 Herrick Gas Producers during his business career.

He was a prolific inventor in his specialty, having taken out over forty United States patents, many of which covered basic principles, and also many foreign patents, and was rated as the dean of the producer gas design and business in this country. His passing is a distinct loss to the engineering profession in his chosen specialty, where he will always be well and favorably remembered. In person he was kind, genial and of a specially lovable disposition which made many true and tried friends for him and whom he retained throughout his life.

He was a member of the American Society of Mining Engineers, joining them in 1873 and contributing a number of articles for their transactions; also was a member of the Technology Clubs at New York and Philadelphia. He was a member of the order of Knights Templars, affiliated with St. John's Commandery, No. 4, of Philadelphia, Pa.

Mr. Herrick leaves no immediate family, his wife, Mary Ada (Davis) and two sons, Edward A. and Cecil, having predeceased him. Interment was made at Bayside Cemetery, Lakeport, N. H., in his family lot beside his wife and sons. A sister, Mrs. Amelia (Herrick) Pond, and three brothers, William H., Rufus F., and Charles H. Herrick, all residing at Winchester, Mass., survive him.

Mr. Herrick was one of the earliest members of our Society, joining in the year of its foundation, 1880.

CHARLES W. JOHNSON

Charles W. Johnson, assistant director of engineering of the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa., died suddenly of pneumonia on April 21, 1920. Mr. Johnson was born in June, 1874, in New Vienna, Ohio, where he received his early education. Later he entered Ohio State University, from which he was graduated in 1896, with the degree of M.E. Upon graduation he entered the employ of the Steel Motor Co., Johnstown, Pa., where he was located for about two years, when he became connected with the Bullock Electric & Manufacturing Co., Cincinnati, Ohio; seven years later he was made superintendent of the works of the Allis-Chalmers-Bullock Co., Ltd., Montreal, Canada.

He entered the employ of the Westinghouse Electric & Manufacturing Co. in 1907, and was soon appointed to the office of chief inspector. Mr. Johnson's promotion was rapid, In 1912 he became general superintendent of the East Pittsburgh Works, and in the early part of 1919 was made assistant manager of works. On January 1, 1920, Mr. Johnson was appointed assistant director of engineering.

Mr. Johnson was also very active in welfare work and social problems. He was a member of the American Institute of Mining and Metallurgical Engineers, the Engineering Society of Western Pennsylvania and several social clubs. He became a member of our Society in 1908.

RICHARD D. KIMBALL

Richard D. Kimball, president of the Richard D. Kimball Co., Boston, Mass., died on November 10, 1920, at Alton Bay, N. H., his summer home. Mr. Kimball was born on September 16, 1847, in Rochester, N. H. His first employment was with the Prentiss knife shops in Greenfield. From 1881 to 1887 he was with the Millers Falls Tool Co., Millers Falls, Mass., as superintendent of the vise shop. While there he invented several attachments for the machines. In 1888 he located in Boston, being interested in a passenger-car heating system which he had invented and patented. Mr. Kimball was also for a short time New England representative for the Johnson Service Co. and in 1897 entered the heating and ventilating engineering business under his own name.

His first large piece of work was the central heating plant at Dartmouth College, which was followed by installations at Groton School, Mount Holyoke College, Wellesley College and many others. Mr. Kimball invented a conduit system of split tile for steam pipes which was patented in 1899 and later sold to the H. W. Johns-Manville Co. His business was incorporated in 1908 under the name of the Richard D. Kimball Co., and his engineering work extended from Canada to Georgia and Kentucky, and west to North Dakota.

Mr. Kimball was a member of the Boston City Club and of several church organizations. He became a member of The American Society of Mechanical Engineers in 1906.

CHARLES OSCAR KLOCKARS

Charles O. Klockars, vice-president and general manager of the Essex Foundry, Newark, N. J., died on September 18, 1920. Mr. Klockars was born in Finland on August 14, 1878, and came to this country at the age of eleven. Two years later he started work at sweeping floors in the tapping room of the Malleable Iron Fittings Co., Branford, Conn. He was advanced to the position of assistant foreman at the age of eighteen. Later he was transferred to the machine shop as machinist and toolmaker apprentice. He worked at these trades for ten years.

After private tutoring he entered Pratt Institute, Brooklyn, N. Y., in 1906 and was graduated in 1908 from the mechanical engineering course. His first professional work was in the engineering department of the Crane Co., Bridgeport, Conn., where he spent two years. In 1910 he entered the employ of Essex Foundry as superintendent and served in that capacity until his appointment in February, 1920, to the office of vice-president and general manager.

He was completing at the time of his death a continuous pouring machine for molding steam fittings, which would increase production 50 per cent. Several other labor-saving machines were designed and built under his supervision at Essex Foundry.

Mr. Klockars became an associate of the Society in 1911. He belonged to a number of clubs and several fraternal organizations.

CHARLES KOPP

Charles Kopp, chief engineer of the Norit department of Joseph Baker Sons & Perkins Co., Inc., New York City, died on October 11, 1920, in Donaldsville, La., while in charge of the erection of new machinery used with the Norit in refining sugar. Mr. Kopp was born in February 25, 1890, in Chicago, Ill. He was educated at the Hebrew Technical School and Cooper Union, New York City. From 1908 to 1918 Mr. Kopp was associated for varying periods with the following firms: the Mosher Water Tube Boiler Co., Westinghouse, Church, Kerr & Co., Gibbs & Hill, E. L. Phillips Co., and the Honolulu Iron Works Co., all of New York City. He was also consulting engineer for the American Burtonizing Co., Staten Island, N. Y., resigning from this firm to take the position he held at the time of his death. Mr. Kopp became an associate-member of the Society in 1919.

JORGE LAGE

Jorge Lage, president Cia Nacional de Navegacão Costeira, Rio de Janeiro, Brazil, died in 1918. Mr. Lage was born in Rio de Janeiro in December, 1882. He was educated in private schools in Switzerland. He returned to Rio de Janeiro and in 1907, and took charge of the machine-shop department of the shipbuilding and repairing yard of Messrs. Lage Trmaos. In July, 1908, Mr. Lage was appointed general superintendent of the steamers of the Compania Nacional de Navagacao Costeira and in 1913 he was elected president of the same firm.

Mr. Lage became a member of the Society in 1914.

JULIUS SHERMAN LANE

Julius S. Lane, for the past eleven years mechanical and consulting engineer with the Engineer Co., New York, died on November 29, 1920. Mr. Lane was born in Akron, Ohio, on November 19, 1841 and received his education in the schools of that city.

Shortly before the Civil War he went to Newark, N. J., to learn the machinist's trade. Upon the completion of his apprenticeship with the Newark Machine Works, he immediately attempted to enlist in the Army but was rejected because of physical disability. He then returned to Akron as a master mechanic in the employ of the Taplin-Rice Machine Works. In 1867 he was called to Ishpeming, Mich., to install a large mining machinery plant for the Lake Superior Mining Co. This was the first attempt in the Lake Superior region to hoist iron ore from a shaft with equipment of any size. As master mechanic of the mine, Mr. Lane made a number of innovations in what was then considered the best practice in iron-mining equipment. A year later he was offered a partnership in a firm in Akron; he accepted the offer and for seventeen years he supervised the building of engines and mining machinery for the Webster, Camp & Lane Machine Co. During this period he invented the Lane band friction hoist as applied to mining machinery.

Mr. Lane made an extended trip to Europe in 1885, visiting mines in Sweden, Germany and France. Upon his return he became associated as partner and general manager with the M. C. Bullock Manufacturing Co., Chicago, manufacturers of mining machinery.

In 1895 he went to South Africa to oversee the erection of machinery for the Johannesburg gold mines and was there at the outbreak of the Boer War. In the succeeding years, as consulting engineer, he visited Canada, Mexico and many of the western states in connection with his mining work.

Mr. Lane became a member of The American Society of Mechanical Engigineers in 1882. He was also a member of the American Institute of Mining and Metallurgical Engineers and of the Ohio Society of New York. He was a member of the New York City Public School Lecture Course and gave many lectures on the gold and diamond mines of Kimberley.

JOHN LANGTON

John Langton, consulting engineer, New York City, died at Inspiration, Ariz., on February 6, 1920. Mr. Langton was born on August 29, 1857, in Toronto, Ont., Canada. He received his early education in Canadian schools and later attended Owen's College, Manchester, England, where he studied mechanical engineering.

From 1876 to 1889 he was connected with the following firms: Portland Co., Northern Railway Co. (Toronto), Brown & Sharpe Manufacturing Co., Electric Tube Co. and the Edison Machine Works. In 1889 he became general manager of the Canadian Edison Manufacturing Co., and the following year manager of the Canadian works of the Edison General Electric Corp. In 1892 Mr. Langton went into consulting work and during the following years he served in that capacity for the Detroit Copper Mining Co., the Moctezuma Copper Co., the Dawson Fuel Co., and the Caranea Consolidated Copper Co. In collaboration with Mr. Charles Separd he did work of a consulting nature for the Moctezuma Copper Co. and the Copper Queen Co., Ariz.: with Mr. F. L. Antisell for the Raritan Copper Works, and with Mr. C. R. Weymouth for the Inspiration Consolidated Copper Co., Arizona.

Mr. Langton became a member of the Society in January, 1920.

SAMUEL A. LINTON

Samuel A. Linton, chief of the estimating department of the Pratt & Whitney Co., Hartford, Conn., was killed in an automobile accident at Milford, Conn., on August 9, 1920. Mr. Linton was born on September 11, 1860, in Hartford, Conn., and received his early education in that city. At the age of seventeen he entered the employ of Pratt & Whitney as an apprentice machinist and tool maker. Twelve years later he became foreman of the machine-tool department of Morris Tasker & Co., Philadelphia. In 1890 he became connected with the S. S. White Dental Co. as foreman of their instrument department. In 1882 he returned to the Pratt & Whitney Co., and with the exception of two years with the C. J. Smith & Sons Co., Milwaukee, and three years with the Ideal Machine Co., Hartford, was with the company until the time of his death. For the past 14 years he held the position of chief of the estimating department.

Mr. Linton became a member of the Society in 1918.

WILLIAM J. LOGAN

William J. Logan, until his retirement in 1914 head of the Logan Iron Works, Brooklyn, N. Y., died on December 15, 1920. Mr. Logan was born in New York City on April 21, 1853. He was a graduate of Cooper Union and became a member of the bar of the State of New York although he never engaged in active practice.

As a boy he was apprenticed to his father who was engaged in the fabrication of steel and maintained a boiler and gas holder works in New York City, conducted under the name of Farrell Logan. Subsequently Mr. Logan became a partner of his father. After the elder Mr. Logan's death, the business was conducted by Mr. William Logan and his brother and was incorporated under the name of the Logan Iron Works.

Mr. Logan was best known for his skill and ability in the fabrication of steel for special purposes, — the construction of tanks, holders, equipment for oil works, sugar houses, bridges, water towers, boilers and gas works. The cutting shields used in underground and tunnel construction, together with air locks and appurtenant mechanical devices, were also features of his work.

Mr. Logan was one of the earliest members of the Society, joining the organization in 1880 and holding a life membership. He belonged also to a number of social clubs in Brooklyn.

JAY HINE LUTZE

Jay Hine Lutze, assistant superintendent, the Bristol Co., Waterbury, Conn., died in December, 1918. Mr. Lutze was born in Denver, Col., in August 1890. He received his technical education in Pratt Institute where he attended the school of science and technology, taking the course in steam and machine design.

He served his apprenticeship in 1906 as a machinist with the Waterbury Farrel Foundry. In 1907 he entered the employ of the Bristol Co. and was assigned to the printing department, but was later transferred to the engineering. department, where he worked for three years before entering Pratt Institute. Upon the completion of his course he returned to the Bristol Co. as a designer of recording instruments and tools for their manufacture. Five years later he was appointed to the position of assistant superintendent.

Mr. Lutze became a junior member of the Society in 1915.

EDMUND MACKENZIE

Edmund Mackenzie, for many years engaged in the design and construction of power plants, pulp and paper mills, and industrial plants in this country and in Norway, died in March, 1919. Mr. Mackenzie was born in Kragers, Norway, in January, 1877. He was graduated from the Skiensfjordens Technical School in 1894 and received shop experience in the Norway and Greater New York Gas Engine & Power Co.

Mr. Mackenzie then came to America and engaged in drafting-room work successively with the Sprague Elevator Company, Westinghouse, Church, Kerr & Co., F. G. Ten Broeck, and the American Brass Company, working on designs and details for power and industrial plants, pulp and paper mills. During the summer of 1908 he designed and estimated structural steel for buildings and bridges for Milliken Bros. of New York. He also assisted in the structural design of the Kings Bridge and Long Island City power stations and was for a time chief draftsman for J. G. White & Co. of New York, and mechanical engineer in Shelburne Falls, Mass.

In 1915 Mr. Mackenzie returned to Norway as supervising engineer of construction work with R. Blakstad, Karl Johansgade, Kristiania, Norway, the position he held at the time of his death. He became a member of the Society in 1909.

ALEXANDER FULLERTON McKENNA

Alexander F. McKenna was born in Belleville, Ontario, Canada, in July, 1870, and was educated in the public schools of that town. He served a five-year apprenticeship in the machine trade and from 1891 to 1893 was connected with the Lloyd & Stinett Engine Works, Erie, Pa., as machinist. For a short time he was with the Erie City Iron Works, when he returned to Canada to work for the Brown Foundry & Machine Shops, Belleville, on boiler and bridge erection. Later for one year he was superintendent and chief engineer of the Paisley Light & Water Works Plant, Paisley, Ont., and for the next six years was with the Canadian Pacific Railway Shops.

For the last seventeen years Mr. McKenna was associated with the Babcock & Wilcox Co., Ltd., in erection work, and at the time of his death, January 9, 1920, held the position of superintendent of erection for Western Canada.

Mr. McKenna became an associate of the Society in 1918.

ARCHIBALD STEVENSON McLUNDIE

Archibald S. McLundie, first lieutenant, Ordnance Department, U. S. Army, died on September 14, 1920, at Fox Hills Hospital, Staten Island, N. Y. Lieutenant McLundie was born in Glasgow, Scotland, in 1889, and obtained both his education and early experience in that city. His first connection in this country was with the Reliance Machine & Tool Co., St. Louis, Mo., from 1906 to 1907, as a designer of brickmaking machinery. Later he became associated with the Arnold Co., Chicago, where he remained until 1908. The following year he became engaged in the design and development of steam turbines for the E. W. Bliss Co., Brooklyn. From October, 1911, to August, 1912, he was connected with the Montague Mailing Machine Co., Chattanooga, Tenn., as machine designer. At that time he entered business for himself in that city as mechanical engineer and patent attorney.

Lieutenant McLundie served in the Ordnance Department of the U. S. Army throughout the war. At the time of his death he had not received his discharge, but had been retained as mechanical draftsman for the Ordnance Department and was attached to an arsenal in New York. He had been cited to receive the Belgian War Medal for meritorious service in his particular line.

Lieutenant McLundie was a member of several Scottish societies and other fraternal organizations. He became a junior member of the A.S.M.E. in 1915.

JAMES H. MANNING

James H. Manning, superintendent of motive power, Delaware & Hudson Co., died in Albany, N. Y., on April 14, 1920. Mr. Manning was born in Cairo, Ill., on February 2, 1862, and was educated in the local schools. He entered the employ of the Union Pacific Railway in 1876 as a machinist's apprentice and was with that road until 1901, when he resigned from his position as division master mechanic to become associated with the Standard Pneumatic Tool Co., San Francisco, Cal. A few years previous to this change, Mr. Manning had invented a piston air drill which received considerable attention throughout the mechanical world. Later he was with the Featherstone Manufacturing Co., Chicago. In 1903 he returned to railroading, becoming assistant superintendent of motive power of the Canadian Pacific Railway, at Winnipeg, Manitoba, and having under his jurisdiction the territory between Winnipeg and the Pacific Coast. The following year he was appointed superintendent of motive power of the Delaware & Hudson Co.

Mr. Manning became a member of the Society in 1917. He was the first president of the Rocky Mountain Railway Club, which was organized at Denver, Colo., in 1900.

WALTER EDWARD MARSHALL

Walter E. Marshall, manager of the Buffalo office of the Warner & Swasey Co., died on August 5, 1920. Mr. Marshall was born on October 24, 1868, in Liverpool, England. He was educated in English schools. He came to this country in 1886 and entered the employ of the Warner & Swascy Co., Cleveland, Ohio, as an apprentice. Having completed his apprenticeship he was appointed foreman of the planer department. In 1892 he became superintendent and foreman for the Badons & Oliver Co., Cleveland, where he remained until 1900, when he was offered the position of manager of Dyson and Sons' Forge Shop, also in Cleveland. In 1904 he returned to the Warner & Swasey Co., and for the past seven years was manager of their Buffalo office.

Mr. Marshall was a member of the Rotary Club of Buffalo. He became a member of the Society in 1919.

EDMUND MILLS

Edmund Mills, engineer in the marine department of the Babcock & Wilcox Co., died on January 26, 1920. Mr. Mills was born on February 15, 1864, in New York City. He was educated in the schools of Jersey City. He served an apprenticeship as machinist and worked for a short time at that trade. For two years he was chief draftsman for the United Electric Traction Co., and from 1891 to 1895 held a similar position with the Wheeler Condenser & Engineering Co.

About twenty-five years ago Mr. Mills entered the marine department of the Babcock & Wilcox Co., as draftsman and continued in that department all the rest of his life. For a number of years he was chief draftsman of the department. In 1908 he was transferred to the marine department of the general office as one of its engineers. He had become an expert on the production and use of steam and was an engineer of unusually fine mechanical judgment.

Mr. Mills was a member of the Society of Naval Architects and Marine Engineers and of the American Society of Naval Engineers. He became a member of our Society in 1903.

WILLIAM FRANCHÈRE MONAGHAN

William F. Monaghan, of the engineering department of the New York Edison Co., died on April 10, 1920, of pneumonia. Mr. Monaghan was born on September 12, 1855, in Brooklyn, N. Y. He received his early education at St. John's College, Fordham, N. Y., and finished his college studies and technical training as a mechanical engineer abroad. Upon his return to the United

States in 1882 he was engaged for about five years in construction work on stationary engines. He then became connected with the Mt. Morris Electric Light Co., as chief draftsman. About 1893 he entered the employ of the United Electric Light & Power Co. as mechanical draftsman. He was subsequently employed by the New York Edison Co. in 1900 and was connected with the engineering department of this company until the time of his death.

Mr. Monaghan joined the Society in 1886 and was a life member.

LOUIS J. MONAHAN

Louis J. Monahan, president and general manager of the Universal Motor Co., died on February 3, 1920, from pneumonia following influenza. Mr. Monahan was born on August 9, 1876, in Oshkosh, Wis., and was educated in the home schools. When still very young he developed a remarkable talent for mechanics and inventions and in 1902 became associated with John D. Termaat in a number of inventions. They established the Termaat & Monahan Co. to build experimental models of gas engines. These models proved successful and the company continued under the original management until 1913, when Mr. Termaat and Mr. Monahan retired from active management. In 1914 they established the Universal Motor Company, of which concern Mr. Monahan was president at the time of his death.

Mr. Monahan became a member of the Society in 1912. He also belonged to the American Chemical Society and to the Society of Automotive Engineers.

JOHN FINDLEY MORRIS

John Findley Morris, superintendent of maintenance of the Portsmouth, Ohio, Works of the Whitaker Glessner Co., died on March 25, 1920. Mr. Morris was born in Altoona, Pa., on December 1, 1869. He was educated in the grammar and high schools of Steelton, Pa., and served his apprenticeship in the shops of the Pennsylvania Steel Co. From 1890 until 1911 he was connected with the following firms in the various capacities mentioned, thus obtaining a wide and varied experience: the Vulcan Road Machine Co., foreman machinist; Pennsylvania Steel Co., in slabbing mill, bessemer and pumping plant; Ensign, U. S. Navy during the Spanish-American War; shop foreman for Thomas Edison; assistant master mechanic, South Works of the Illinois Steel Co., Chicago; master mechanic, Dominion Iron & Steel Co., Sydney, N. S., Canada; assistant general master mechanic, Lackawanna Steel Co.; superintendent of construction of the Millikin Brothers Steel Co.; shop foreman and superintendent of M. H. Treadwell Co.; master mechanic with the Glessner Co., Porthmouth Works. From 1916 to 1917 he was connected with the Algoma Steel Co., Sault Ste. Marie, as superintendent of maintenance. He returned to the Glessner Co. at the end of that period to assume the position which he held at the time of his death.

Mr. Morris belonged to a number of clubs and fraternal organizations and was especially interested in the civic affairs of Portsmouth. He became a member of the Society in 1918.

HOMER N. MOTSINGER

Homer N. Motsinger, industrial and mechanical engineer, Chicago, died on August 13, 1920. Mr. Motsinger was born on March 16, 1875, in Pendleton, Ind. He received his early education in the schools of Shoals, Ind., and later he entered Purdue University, where for three years he studied electrical engineering. From 1900 to 1916 Mr. Motsinger was vice-president and general manager of the Motsinger Device Manufacturing Co. At the close of that period he entered professional engineering work, in which he was engaged for two years, when he became works manager of the U. S. Ball Bearing Manufacturing Co. in charge of all production and design of special machines and tools. Early in 1920 Mr. Motsinger opened consulting offices in Chicago, and was actively interested in this field of work at the time of his death.

Mr. Motsinger became a member of the Society in 1919. He was also a member of the Society of Automotive Engineers and a charter member of the Society of Industrial Engineers. He was much interested in and belonged to several fraternal organizations.

JAMES DOUGLASS MUDGE

James D. Mudge, manager of Mudge-Hoffman & Co., Seattle, Wash., died on February 12, 1920. Mr. Mudge was born in Harpersville, N. Y., on August 9, 1879. He was educated in the Binghamton schools and Cornell University, from which he received his M.E. degree in 1904. He spent a short time with the Buffalo Forge Co. and then left in January, 1906, for the West, where, for a time, he worked in the city engineer's office in Seattle, resigning to take charge of the office of the Halliday Machinery Co. in Spokane.

For the past ten years Mr. Mudge was engaged in engineering contracting work in his own interest, with offices in Seattle and Vancouver. He designed and installed numerous plants for power generation, pumping, mining, crushing and milling, as well as many special mechanical devices.

During the war Mr. Mudge rendered valuable assistance at the University of Washington by instructing in mechanical engineering. After many attempts he was accepted for active service late in 1918.

Mr. Mudge was a member of the Sons of the Revolution. He became an associate-member of the Society in 1919.

ROBERT E. MUNRO

Robert E. Munro, chief inspector of the Baltimore Department of the Hartford Steam Boiler Inspection & Insurance Company, died on March 29, 1920, after a prolonged illness. He was born June 14, 1862, and was educated in Liverpool, England, being graduated from the Liverpool Institute in 1877. After serving his apprenticeship with Rollinson's Engineering Works, Liverpool, his early career was as engineer for various steamship lines, his last engagement being with the Red Star Line, on board the *Pennland*, one of the largest ocean liners of her time. In 1888 Mr. Munro settled in this country and accepted the position of chief engineer for a large oilcloth-manufacturing establishment, at Astoria, L. I., N. Y., remaining there until September, 1891, when he became an inspector in the Baltimore office of the Hartford Steam Boiler Inspection & Insurance Company. He was promoted to the position of chief inspector in 1893 and served in this capacity until his death. Mr. Munro became a member of the Society in 1916.

MELVIN B. NEWCOMB

Melvin B. Newcomb was born in October, 1888, in Newport, N. J. He was educated in the Newport schools and received his mechanical engineering training at the Drexel Institute Evening School and the University of Wisconsin.

From 1906 to 1909 he was employed by the I. P. Morris Co., Philadelphia, as a draftsman on hydraulic turbines and centrifugal pumps. The next two years he spent in West Allis, Wis., with the Allis-Chalmers Co., as a designer and checker on turbines and pumps. In 1911 he became connected with the Wisconsin Engine Co., Corliss, Wis., where his work dealt with estimating, sales engineering, testing and experimenting. From 1914 to 1916 Mr. Newcomb was general manager of the machine business of Albert H. Newcomb & Co., Bridgeton, N. J., resigning to take a position with the Firestone Tire & Rubber Co., as designer on power-plant work. In January, 1918, he joined the hydraulic-turbine engineering department of the Wellman-Seaver-Morgan Co., and a few months later was appointed chief engineer of the rubber-machinery department of the company in Akron, Ohio, which position he was holding at the time of his death, March 13, 1920.

Mr. Newcomb was a member of the Cleveland Engineering Society and of the Akron Engineering Society. He became a junior member of our Society in 1917.

PATRICK NOBLE

Patrick Noble, who was president of the Pacific Rolling Mill Co., San Francisco, Cal., and who played such an important part in the early development of the iron and steel industry on the Pacific Coast, died on October 2, 1920. Mr. Noble was born in Abbeville, S. C., on January 14, 1849. He served during the last year of the Civil War in the Confederate Army, having just reached his sixteenth year.

In 1868 he went to San Francisco, where he started work with the Pacific Rolling Mill Co., then a very small mill in its early stages. From that time Mr. Noble was associated with the mill in various capacities, finally becoming general manager. In 1898 he took over the entire business and operated it until the time of his death.

As the mill was located at such a distance from the Pittsburgh region and transportation facilities were poor, the equipment had to be developed in great part without any previous experience or plans from which to work. Mr. Noble was responsible in a very large degree for the reputation achieved in this direction by the mill.

He was graduated from Charleston College and was a member of the Engineering Society of Western Massachusetts and of several clubs of San Francisco. He became a member of our Society in 1892.

JOHN H. NORRIS

John H. Norris, chief engineer of the National Meter Co., Brooklyn, N. Y., died on June 21, 1920. Mr. Norris was born in London, England, in 1857, and received his early education in English schools. Upon his coming to the United States he studied at Cooper Union Institute. He then served an apprenticeship with R. Hoe & Co., New York, from 1872 to 1877. For short periods directly following he worked for the firm of Beckett & McDowell, Arlington, N. J., and with the Pawtucket Manufacturing Co., Pawtucket, R. I. From 1882 to 1887 he was connected with Walter Scott & Co., Plainfield, N. J., having charge of the drawing room, including the design and equipment of a new factory. In 1887 he became associated with the National Meter Co., with which concern he was engaged until the time of his death, a period of thirty-three years. In this connection he was especially well known in the water-works field throughout the country, and was an unquestioned authority on the design and manufacture of water meters.

Mr. Norris was a particularly active member of the Society, which he joined as a junior member in 1891, receiving his full membership in 1905. He was chairman of the New York Section during 1917 and for four years served on the executive committee of the Section. He was also chairman of the excursion committee for the Annual Meeting of the Society in 1918.

Personally Mr. Norris was kindly, quiet and unassuming. Patient and thorough with the many problems of his calling, generous and genial with friends and associates, his memory will long be held in affectionate esteem by all who knew him.

JULIUS A. PERKINS

Julius A. Perkins, formerly mechanical director of the Universal Roller Bearing Co., died on December 7, 1920. Mr. Perkins was born in Hudson, Mich., in November, 1848. He served his apprenticeship as a machinist in Sublette, Ill., and was employed in various machine shops throughout this country and in Bremen and Berlin, Germany, thus obtaining an unusually wide and varied experience. For a number of years he held the position of inspector of manufacturing plants for several fire-insurance companies and his duties concerned all features dealing with fire losses. For sixteen years he designed and superintended the drafting and the manufacture of special machinery and parts in the development of Moffett and Perkins roller and ball bearings, of which he was the inventor.

Mr. Perkins was a veteran of the Civil War, having served in an Illinois regiment. He became a member of The American Society of Mechanical Engineers in 1911. He was also a member of the Society of Automotive Engineers.

JOSEPH W. PETERS

Joseph W. Peters, sales engineer, Reeves & Skinner Machinery Co., St. Louis, Mo., died on December 15, 1920. Mr. Peters was born in St. Louis on December 17, 1886. He received his early education in the grammar schools of that city and also attended the Peabody School. During his vacation period he entered Government service in the coal testing division of the Geologic Survey. He was sent later to Pittsburgh, where he became an expert computer. From Pittsburgh he was transferred to Washington and in 1907 was appointed junior chemist in the technologic branch of the Geologic Survey. While in Washington, Mr. Peters devoted all of his spare time to study and in 1912 received from Georgetown University the degrees of Bachelor of Laws and Master of Patent Law. About six months later he became the secretary of the Engineers' Club of St. Louis, where he remained until he entered the Second Officers' Training Camp at Fort Sheridan, Ill., in the summer of 1917. He received his commission as First Lieutenant Signal Aviation and was in command of the 335th Aero Squadron, serving overseas. Upon his return he again took up his duties as secretary of the Engineer's Club of St. Louis, resigning in the early part of 1919 to become connected with the Reeves & Skinner Machinery Co., where he was located at the time of his death.

Mr. Peters became an associate member of the Society in 1920.

CLEON EDGAR PHELPS

Cleon E. Phelps, plant engineer, the Allen Spool & Wood Turning Co., Mystic, Conn., died on December 16, 1920. Mr. Phelps was born in Leominster, Mass., in November, 1889, and received his early education in the schools there. Later he attended Worcester Polytechnic Institute, from which he received his B.S. in 1913. From 1913 to 1915 he was connected with the American Steel & Wire Co., Worcester, Mass., his work there being of an experimental nature. From 1915 to 1917 he was with the American Optical Co., Southbridge, Mass., engaged in development work. The following year he became assistant director of the mechanical engineering shop laboratories of the University of Illinois, and served as a machine-gun instructor during the summer months. In 1918 Mr. Phelps entered the service as a second lieutenant in the Ordnance Department of the U. S. Army. He was located as a production officer of small arms at the Marlin-Rockwell plant in New Haven, Conn. Upon leaving the service in 1919 he became factory manager of the Embossograph Products Corporation, Salem, Mass.

Mr. Phelps became a junior member of the Society in 1914. He was also a member of the American Legion and of the American Ordnance Association and of several fraternal organizations.

EDGAR RAYMOND PIPER

Edgar Raymond Piper, head of the department of drawing and design, Wentworth Institute, Boston, Mass., died on December 20, 1920, of heart trouble, after an illness of nine months. Mr. Piper was born on August 4, 1872, in Boston, and received his early education in the schools of that city. Later he was graduated from George Washington University with the degree of A.B.

Mr. Piper's first work was with his father who was a contractor and builder in Boston. In 1896 he was employed as draftsman with the Draper Co., Hopedale, Mass., and from there went to Lombard Governor Co., Boston, in the same capacity. From 1902 to 1904 he served as electrical draftsman in the U. S. Navy Yard, N. Y., and at the close of that period became connected with the Yale & Towne Manufacturing Co., Stamford, Conn., as general plant efficiency expert. A short time later he accepted the position of draftsman and designer for the U. S. Naval Gun Factory at Washington, D. C. and in 1906 became leading draftsman in the office of the Chief of Ordnance, U. S. Army, Washington. From 1909 to 1918 Mr. Piper was connected with the office of the Chief of Engineers, U. S. Army, as chief draftsman and junior engineer. Throughout the period in which he worked in Washington, Mr. Piper held the position of instructor in charge of mechanical drawing and machine design in the Carroll Institute and Y.M.C.A. Evening Schools. In 1918 he became associated with the Wentworth Institute in Boston.

Mr. Piper was a member of the Washington Society of Engineers. He became a member of our Society in 1920.

HENRY A. PORTERFIELD

Henry A. Porterfield, owner and manager of the Dexter Oil Co., Pittsburgh, Pa., and known as an expert in the testing of oils, died on February 8, 1920. Mr. Porterfield was born in Emlenton, Pa., on May 1, 1863. He attended St. Paul's School in Concord, N. H., and then entered Lehigh University, from which he was graduated in 1883. For a short period he was engaged as a mining engineer in coal regions of Wilkes Barre, Pa. He then became connected with the Cambria Iron Co., Johnstown, Pa., as a tester of steels; he was with this company for ten years and was the first to introduce into their work the testing and mixing of oils. After the Johnstown flood, which so badly crippled many of the industries of that city, Mr. Porterfield entered the employ of the Carnegie Steel Co., Pittsburgh, Pa., where he organized laboratories in the various works of the company for the testing of oils. For many years prior to his death he managed the Dexter Oil Works.

Mr. Porterfield became a member of the Society in 1890. He belonged also to several university clubs in Pittsburgh.

WHITFIELD P. PRESSINGER

Whitfield P. Pressinger, vice-president of the Chicago Pneumatic Tool Co. and actively engaged in the pneumatic-tool and allied machinery industry for many years, died on June 10, 1920. Mr. Pressinger was born in New York City in 1871. He was general manager of the Clayton Air Compressor Co. for seven years and became widely known through his numerous activities in connection with his work in that line. He became an associate of the Society in 1903. He was also a member of the Compressed Air Society, the Sons of the Revolution, the Masonic fraternity, and belonged to a number of New York clubs.

WILLIAM T. PRICE

William T. Price, chief engineer of the oil-engine department of the Ingersoll-Rand Co., died on November 7, 1920. Mr. Price was born on September 28, 1883, in Montreal, Quebec, Canada. He was graduated from Cornell University in 1906 with the degree of M.E. His first employment was with the Wheeling Mold & Foundry Co. as assistant engineer, where he was located for about a year when he became works manager of the Contractors' Plant Manufacturing Co. In 1908 he became connected with the De La Vergne Machine Co., as sales engineer, and during the course of the next six years held the positions of sales manager, chief engineer of the power department and manager and chief engineer. At the time of his death he was actively associated with the Ingersoll-Rand Co. as chief engineer of the oil-engine department. He was also vice-president of the Rathbun-Jones Engineering Co., Toledo, Ohio.

Mr. Price was a member of the Society of Automotive Engineers and the Engineers' Club of New York. He became a junior member of the Society in 1910 and was promoted to full membership in 1915.

NELSON STEPHEN PRINGLE

Nelson S. Pringle, assistant engineer in charge of production with the Autocar Co., Ardmore, Pa., died on February 2, 1920. Mr. Pringle was born in Newark, N. J., on November 19, 1885, and attended the schools of that city. He left school to enter commercial life, and was for six years connected with the New York Export & Import Co.

He then became a student at Stevens Institute of Technology, from which he was graduated in 1911 with the degree of M.E. In the same year he entered the employ of the Autocar Co. as experimental engineer and remained with the company until the time of his death in various capacities.

Mr. Pringle became an associate member of the Society in 1914. He was also a member of the Society of Automotive Engineers, the Engineers' Club and the Philadelphia Rifle Club.

ALFRED RAYMOND

Alfred Raymond was born in New Orleans on May 13, 1865. He received his early education in that city and then entered Tulane University, receiving his B.S. degree in 1895 and his M.E. in 1897. Upon graduation he engaged for a short period in work for the U. S. Geodetic Survey. He then took a position with the American Sugar Refining Co., New Orleans. In 1889 he was appointed electrician in charge of the meter department of the Edison Electrical Illuminating Co. and the following year was made superintendent.

Mr. Raymond became prominently and favorably known for his ability in electrical public-service work, shown by the following record of his activities: superintendent, Edison Electrical Illuminating Co., 1890 to 1893; superintendent of construction, Southern Electrical Manufacturing Co., 1893 to 1895; superintendent, Louisiana Electric Light Co., 1895 to 1896; city electrician, New Orleans, 1896 to 1898.

As city electrician Mr. Raymond prepared plans for the city fire-alarm system, and for the initial underground electric-transmission system, collaborating with the then Drainage Commission of New Orleans, in the plans and specifications for the electric-power and transmission work in connection with the complex and unique system of drainage without which New Orleans could not function as a modern city.

In 1898 Mr. Raymond was appointed general manager of the operating department of the New Orleans Drainage Commission, and when in 1904 this Commission was merged into the Sewerage and Water Board of New Orleans, he retained the title and functions of general manager.

Mr. Raymond became a member of our Society in 1904. He belonged also to the American Society of Civil Engineers, the American Institute of Electrical Engineers and was an active member and past president of the Louisiana Engineering Society. His activities were many and unselfish. He was an administrator of the Tulane Fund, associate architect on the Ursuline Convent, member of the New Orleans Garbage Commission and during the War chairman of one of the local draft boards.

Mr. Raymond died on January 13, 1920. By his death the engineering profession loses one of its most prominent and versatile members.

ROBERT CLARKSON REID

Robert C. Reid, secretary of the Harrolds Motor Car Co., New York, died in that city on January 11, 1920, from injuries received in an automobile accident. Mr. Reid was born on July 8, 1875, in Kingston, N. Y. He was educated in the schools of Nyack, N. Y., and Englewood, N. J. He also attended Pratt Institute, Brooklyn, and Cooper Union, New York, to obtain his technical training.

He served his apprenticeship with R. Hoe & Co., manufacturers of printing presses, and in 1897 became connected with the American Sugar Refining Co. as draftsman. He left this concern to become salesman for the Chapman Valve Co. Later he held a similar position with the Hewitt Motor Co. In 1907 he became associated with the Harrolds Motor Car Co. as manager of their truck department, and four years later became secretary of the company.

Mr. Reid was a member of the Society of Automotive Engineers, and of the Motor Truck Club of America. He became a member of our Society in 1907. During the Spanish-American War he served as corporal in the Fifth New Jersey Volunteer Infantry.

ALVA C. RICE

Alva C. Rice, who retired from active consulting work several years ago, died on July 8, 1920, in Worcester, Mass. Mr. Rice was born on July 23, 1845, in Nicholville, N. Y. He received a common-school education and began work in a flour mill. He became interested in millwright and contracting work and for a time was employed in this business in northern New York. In 1874 he became associated with the Holyoke Machine Co., Holyoke, Mass., having charge of the wood-shop and millwright work. Later he built the Worcester shops for this company and upon their completion became superintendent. In 1885 he removed to Dayton, Ohio, as superintendent and engineer for the Stillwell & Bierce Manufacturing Co., in which position he remained until 1899 when he returned to Worcester and opened consulting offices. His work was mostly on hydraulic power plants and pulp and paper-mill installations and until his retirement he was considered one of the leading engineers in those lines.

He was the designer of plants for the Chambly Manufacturing Co., the Hudson River Power Co., Montmorency Electric Power Co., Belgo-Canadian Pulp Co., Paul Smith Hotel Co., Hartford Electric Light Co., Greenfield Electric Light Co., and many others. He was also the designer of the Lachine Rapids turbine installation on the St. Lawrence River, and served as consulting engineer on several of the Niagara Falls plants and the plant of the St. Lawrence River Power Co. at Massena, N. Y.

Mr. Rice was a member of the masonic order. He became a member of the Society in 1890.

LEVI S. RICHARDSON

Levi S. Richardson was born in Cincinnati, Ohio, on September 30, 1872. Upon the completion of his high-school course he entered the University of Michigan, where he spent two years. From that time until 1919 Mr. Richardson was connected with the following firms for periods of from one to four years, gaining a wide and varied experience: the Sprague Electric Co., Watsessing, N. J.; R. Hoe & Co., manufacturers of printing presses, etc.; Ford, Bacon & Davis, Engineers; Colgate & Co., the New York Edison Co., all in New York City; the Transit Development Co., Brooklyn; the Lenher Engineering Co., New York; S. Firestone, Rochester, W. S. Barstow & Co., New York, and Westcott & Mapes, New Haven, Conn. In May, 1919, Mr. Richardson returned to the employ of Barstow & Co., to work on the erection of a leaching plant in Ironville, Ohio, and a little later on a plant in Hastland, West Virginia. After an illness of six months, following his return form Hastland, he died on April 18, 1920.

Mr. Richardson became a member of the Society in 1909.

HAROLD PRESTON RIX

Harold P. Rix, vice-president and branch manager of the Los Angeles office of the Rix Compressed Air & Drill Co., died on February 11, 1920. Mr. Rix was born in San Francisco, Cal., on August 2, 1888, and was educated in the schools of Redwood City and Palo Alto, Cal. He served a four-year apprenticeship in the Asiatic squadron of the U. S. Navy and at the age of twenty was honorably discharged with the rating of coxswain. For about two years he was connected with the North Star Mines, Grass Valley, Cal., where he worked first as a machine-drill miner and later in the compressor room of the cyanide plant. He then became assistant to the master mechanic of the Nevada Petroleum Co., Coalinga, Cal.

In 1912 he became associated with the Rix Compressed Air & Drill Co., San Francisco, Cal., as salesman; later he was appointed branch manager of the Los Angeles office, where for several years he had charge of the installation of many compressed-air plants for industrial, mining and pumping purposes.

Mr. Rix belonged to several social clubs in Los Angeles. He became an associate of the Society in 1919.

WILLIAM E. ROBERTS

William E. Roberts was born in Horsforth, England, in February, 1859. He was educated in private schools and in Ilkley and Yorkshire Colleges, England.

Mr. Roberts served his apprenticeship with Joseph Whitham & Sons, Leeds, spending two years in the drafting room and four in the various shops of the firm. From 1881 to 1918 he was engaged for periods of from one to four years with the following concerns, thus gaining an unusually valuable experience: Greenwood & Batley, Leeds, draftsman; John Butler & Sons, Stanningley Iron Works, mechanical engineer; Leeds Forge Co., Leeds, chief draftsman; Fox Solid Pressed Steel Co., Joliet, Ill., superintendent; Anaconda Mining Co., Butte, Mont., chief draftsman; W. A. Clark, Smelter, Houston, Idaho, master mechanic; Parrot Mining Co., Butte, Mont., mechanical engineer; F. A. Heinze Smelter, Butte, Mont., construction engineer; Illinois Steel Co., Chicago, checker; Indiana Steel Co., Gary, Ind., checker and designer; Fairbanks, Morse & Co., Chicago, checker and designer; Koppers Coke Oven Builders, Chicago, checker and designer; Willputte Coke Oven Corporation, New York, checker and designer. In 1918 Mr. Roberts became connected with the Foundation Oven Corporation, New York, as superintendent of maintenance and construction in their Chicago office, which position he held at the time of his death, February 23, 1920. He was a recognized expert in the coke-oven industry.

Mr. Roberts became a member of the Society in 1918.

S. SAKATA

S. Sakata, professor of mechanical engineering, Tokio Technical College, Asakusa, Tokio, Japan, died on December 1, 1920. Professor Sakata was born in Tokio, Japan, on August 27, 1857. He was educated in the Imperial University of Tokio, where the full mechanical engineering course covered a period of five years. He was graduated in 1880. In July of that year he became chief engineer of the Government printing office, remaining there for seven years, when he was appointed to a professorship in the Tokio Technical College.

In 1887 and again in 1890 Professor Sakata acted as examiner for the National Industrial Exhibition of Japan. He became a member of the Society in 1891.

JAMES WARING SEE

James Waring See, a charter member of the A.S.M.E. and a widely known engineer, died at his home in Hamilton, Ohio, on January 31, 1920. Mr. See was better known to the readers of mechanical journals as "Chordal," a pen name used by him for much of his writing. About 1880 he began a series of articles in the *American Machinist* under the title of "Extracts from Chordal's Letters." These articles were of the shop by a shop man and it would be difficult for a person of the present generation to understand the deep impression which they made on the mechanical men of the country and the widespread interest which they created.

Mr. See was born in New York City on May 19, 1850. He received his earliest education in a country school at Rutland, N. Y., though he later attended school at St. Louis, Arcadia and Springfield, all in Missouri.

At the outbreak of the Civil War Mr. See was employed in the Springfield military hospital as an assistant in the dispensary and in the operating ward. After the battle of Carthage he was made telegraph messenger to the Federal forces in and around Springfield. Upon the ending of hostilities Mr. See served an apprenticeship as machinist at the Springfield Iron Works, and it was there he laid the foundation for his future career. At the completion of his apprenticeship he found employment in various shops located from St. Louis to Yankton, S. D., eventually settling at Omaha, Neb., where he started a shop of his own.

The story is told of him that, being disappointed in a lathe he bought from the Niles Tools Works, he wrote to that firm to the effect that if he could not design a better lathe he would eat it. Alexander Gordon, head of the Niles works, testily replied that if he was able to design a better lathe than the one in question, he wished he would come to Hamilton and do it.

Long after Mr. Gordon had forgotten the incident there appeared in his office a tall, lanky, individual who said with a drawl, "My name is See and I came up here to design that lathe for you." He was put off with one excuse after another until in disgust he appropriated an empty drawing board and did design a lathe that made Mr. Gordon sit up and take notice.

He was at once employed by the Niles Works, filling positions as foreman, chief draftsman and chief engineer, respectively. In 1876 Mr. See opened an office in Hamilton, Ohio, as a consulting mechanical engineer and his keen insight into mechanical matters soon brought him a wide practice in connection with some of the largest machine establishments in the United States and Europe.

With the invention of the telephone, Mr. See became greatly interested in its practical workings and was the inventor of several valuable devices connected with central-station apparatus. In recognition of his work he was made an honorary member of the Telephone Exchange Association. For a time he was editor of the *Telephone Exchange Reporter*. He built the first telephone line in Hamilton. Also, in connection with Alexander Gordon and James K. Cullen, he built an electric-light plant at the Niles Tool Works, the first in Hamilton.

As Mr. See's business increased, he became a patent attorney and as such developed an enviable reputation as an expert in patent litigation, and was called on to give expert testominy in more than three hundred cases.

By appointment of Governor Campbell, Mr. See acted as one of the commissioners for Ohio to the World's Fair held at Chicago in 1893.

ALBERT ROBOSSON SHIPLEY

Albert R. Shipley, production superintendent of Old Line Products, died on February 15, 1920. Mr. Shipley was born on April 20, 1876, in Alpha, Md. He attended school in Baltimore and was graduated from both the McDonough and the Maryland Institutes of that city. He was a machinist with the Tabor Manufacturing Co., Philadelphia, at the time that Frederick W. Taylor was successfully working out in practice there the principles of his system of scientific management. His association with Mr. Taylor continued over seven years.

After serving the New England Butt Co., the Seth Thomas Clock Co. and the Sentinel Manufacturing Co. in industrial engineering and executive capacities, he was called by the Government in August, 1916, to become superintendent of production at the Watertown Arsenal. In January, 1918, Mr. Shipley joined the industrial engineering department of the Winchester Repeating Arms Co. and remained with that department until his appointment in October to Old Line Products department of the company.

Mr. Shipley became an associate of the Society in 1912. He was also a member of the Taylor Society and belonged to several fraternal organizations. He frequently lectured on scientific management, in which he was regarded as an authority in view of his work under the immediate direction of Mr. Taylor.

FRED A. SHORT

Fred A. Short, mechanical engineer, Stone & Webster Corporation, Boston, Mass., died suddenly on June 28, 1920, of pneumonia. Mr. Short was born on May 30, 1883, in South Middleboro, Mass., where he received his early education, which was later supplemented by travel and special studies.

Mr. Short started work in 1901 with George Gray, consulting engineer, Providence, R. I., as a draftsman. From 1902 to 1907 he was with Fred S. Hinds, Boston, where he held the position of assistant equipment engineer, when he resigned to travel for two years in Europe. There he devoted himself especially to the study of building construction and architectural design, together with cost production and labor. Upon his return to this country Mr. Short became connected with the B. F. Sturtevant Co., Hyde Park, Mass., in the mechanical designing of large apparatus in connection with heating and ventilating for all classes of building. In 1913 he became associated with the Stone & Webster Corporation, Boston, as chief checker. He was advanced through the positions of squad boss and estimator to equipment engineer on power and industrial plants. From 1917 to 1919 he was "loaned" to the engineering division of the Ordnance Department of the Army. In January, 1919, he returned to Stone & Webster an mechanical engineer.

Mr. Short became an associate member of the Society in 1918.

EDMUND GYBBON SPILSBURY

On May 20, 1920, Edmund G. Spilsbury, mining and metallurgical engineer of international reputation, died suddenly after many years of useful service to the profession and to the engineering societies.

Mr. Spilsbury was born in London, England, in 1845. He was educated in Belgium and was graduated from the University of Louvain in 1862. He then took up practical work in Germany, entering in 1864 the service of the Eschweiler Zinc Co., Stolberg, one of the largest miners and smelters of lead and zinc in the world. The following year he was given charge of that company's mines and works in Sardinia. In 1867 he became connected with McClean and Stillman, of London, in charge of the construction of the lock gates for the Surrey Commercial Docks; in 1868 he accepted the position of designing engineer with J. Casper Harkort and was assigned to the detail work of the Keulenberg Bridge in Holland, the Danube Bridge at Vienna and the Rhine Bridge at Düsseldorf. Later he returned to the Eschweiler Company as chief engineer, and in 1870 came to the United States for the Austro-Belgian Metallurgical Co. to investigate the resources of this country in lead and zinc.

After two years in this country Mr. Spilsbury decided to take up practice here, in the course of which he introduced the Harz system of ore dressing for the zinc ores of Pennsylvania and New Jersey. During this period he was also engaged in explorations on the northern shore of Lake Superior and in Colorado, Montana, Utah and California. From 1873 to 1875 he was engaged as general manager of the Bamford (Pa.) Smelting Works; in 1879 he designed and built the Lynchburg Blast Furnace and Iron Works; he was also consulting engineer for the Coleraine Coal & Iron Co., Philadelphia. In 1883 he became general manager of the Hailer Gold Mine in South Carolina and in 1887 became connected with Cooper, Hewitt & Co., New York, From 1888 to 1897 he was managing director of the Trenton Iron Co., Trenton, N. J. While manager of these works he introduced as specialties of their business the Eliot locked wire rope and the Bleichert system of aerial tramways.

His practice as a consulting mining engineer and metallurgist took him into many parts of Europe, Africa, the United States, Mexico and South America.

During the winter and early spring of 1920 he spent a number of weeks in Brasil on a mining project for clients in the United States and had returned to the United States only a few weeks prior to his death.

Mr. Spilsbury was the author of a number of important technical papers, mostly on mining processes and on mine appliances. He also wrote entertainingly in a lighter vein, and the following paragraphs from "Technical Reminiscences," published in *Mining and Scientific Press* in 1915, recall early interesting experiences.

> Early pioneer work in Sardinia was most interesting and instructive. In the first place, the island was almost as much a terra incognita as our own West a century ago. And yet the civilization of Sardinia was one of the oldest in the world. The Phœnicians, the Egyptians, and later the Carthaginians and the Romans, had all left their marks, in old excavations, dumps, slag piles and more especially in the tombs cut out of solid rock. In the early 60's the northern and eastern parts of the island were dangerous, as well as difficult of access. The tribes occupying this mountainous district did not recognize the Italian Government and were generally classed as bandits. During the second year of my stay one evening I heard a disturbance outside my camp and the clatter of a number of horses. Thinking that I was about to be raided by the bandits, I called the two foremen who were in the house and, getting our guns, we started to see what the trouble was. Opening the door, I saw a small calvacade. The leader was a very tall and slim young gentleman, dressed in full hunting costume. He addressed me by name, stating that he had a letter of introductuon and asking if he and a friend could stay the night. I thought I had never seen a taller or lankier fellow, but withal, a most pleasant face. He handed me a letter from one of the directors of our company, asking me to make the stay of the bearer as pleasant as I could, as he was a friend of theirs, being the Duke of Brebant, afterward Leopold the Second, King of the Belgians. The Duke had come to Sardinia on a mountain sheep hunt. Years afterward, when he had succeeded his father, I always received a hearty welcome whenever I met him either in Brussels or Paris.

And again we find these paragraphs telling of the changes which have taken place in mining practice:

> I wonder how many mining engineers of the present day can call to mind changes and advances in the practice of mining which have taken place in the last half century? Practically every appliance beyond the pick and shovel, hand hammer and drill, has been introduced during that short period. When I first started my practical course in underground work, we knew none of the aids now considered requisite to work economically. We had no machine drills to save the physical labor of hand work. Dynamite or its precursor, nitroglycerine, had not yet been invented; fuse was still unknown; even steel was more or less of a luxury, and our hand drills were still made of iron with steel bits welded on to them. The mushrooming of the head of the drill under the hammer blows was a constant source of trouble and injury to the hands of the miners. The only explosive known was the large-grained black powder.
>
> It seems hardly credible today, and yet it is a fact that the contract price for driving or stoping ground was influenced by the condition of the rye crop. In seasons of drought the height of the rye was much curtailed and the length of the straw between joints would be so short that the time occupied in jointing and filling the stalks which were used as fuses, greatly lessened the driving capacity of the miners. The straw primers naturally limited the possible depth of the hole and three feet was considered a good average. These straw fuses were very rapid and it was necessary to ignite them by a slow-burning sulphur wick.

Mr. Spilsbury was intensely interested in the work of the national engineering societies. He was a member of the American Institute of Mining and Metallurgical Engineers, of which he was president in 1896, the American Society of Civil Engineers, the Institution of Mining and Metallurgy of Great Britain, the Mining and Metallurgical Society of America, and the American Electrochemical Society; he became a member of The American Society of Mechanical Engineers in 1890. In 1916-1917 he was president of the Engineers' Club of New York. He was a trustee of the United Engineering Society from 1916 until his death; a member of the Engineering Societies Library Board from its organization in

1913 and a member of the Engineering Foundation Board from 1916. He was also a member of the National Research Council. In all these associations he was an active and useful member and served on numerous committees and contributed freely of his time and ability.

HAL R. STAFFORD

Hal R. Stafford, chief engineer of the Franklin Railway Supply Co., New York City, died on December 9, 1919. Mr. Stafford was born in Saginaw, Mich., on November 27, 1877. He was educated in the Saginaw schools. In 1900 he went to Schenectady, New York, where he was employed in the erecting shops of the Schenectady Locomotive Works.

The following year he became connected with the American Locomotive Co. as draftsman; from 1902 to 1909 he was in charge of the cylinder, valve and compound locomotive design for the company and at the close of that period became assistant to the consulting engineer of the firm. In 1913 Mr. Stafford became chief engineer of the Economy Devices Corporation, New York, and served in that capacity until the company was combined with the Franklin Railway Supply Co., under the name of the latter, of which he was made chief engineer.

Mr. Stafford was a member of the Central Railway Club, the New York Railway Club and the Machinery Club of New York. He became a member of the Society in 1917.

PERCY BELL TAYLOR

Percy Bell Taylor, consulting engineer in mechanical and electrical lines, Newark, N. J., died at his home in that city on October 21, 1920. Mr. Taylor was born in Manchester, England, on May 26, 1865, and when eight years of age came to the United States. In 1881 he entered the employ of Cyrus Currier & Sons, machinists, at Newark, N. J., and remained with that organization for sixteen years, during which time he occupied various positions, finally becoming mechanical engineer for the company.

He later organized the Taylor-Martin Co., designers and manufacturers of special machinery and tools. In 1897, Mr. Taylor dissolved this organization to engage independently as a consulting engineer. Many notable power-plant, machinery and general engineering installations were made under his design and supervision. Of recent years, Mr. Taylor undertook complete plant construction, including power and electrical installations as well as machinery. For a number of years he was employed as consulting expert by the Board of Education for the installation of heating and ventilating equipment and electrical apparatus in the local public schools.

Mr. Taylor was a registered architect in New Jersey and a member of the New Jersey Society of Architects and of the Board of Trade. He also belonged to several clubs and to the masonic order. He became a member of the Society in 1909.

EDWARD W. THOMAS

Edward W. Thomas, former president of the National Association of Cotton Manufacturers and for the last eleven years agent of the Boott Mills, Lowell, Mass., died of heart disease at his residence in that city on November 15, 1920.

Mr. Thomas was generally regarded as one of the ablest cotton manufacturers in the country, having held responsible positions with mills both in the North and in the South. His eminent fairness in all matters of mill routine which came to him for discussion had won the confidence of the employees, and this quality, combined with an unfailing courtesy even to the most humble operative, was a potent factor in the successful development of the large plant which he directed for the past eleven years.

He was born on February 22, 1858, in Lowell, Mass. He spent all of his business life in cotton manufacturing, starting as a draftsman with the old Lowell Machine Shop and leaving that concern to enter the employ of the American Thread Company at their plant in Willimantic, Conn. In 1882 he became superintendent of the Tremont & Suffolk Mills, Lowell, and four years later became agent for the corporation.

During this period he was particularly active in the affairs of the National Association of Cotton Manufacturers, serving as vice-president from 1892 to 1894 and as president in 1894 and 1895.

About 1900 Mr. Thomas went south to become general manager of the Cooleemee (N. C.) Cotton Mills, and in 1904 he assumed the more responsible position of general manager of the Olympia and Granby Cotton Mills, Columbia, S. C. Three years later he became general manager of all the plants of the Consolidated Cotton Duck Company, Baltimore, Md. In 1909 Mr. Thomas returned to Lowell as agent for the Boott Mills, which position he held at the time of his death.

Mr. Thomas was a member of both the National and American Associations of Cotton Manufacturers, the Boston Textile Club, the Yorick Club of Lowell and the Lowell Historical Society, of which latter he was vice-president. He had long been prominent in masonry and was a Knight Templar and a thirty-second-degree mason.

He became a member of The American Society of Mechanical Engineers in 1880, attending the organization meeting at Stevens Institute of Technology on April 7, 1880. He served as secretary of the sub-committee on Textiles from 1912 to 1914.

LYMAN HANBRIGHT TREADWAY

Lyman H. Treadway, president of the Peck, Stow & Wilcox Co., Cleveland, Ohio, and Southington, Conn., died on December 7, 1919. Mr. Treadway was born on March 27, 1862, in New Haven, Conn., and was educated in the schools of that city. At the age of eighteen he entered the employ of the Peck, Stow & Wilcox Co. as clerk. He was advanced through various positions in mercantile industrial and financial departments, finally being elected to the presidency of the concern in 1911.

Mr. Treadway became an associate of the Society in 1917. He was president of the Cleveland Chamber of Commerce from 1907 to 1908.

CLARENCE VOSE

Clarence Vose, who became a member of the Society in 1891, died on August 21, 1920. Mr. Vose was born in Providence, R. I., on January 8, 1845. He received his education in the public schools of that city and later attended business college. Mr. Vose was connected for a time with the United States Engineer's Office at Newport, under General G. K. Warren. From there he entered the service of the Standard Oil Co., with whom he remained for about thirty years until his retirement nearly twenty years ago. During his active career he also held the position of superintendent of the Queens County Oil Works and of works manager of the Pratt Manufacturing Co. Later he was with the Standard Oil Co. of New York.

RUDOLPH VUILLEUMIER

Rudolph Vuilleumier, chief engineer, Pintsch Compressing Co., New York, died on May 5, 1920. Mr. Vuilleumier was born in Basle, Switzerland on April 19, 1869. He received his early education in Basle and later entered Stevens Institute of Technology from which he was graduated in 1902 with the degree of M.E.

Upon graduation he became connected with the Pintsch Compressing Co. as assistant engineer on designing, development work and construction. Later as engineer of operation he had charge of the various gas works of the company in the United States and Canada. In 1913 Mr. Vuilleumier was appointed chief engineer, his duties having to do with the development, new design, construction and operation of the gas and oxygen plants.

About fifty patents were issued to Mr. Vuilleumier, among the principal ones being a new process of obtaining oxygen and one on heat induction, the work on which was not fully completed.

Mr. Vuilleumier became a member of the Society in 1918.

ARTHUR M. WAITT

Arthur M. Waitt, well-known consulting engineer of New York City, died on November 10, 1920. Mr. Waitt was born in October, 1858, in Boston, Mass. He was educated in the Boston schools and attended the Massachusetts Institute of Technology, from which he received his S.B. degree in 1879.

Upon graduation he became connected with the C. B. & Q. Railroad, leaving that road several years later to become general foreman of the car shops for the Eastern Railroad Company. From 1885 to 1888 he was general foreman of the car department for the same company. In 1888 he became connected with the Boston & Maine Railroad as assistant master car builder and the following year took a position in the same capacity with the L. S. & M. S. Railroad. Three years later he became general master car builder for that company and was with them until 1899 when he became superintendent of motive power and rolling stock of the New York Central Railroad. He held this position until 1903 when he resigned to open consulting offices in New York City, in which field of work he was engaged until the time of his death.

During the War, however, Mr. Waitt, who held the commission of Major, gave up his practice temporarily to organize one of the largest purchasing branches of the Ordnance Department in New York and Hartford. His death was due to overwork in this undertaking into which he entered so wholeheartedly.

During his connection with the New York Central he was made a member of the Electric Traction Commission, which planned the electrification of the road from New York to Harmon. He also assisted in designing one of the best locomotives in use at that time, among them the engines of the series 1400 and 2900. He was associated after his retirement from the New York Central with Colonel William J. Wilgus in the appraising of the rolling stock of the Lehigh Valley Railroad. He was an enthusiastic supporter of the automatic train control and a director of the Sprague Safety Control & Signal Corporation.

In 1893 Mr. Waitt was elected president of the Central Railroad Club of Buffalo and in 1895 president of the Western Railroad Club of Chicago. As one of the officers of the Lake Shore & Michigan Southern Railroad, he was a participant in the famous high-speed run of October 24, 1895, from Chicago to Buffalo; a run which, it is believed, has never been equaled. The test started at 3:29 A.M. in Chicago and ended at 11:30 A.M. in Buffalo, a total distance of 510.1 miles, done in eight hours, one minute and seven seconds.

In 1901 Mr. Waitt was elected president of the American Railway Master Mechanics' Association. In 1916 he became a member of the Connecticut Assembly and it was he who wrote the automobile laws now in force in that state.

Mr. Waitt was also a member of the Connecticut Chamber of Commerce, the New York Railway Club, the St. Louis Railway Club and the Engineer's Club of New York. He was a thirty-second-degree mason.

He became a member of The American Society of Mechanical Engineers in 1893 and held the office of vice-president from 1900 to 1902.

EDWARD T. WALSH

Major Edward T. Walsh was born in April, 1872, in South Orange, N. J. He obtained his technical education in Pratt Institution, Brooklyn, and was for some years connected with the maintenance department of the New York and Brooklyn Bridge. Later he became connected with the Atlas Portland Cement Co. as a designer of power-plant and building equipment in their mechanical department. From 1901 to 1906 he was employed by the Interborough Rapid Transit Construction Co. as draftsman, resigning to go to Dayton, Ohio, as construction engineer with the National Cash Register Co. where he superintended the construction of several large factory buildings.

From Dayton Major Walsh went to Philadelphia, where he continued his work in the design and construction of industrial buildings, becoming associated in 1909 with Frederick A. Waldron in industrial engineering; one of his notable undertakings at this time was the construction of the large service station for the Edison Illuminating Co., Boston, Mass.

Shortly after the opening of the world war Major Walsh became chief engineer of the Canadian Car & Foundry Co., New York, which handled a number of large shell contracts for the Russian Government. He also spent nearly one year with the Maxim Munitions Corporation on a cartridge contract at Watertown, N. Y. When the United States entered the war he was commissioned as major in the Ordnance Department, serving in some of the most active divisions of the work. Following the armistice, he was connected with the salvage board of the Western New England District, and at the time of his death, January 13, 1920, was chairman of this board.

Major Walsh became a member of the Society in 1913.

ARTHUR C. WALWORTH

ARTHUR C. WALWORTH, a member of The Society since 1886 and Manager from 1894 to 1897, died on June 23, 1920. He was born in Boston, Mass., on April 29, 1844, a son of James J. Walworth, the founder of the Walworth Manufacturing Company, and a pioneer of the steam-heating business in this country. He attended the Boston Latin School and later entered Yale University, being graduated in the class of 1866. He received his technical training at the Lawrence Scientific School, Harvard, and the Écoles des Pontes et Chaussées, Paris. He entered upon the profession of mechanical engineeering, and became one of the leading authorities on heating and ventilating during the years of greatest expansion of this business. He was once the President of the National Association of Steam and Hot Water Engineers.

Although a tireless worker in his chosen profession, his activities and interests reached far beyond into the field of public and social service. Throughout his mature life a resident of the city of Newton, Mass., he was closely identified with the progress and welfare of the city. Perhaps his greatest service was his part in the securing of a large and beautiful tract of land for a public playground in Newton Centre, and this at a time when the value of playgrounds was little appreciated. He represented his city in the legislature in 1887–8.

He was president of the Boston Yale Club and of the National Association of Steam and Hot Water Engineers. He was president of the Malleable Iron Fittings Co., and also of the Walworth Construction and Supply Co.; he was a trustee of Atlanta University, an Incorporator of the University Club and a member of Boston Athenaeum, Engineers' Club, Massachusetts Club of Mechanical Engineers. He served on the first standardization committee for the standardization of flanged fittings and he took special interest in the engineering for the heating and ventilating of the buildings at Yale University.

WALTER J. WARDER

Walter J. Warder was born in Chicago, Ill., on August 14, 1879. He was educated in the Chicago public schools and later entered Lewis Institute from which he was graduated as an electrical engineer.

For four years upon graduation he worked for the Western Electric Co., Chicago, where he gained experience in the testing and design of motors and generators. He was then employed by Roth Brothers & Co., Chicago, and during his five years with the concern he designed a complete line of direct-current motors and generators, which machines are still being manufactured by the company.

To broaden his experience he resigned from this position and was for a time with the Westinghouse Electric & Manufacturing Co., Pittsburgh, Pa., the Adams & Westlake Co., Chicago, and for three years was with the Crocker-Wheeler Co., Ampere, N. J. Leaving the electrical line, he engaged with the Tampa Steam Ways Co., Tampa, Fla., in ship building. He was also interested in the automobile business there. In 1917 Mr. Warder returned to Roth Brother & Co., as chief engineer in electrical and mechanical design, which position he held at the time of his death, January 20, 1920.

Mr. Warder was particularly able in the design of direct-current motors and generators and had great influence in the design of motors for elevator work. During the past three years he was very much interested in the development of a line of plating generators, and excellent designs have been developed under his supervision.

Mr. Warder was financially interested in Roth Brothers & Co. and for ten years was also vice-president of the Pitt Engineering Co., manufacturers of elevator equipment.

He was a Fellow of the American Institute of Electrical Engineers and a member of the Western Society of Engineers. He belonged to several clubs in Chicago and was a member of various fraternal organizations. He became a member of our Society in 1911.

MURRAY WARNER

Murray Warner, Major in the Quartermaster Corps of the United States Army, died suddenly on October 2, 1920, in San Francisco, Cal. A military funeral was held on the afternoon of October 4 from the home of Mr. Walter Bliss (a Technology classmate of Major Warner's) with an escort of one battalion of infantry, pallbearers of non-commissioned officers and a military band. Services were held in the Presidio Chapel.

Major Warner was born on March 8, 1869, in Clinton, Ill. He was prepared for college in Phillips Academy, Exeter, and then entered the Massachusetts Institute of Technology, from which he received the degree of B.S. in 1892.

His first position was as draftsman with the American Wheelock Engine Co., where he remained two years, when he became connected with the California Engineering Co. From 1896 to 1900 he was with the International Power Co., his work dealing with the design and erection of power plants. In 1900 Major Warner was appointed by the American Trading Co. to establish an engineering department for them in China. Three years later the firm of Murray Warner & Co., consulting engineers and contractors, was established in Shanghai. In 1910 Major Warner retired from active engineering work not to enter it again until July, 1917, when, as a "dollar-a-year-man," he was made construction quartermaster at Camp Dix, Wrightstown, N. J. In November, 1917, he was commissioned a Major in the Q. M. C. and put in charge of the utilities at Camp Dix. where he served until the signing of the armistice. Major Warner refused a colonelcy in Washington, believing that he could render a greater service at Camp Dix. After the armistice he was transferred to the Mexican border, where he rendered unusually efficient service. At the time of his death he was stationed at San Francisco on General Liggett's staff, in charge of the utilities of the Western Department.

During the Spanish-American War, Major Warner had served as a lieutenant in the Navy. At the time of the Boxer Rebellion he was in charge of the American Company of Volunteers, organized to protect foreign settlement in Shanghai, China. He was twice president of the American Association of China, which represented and controlled all American interests in that country. He was a thirty-second degree Scottish Rite mason, belonging to the Shanghai Chapter. He became a member of our Society in 1918.

JESSE WARRINGTON

Jesse Warrington, who became a member of The American Society of Mechanical Engineers in 1887, died at his home in Indianapolis, Ind., on November 25, 1920, of heart failure. He was born in Damascus, Ohio, on January 20, 1846, and was educated in the Damascus Academy and the Salem, Ohio, high school.

In his early years Mr. Warrington was employed in various works in Salem, first as wood turner and later as machinist, draftsman and designer. He left Salem in 1876 and after two years in Jackson, Mich., with E. Dennis & Co., became connected with the Nordyke & Marmon Co., Indianapolis, where he held the position of designer and mechanical superintendent until 1909, when he was physically incapacitated for business by a slight stroke of apoplexy. While with this company he made important improvements in the machinery and methods for flour production, and a recognition, which he greatly appreciated, was the unexpected receipt of a Diploma of Honorable Mention from the Columbian Exposition for his work in connection with the perfection and production of the machinery upon which his company had received an award at the Exposition.

Mr. Warrington was richly endowed with intellectual gifts, among them being an apparently intuitive knowledge of the principles of mathematics, mechanics and physics. He was the final resource for mathematicians in the several communities in which he resided when they were balked by seemingly insolvable problems.

While at Salem, Ohio, in the middle seventies, he was connected with the Buckeye Engine Co., and his was the mind behind the unusual development of that concern in the theory of steam engineering and the literature which they distributed on that subject. These circulars did much to educate the mass of engine builders, users and operators in the theory of the steam engine.

Mr. Warrington was of Quaker parentage and possessed many of the best traits of that sect. He was conscientious, generous and peace-loving to a fault. Though undoubtedly a real genius he had none of the idiosyncrasies which so frequently accompany extraordinary talent. He was moderate in all of his views, a close thinker, well informed and well read. He contributed much to the development of the past half century.

HENRY WEBBER

Henry Webber was born in Bishops Tawton, Devonshire, England, on September 15, 1848. He was brought to this country when about three years of age, his parents settling in Dunmore, Pa., then known as Bucktown. He lived in Dunmore until the time of his death, January 2, 1920.

Mr. Webber served his apprenticeship as a machinist with the Pennsylvania Coal Co. in Dunmore shops for four years, then becoming connected with the Dixon Manufacturing Co., Scranton, Pa., where he was employed for five years as a machinist. At the end of that period he was advanced to the position of draftsman and later on made mechanical engineer. In this capacity he worked on the designing of hoisting engines and breaker machinery for the anthracite coal mines. He also had charge of designing and erecting blast-furnace engines for both the Lackawanna Coal & Iron Co. and the Lackawanna Iron & Steel Co.

In 1901 Mr. Webber resigned from this company to become mechanical and consulting engineer with the Finch Manufacturing Co., Scranton, Pa., which position he held at the time of his death.

Mr. Webber was for some time a member of the Engineering Society of Northeastern Pennsylvania, resigning two years before his death.

He belonged to several fraternal organizations. He became a member of our Society in 1886.

AMOS WHITNEY

Amos Whitney, one of the most conspicuous figures in the early growth of the New England machine-tool industry, a member of the A.S.M.E. since 1913, died on August 5, at the Poland Springs House, Poland Springs, Me. Mr. Whitney was born eighty-eight years ago on October 8, 1832, in Biddeford, Me. He came from distinguished Colonial and English ancestry. In this country the family has continuously held a prominent place, many of its members showing decided mechanical tastes, as Eli Whitney, inventor of the cotton gin, Baxter D. Whitney, the Winchendon machine builder, and Amos Whitney.

He was apprenticed at the age of fourteen to the Essex Machine Co. of Lawrence, Mass. Six years later he moved to Hartford, where he was employed at Colt's Armory, at that time the headquarters for many New England mechanics.

There he met Francis E. Pratt, who left shortly afterward to take charge of the Phoenix Iron Works, and Asa A. Cook. In 1853 Mr. Cook also went to the Phoenix Works as a contractor, taking young Whitney with him as a full partner. While there Mr. Whitney became closely associated with Mr. Pratt, who designed the "Lincoln" milling machines. Whether the idea of the milling machine came originally from Windsor, Vt., as some claim, or originated in Hartford, is difficult to prove at this time; but at any rate the machine is known as the Lincoln milling machine all over the world. In 1860 they determined upon entering business together. They rented a small room where, in addition to their regular employment, they started the manufacture of a little machine for winding thread known as a spooler. Within two years the business had increased to such an extent that they gave up their positions at the Phoenix Works and in 1865 erected their first building on the present site. In 1869 the Pratt & Whitney Co. was formed with a capital of $350,000, later increased to $500,000, and in 1893 reorganized with a capitalization of $3,000,000. In 1893 Mr. Whitney was made vice-president; later he was president, in which office he continued until January, 1901, when the control of the company was acquired by the Niles-Bement-Pond Co. Mr. Whitney remained as one of the directors.

Mr. Whitney's unbounded optimism was well displayed when the Pratt & Whitney Co. went through its first panic. It kept right on making standard machine tools, but selling almost nothing, until all the available storage room was filled. Then a large space was hired from the Weed Sewing Machine Co. and when this was filled another large space was hired in Colt's "West Armory" and this in turn was filled with finished machinery. It is well to note as a matter of history that when this immense stock finally did begin to move, it was practically sold out in 30 days.

Mr. Whitney took an important part in the development, in this country, of standard measuring instruments, one of the first moves being a determined effort to secure a standard inch block. His company purchased at considerable expense a standard rectangular bar, 1 in. square and 12 in. long, which had been used as a standard of measurement. Twelve 1-in. cubes were then made as accurately as possible and tested by the 12-in. piece. It was found that the twelve 1-in. cubes were not as long as the single bar, supposed to be exactly 12 in. long. Careful measuring and comparison with such standard instruments as were available led the company to believe that the individual inch-blocks were more nearly accurate than the long piece, and this was afterwards proved by the Rogers-Bond comparitor, which was developed in the Pratt & Whitney Works.

At the time of his death Mr. Whitney was president of the Gray Telephone Pay Station Co., and treasurer of the Whitney Manufacturing Co., Hartford, which was organized by the son, Clarence.

WILLIAM SCOTT WHITNEY

William S. Whitney, supervising engineer of the American Woolen Co., Boston, Mass., died suddenly on December 16, 1919. Mr. Whitney was born on March 4, 1854, in Beverly, Mass. He received his early education in the schools of Salem, Mass., and later attended the Massachusetts Institute of Technology, from which he was graduated with the class of 1874. He served his apprenticeship as a machinist with the Saco Water Power Machine Co., Biddeford, Me. He then engaged in the cotton manufacturing business, gaining a wide and varied experience in the many departments of the Boott Cotton Mills, Lowell, Mass. In 1876 he became superintendent of the Newton Cotton Mills, Newton Upper Falls, Mass., and two years later accepted the position of agent for the Pocasset Manufacturing Co., Fall River, Mass. From 1893 to 1899 he was connected with Tuttle Milne & Co., Fall River, as a cloth broker. At the close of that period he was appointed superintendent of the mechanical department of the Washington Mills in Lawrence, Mass. In 1899 he became supervising engineer of the American Woolen Co., Boston, which position he held at the time of his death.

Mr. Whitney became a member of the Society in 1911.

THOMAS EDWARD WILDER

Thomas Edward Wilder, president, Wilder & Co., Chicago, Ill., died on August 22, 1919. Mr. Wilder was born in Lancaster, Mass., on August 15, 1855. He was graduated from Worcester Polytechnic Institute in 1874 with the degree of B.S. Mr. Wilder's first position was with Walker, Oakley & Co., Chicago, where he worked for three years when he established his own leather business. In 1879 he became a member of the firm of Wilder & Hale, manufacturers of cut soles. Later the firm became Wilder & Co. with Mr. Wilder as president. He was also chairman of the board of directors of the Wilder Tanning Co., Waukegan, Ill., and of the board of managers of J. W. & H. P. Howard Co., Corry, Pa.

Mr. Wilder was a director of Counselman & Co., investment bankers, Chicago. He was an organizer of the Chicago Association of Commerce, the National Association of Tanners, and the Chamber of Commerce, U. S. A. He was vice-president of the National Rivers and Harbors Congress and director for the Great Lakes District. He was chairman of the American delegation to the sixth International Congress of Chambers of Commerce, Paris, 1914. He was a trustee of Worcester Polytechnic Institute and a member of the Union League and University Clubs of Chicago as well as many others. He became a member of our Society in 1917.

ERNEST HENRY WOOD

Ernest H. Wood, mechanical engineer, Eastern Malleable Iron Co., died on February 24, 1920. Mr. Wood was born on June 10, 1880, in Naugatuck, Conn. He was educated in the schools of Naugatuck and later attended Pratt Institute, from which he was graduated in 1903. Mr. Wood was associated throughout his professional career with the Eastern Malleable Iron Co., being transferred in 1905 from their Naugatuck plant to the works at Wilmington, Del. Mr. Wood's work was in industrial engineering, covering mechanical and electrical design, and the construction and maintenance of industrial plants and equipment connected with the iron and steel industry.

During the war Mr. Wood was appointed Director of Conservation for the Fuel Administration of Delaware. He became a member of the Society in 1913.

OROSCO CHARLES WOOLSON

Orosco C. Woolson, designing engineer and manufacturer, New York City, died on July 2, 1920, of heart failure. Mr. Woolson was born in Cuba on May 14, 1848. He received his early education at private schools in Massachusetts and later attended the Worcester Military Academy. From 1865 to 1869 he served an apprenticeship with the Hewes & Phillips Iron Works, Newark, N. J. He was associated for short periods with the John Cooper Engineering Works, Mt. Vernon, Ohio, as draftsman; the United States Government as draftsman on the Emery testing machines; the New York Elevated Railroad as inspector of construction; the Babcock & Wilcox Co. as draftsman, and the National Water Tube Boiler Co. as manager.

In 1893 Mr. Woolson established an engineering business in New York City specializing in designing and construction of combustion apparatus, power-boiler equipment and furnace accessories. At the time of his death he was also eastern representative of the Automatic Furnace Co., Dayton, Ohio.

Mr. Woolson became a member of the Society in 1886. He was also a member of the New York Railroad Club and the Board of Trade, Newark., N. J.

L. D. YORK

L. D. York, president of the Portsmouth Street Railroad and Light Company, Portsmouth, Ohio, died on September 19, 1920. Mr. York was born in England in 1847. He served an apprenticeship as roll turner in England and served also in that capacity in the Passaic Rolling Mills, Paterson, N. J., in 1869. From 1869 to 1889 he acted as superintendent in various rolling mills, his last connection being with the Burgess Steel and Iron Company. Mr. York joined the Society in 1889.

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NOTE

1. Names of authors and discussers, also those of 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 italics. Papers are indexed not under their titles but under their subject-matter.

[1] See MECHANICAL ENGINEERING, June 1920.

[1] See MECHANICAL ENGINEERING, March 1920.
[2] See MECHANICAL ENGINEERING, November 1920.
[3] See MECHANICAL ENGINEERING, May 1920.

[1] See MECHANICAL ENGINEERING, November, 1920.

[1] See MECHANICAL ENGINEERING, July 1920.

[1] See MECHANICAL ENGINEERING, January 1921.
[2] See MECHANICAL ENGINEERING, May 1920.
[3] See MECHANICAL ENGINEERING, January 1921.

[1] See MECHANICAL ENGINEERING, May 1920.